Mechanical simulation and installation position optimisation of a lifting cylinder of a scissors aerial work platform

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Abstract: The force of a lifting cylinder is the main factor of the alternating stress of scissors arms in a scissors aerial work platform, which seriously affects the fatigue life of structures. Aiming at this problem, the installation position parameters that affect the force of the lifting cylinder were determined. Then, the advanced modeling environment for performing simulation of engineering systems (AMESim) simulation model was used to simulate the lifting process of the lifting system, and the reliability of the simulation model was verified by the pressure test of the hydraulic system. Finally, based on the simulation model, the design of experiment method was used to optimise the installation position of the lifting cylinder in the design and development environment of AMESim, and the optimisation effect was verified through the test on a real vehicle. The result shows that the peak load of the lifting cylinder is reduced by \(<12\%\), and the average of the steady-state value decreases by \(<20\%\) after the installation position is optimised.

1 Introduction

The scissors aerial work platform is a special equipment to a manned aerial work, which has the characteristics of flexible operation, good spatial extensibility, large range of motion, safe and reliable, and field adaptability [1, 2]. It has two main working states as follows. First, the scissors mechanism be retracted, and the equipment can be moved quickly to the target site. Second, the scissors mechanism be extended, and it could lift objects to a high altitude to carry out the aerial work. The scissors aerial work platform is used widely, and related research and analysis have been done to understand its special work nature.

At present, research on the scissors aerial work platform mainly focuses on the scissors mechanism and the lifting cylinder. Petru et al. studied the influence of the acceleration and deceleration processes on structural alternating stress by using the Runge–Kutta method [3]. Dong and Xiaozhou studied the key parameters which affect the mechanical properties of the scissors mechanism [4, 5]. Hongguang optimised the key parameters of the scissors mechanism by analysing the mechanical characteristics and provided reference for the design of the scissors mechanism [6]. Hamidi and Islam analysed the stress and structural performance through the dynamic analysis on the scissors mechanism [7, 8]. Zhao et al. [9] made a quantitative analysis on the kinematic characteristics of the scissors mechanism, which provided a basis for the design of the scissors mechanism. Dongming analysed the limitation of a single-hydraulic cylinder-driving scissors mechanism, and put forward a design project of dual-cylinder driving [10]. Liu and Chaoqun analysed the force curve of the lifting cylinder through the dynamic simulation of the lifting mechanism [11, 12].

This paper takes the lifting cylinder of a scissors aerial work platform as the research object. A simulation model was built in AMESim to optimise the installation parameters that affect the force of the lifting cylinder.

2 Working principle of a scissors aerial work platform

Fig. 1 shows the structure of a scissors aerial work platform. It can mainly be divided into two parts: the walking system and the lifting system. The walking system is mainly the chassis, and the lifting system includes the scissors mechanism and the working platform. Among them, the scissors mechanism, composed of the hinged scissors arms and lifting cylinder, is connected with the chassis and the working platform through the hinges and a slider. As the main executing agency, the scissors mechanism achieves the lift of the working platform by converting the telescopic motion of the lifting cylinder piston to the vertical extension and recovery movement.

3 Simulation model of the lifting system of a scissors aerial work platform

3.1 Mechanical model of the lifting system

The lifting system of a scissors aerial work platform is a spatial symmetrical structure, which can be regarded as a plane structure. In order to analyse conveniently, the structure is considered as a rigid body without deformation, friction-free, ideal hinge-point constraint, and the mass centre overlap the geometric center. The simplified mechanical model of a lifting system is shown in Fig. 2.

In Fig. 2, the inner fork arm AD and the outer fork arm BC are articulated at point O1, forming the first group of the scissors arm, and the self-weight is \(W_1\); the inner fork arm CF and the outer fork arm DE are articulated at point O2, forming the second group of the scissors arm, and the self-weight is \(W_2\); the inner fork arm EH and the outer fork arm FG are articulated at point O3, forming the third group of the scissors arm, and the self-weight is \(W_3\); the inner fork arm GJ and the outer fork arm HI are articulated at point O4, forming the fourth group of the scissors arm, and the self-weight is \(W_4\).

A, B, C, D, E, F, G, H, I, and J express the outer points of the junction of the four groups of the scissors arms. Among them, hinge points A and B connect the first group of the scissors arm to the chassis, and the hinge points I and J connect the fourth group of the scissors arm to the working platform, and point O5 is the midpoint of the points I and J.

The lifting cylinder is connected to the scissors mechanism by the hinge points S and N. a represents the distance from the hinge point S to point O3 along the inner arm EH, b is the vertical distance from the hinge point S to the inner arm EH, c expresses the distance from the hinge point N to point O5 along the inner arm

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According to Fig. 2, the principle of virtual work can be represented by

\[ \sum (F_i \delta \alpha_i + F_{1i} \delta \beta_i + F_{ci} \delta \gamma_i) = 0 \]  \hspace{1cm} (1) 

The virtual displacements in (2):

- \( \delta_{o1} \) is the virtual displacement of the hinge point O1 along the Y-axis, 0.5icos \( a \delta \alpha \) (l is the length of fork arms);
- \( \delta_{o2} \) is the virtual displacement of the hinge point O2 along the Y-axis, 1.5icos \( a \delta \alpha \);
- \( \delta_{o3} \) is the virtual displacement of the hinge point O3 along the Y-axis, 2.5icos \( a \delta \alpha \);
- \( \delta_{o4} \) is the virtual displacement of the hinge point O4 along the Y-axis, 3.5icos \( a \delta \alpha \);
- \( \delta_{o5} \) is the virtual displacement of the hinge point O5 along the Y-axis, 4cos \( a \delta \alpha \);
- \( \delta_{o6} \) is the virtual displacement of the hinge point O6 along the Y-axis, \( a \delta \alpha \);
- \( \delta_{o7} \) is the virtual displacement of the hinge point S along the Y-axis \( (2l - a) \cos \alpha - b \sin \alpha \delta \alpha \);
- \( \delta_{o8} \) is the virtual displacement of the hinge point S along the X-axis \( (a - 0.5l) \cos \alpha - b \sin \alpha \delta \alpha \);
- \( \delta_{o9} \) is the virtual displacement of the hinge point N along the Y-axis \( (0.5l + c) \cos \alpha + d \sin \alpha \delta \alpha \);
- \( \delta_{o10} \) is the virtual displacement of the hinge point N along the X-axis \( d \cos \alpha - (0.5l + c) \sin \alpha \delta \alpha \).

The length \( L \) of the lifting cylinder can be obtained by

\[ L = \sqrt{(a + c) \cos \alpha + (b + d) \sin \alpha} \]

\[ + \frac{1}{2} \left( (2l - a - c) \sin \alpha + (b + d) \cos \alpha \right) \]  \hspace{1cm} (3)

The relationship between \( \theta \) and \( \alpha \) can be given by

\[ \cos \theta = \frac{(a + c) \cos \alpha + (b + d) \sin \alpha}{L} \]  \hspace{1cm} (4)

\[ \sin \theta = \frac{(2l - a - c) \sin \alpha + (b + d) \cos \alpha}{L} \]  \hspace{1cm} (5)

The protruding speed of the lifting cylinder is \( v = L' \), and it can be expressed as

\[ \frac{v}{(dt/dt)} = \frac{(b + d) \cos \alpha - (a + c) \sin \alpha}{L} \cos \theta \]

\[ + \frac{1}{2} \left( (2l - a - c) \sin \alpha + (b + d) \cos \alpha \right) \sin \theta \]  \hspace{1cm} (6)

Substituting (3)–(6) into (2), the force \( P \) of the lifting cylinder can be shown as

\[ P = \frac{4l(W_1 + P_1) + 3.5W_1 + 2.5W_2 + 1.5W_3 + 0.5W_4}{v} \]

\[ \times \cos \alpha \frac{da}{dt} \]  \hspace{1cm} (7)

According to (7), in case the protruding speed \( v \) of the lifting cylinder is constant, the force \( P \) is related to the self-weight of structures, the length \( l \) of fork arms, the angle \( \alpha \), and \( da/dt \). From (6), \( da/dt \) is affected by the installation position parameters of the lifting cylinder. It can be concluded that, under the premise of keeping the product structures unchanged, the main factors that affect the force \( P \) of the lifting cylinder are four installation position parameters: \( a, b, c, \) and \( d \).

### 3.2 Simulation model of the lifting system

It can be seen from Fig. 3 that the AMESim simulation model was established on the basis of the mechanical model of the lifting system as shown in Fig. 2 [14], which applied the sub-model of ‘Hydraulic’, ‘Mechanical’, ‘Hydraulic Component Design’, and ‘Signal Control’ [15]. This model can also be divided into two parts: the hydraulic system and the mechanical system.

#### 3.2.1 Hydraulic system

From the hydraulic system, as shown in Fig. 3, the gear pump transfers power to the hydraulic system, and the overflow valve controls the system pressure. The input and output of the system are controlled by a two-position three-way magnetic valve and two non-returning valves, and the control signal of the magnetic valve is simulated by an electrical signal. A
3.2.2 Mechanical system: The mechanical system, as shown in Fig. 3, is composed of four inner fork arms, four outer fork arms, and a working platform, and it is connected according to the mechanical model shown in Fig. 2.

The parameters of the simulation model are set up according to the actual conditions.

4 Mechanical simulation of the lifting cylinder

Considering the complexity of the simulation, the model was simplified to ignore the back pressure and leakage of the hydraulic system. Table 1 shows the main parameters of the simulation model and Table 2 shows the self-weight of structures and load applied on the working platform.

According to the actual working conditions, the velocity of the piston is $1.3 \text{ m min}^{-1}$ and the oil temperature is $55^\circ\text{C}$.

The reliability of the simulation model was verified by testing the force of the lifting cylinder, and the force was indirectly tested through the pressure of the hydraulic system.

Fig. 4 shows the pressure test scene of a hydraulic system. The CHPM Reynolds intelligent tester (precision $\pm1\%$) was connected to the lifting solenoid valve in the chassis to collect the pressure data. The machine was run for a period of time to make the oil temperature reach the normal working temperature $55^\circ\text{C}$ before data collection. When collecting data, the working platform was lifted from the lowest position to the highest, and a load of $450 \text{ kg}$ was applied.

Fig. 5 shows the force curves of the simulation and experiment of the lifting cylinder. At the initial stage, the two curves rise sharply and peak at $\sim3\text{s}$, of which the peak value of the simulation curve is $\sim8\%$ lower than that of the experimental curve; the two curves descend slowly to the steady state, and the steady-state mean value of the simulation curve is $\sim16\%$ lower than that of the experimental curve. The deviation between the two curves is mainly related to the idealised treatment of the simulation model and the friction between structures. The peak value and the general trend of the two curves coincide, which verify the reliability of the simulation model.

From the experimental curve in Fig. 5, the peak load of the lifting cylinder is $\sim30\%$ higher than the average value of the steady state, which affects the fatigue life of the structure. In the case of keeping the structural parameters constant, the lower the force of lifting cylinder is, the better it will be to reduce the energy consumption and improve the structural performance. In order to reduce the peak load, the next content optimised the installation position that affects the force of the lifting cylinder.

5 Installation position optimisation of the lifting cylinder

The design of experiment (DOE) method is a theoretical statistical method for arranging experiments and processing experimental data by which the experimental results and conclusions can be obtained by reasonably reducing the complexity and frequency of experiments [16]. From (6) and (7), it is understood that the mechanical model of the lifting cylinder is a multi-variable model.
Based on the simulation model of the lifting cylinder, the DOE method was used in the design and development environment of AMESim to optimise the system by using four installation position parameters of the lifting cylinder as the design variables, and the minimum peak load of the lifting cylinder as the optimisation index.

Table 3 shows the range of four installation position parameters a, b, c, and d. When optimising, the minimum height of the working platform is set as 7430 mm, and the minimum length of the lifting cylinder as 330 mm. Table 4 shows four installation position parameters of the lifting cylinder before and after optimisation.

The optimisation result was applied to the simulation model and compared with the simulation curve before optimisation. From Fig. 6, it can be seen that the force of the lifting cylinder is obviously reduced after optimisation, and its peak load is reduced from 112.19 to 89.23 kN, which is 20.4% lower than that before optimisation, and the average of the steady-state value is reduced by ∼14%. The force of the lifting cylinder is improved.

In order to verify the optimisation effect, a machine was used to change the installation position parameters of the lifting cylinder according to the optimisation result, and tested the force of the lifting cylinder under the same working condition.

From Fig. 7, it can be seen that the peak load of the lifting cylinder is reduced from 121.5 to 106.4 kN after optimisation, which is ∼12% lower than that before optimisation, and the average force of the steady state is reduced by ∼20%. The force of the lifting cylinder is obviously reduced after the optimisation, which is basically consistent with the simulation result. Also, the machines produced according to this optimisation scheme have been put into the market and the feedback is good.

6 Conclusion
The authors have established the mechanical model of the lifting system of a scissors aerial work platform, and it was found that the force of the lifting cylinder was closely related to four installation position parameters.

The simulation result proved that the force of the lifting cylinder reached the peak value at ∼3 s, and the peak load was ∼30% higher than the mean value of the steady state. Also, it was proposed to reduce the peak load by optimising the installation position of the lifting cylinder.

The installation position parameters of the lifting cylinder were optimised on the basis of the simulation model, and the optimisation effect was verified through the test on a real vehicle. The test result shows that the peak load of the lifting cylinder decreases by ∼12% compared with that before optimisation, and the average of the steady-state value is ∼20% lower than that before optimisation.

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References

[1] Pan, Q., Zhang, Z., He, S.H., et al.: ‘Fatigue life analysis and forecast of scissors arms in scissors aerial work platform’. Food Mach., 2017, 33, (5), pp. 119–124

[2] Yin, B.: ‘The design and structural optimization of the scissors-like mobile platform’ (Changsha University of Science and Technology, Changsha, 2012), pp. 1–4

[3] Petru, M., Sevick, L., Masin, I., et al.: ‘Dynamic analysis of lifting platform construction for car relocation’ (Springer International Publishing, Basel, 2014), pp. 517–524

[4] Dong, R.G., Pan, C.S., Hartsell, J.J., et al.: ‘An investigation on the dynamic stability of scissors lift’. Open J. Safety Sci. Technol., 2012, 02, (1), pp. 8–15

[5] Hu, X.Z., Hu, J.K., He, G.H.: ‘The modeling of scissors elevation mechanism and research of key parameters’. Mech. Res. Appl., 2006, 19, (4), pp. 84–85

[6] Deng, H.G., Sun, D.G., You, S.K., et al.: ‘Modeling and key parameters research of fork lift platform’. Mech. Electr. Eng. Technol., 2005, 34, (7), pp. 20–22

[7] Hamidi, B.: ‘Design and calculation of the scissors-type elevating platforms’, Open J. Safety Sci. Technol., 2012, 4, pp. 22–25

[8] Islam, M.T., Yin, C., Jian, S., et al.: ‘Dynamic analysis of scissors lift mechanism through bond graph modeling’. IEEE/ASME Int. Conf. Advanced Intelligent Mechatronics, Besancon, France, 2014, pp. 1393–1399

[9] Zhao, J.S., Chu, F., Feng, Z.J.: ‘The mechanism theory and application of deployable structure based on SLE’. Mech. Mach. Theory, 2009, 44, (2), pp. 324–335

[10] Sun, D.M., Dong, W.M., Li, S.: ‘Design of heavy-loading hydraulic fork lifting deck propelled symmetrically’. Mach. Des. Manuf., 2006, 6, pp. 23–24

[11] Liu, T., Sun, J.: ‘Simulative calculation and optimal design of scissors lifting mechanism’. 2009 IEEE Control and Decision Conf. (CCDC V9), Guilin, China, 2009, pp. 2133–2136

[12] An, C.Q.: ‘Simulation study of an electric self-propelled scissors aerial work platform’. PhD thesis, Nanjing University of Science and Technology, 2012

[13] Ni, S.H., Zhang, C.: ‘Development of a new type hydraulic lifting platform’. Constr. Mach. Equip., 2011, 42, (1), pp. 59–61

[14] Fu, Y.L., Qi, X.Y.: ‘Reference manual of LMS imagine. Lab AMESim system modeling and simulation’ (Heihang University Press, Beijing, 2011)

[15] Shi, R.S., Liang, Q.: ‘Computer simulation of the hydraulic control lift platform with AMESim’, Machinery, 2016, 43, (8), pp. 33–35

[16] Min, Y.N.: ‘Application guide for design of experiment (DOE)’ (China Machine Press, Beijing, 2011)