Simulation research on pressure shock of load sensing system of large boom tower crane

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Abstract. Large boom tower cranes have large masses, heavy loads, and large inertial dynamic loads during the conversion of working conditions. Even if there is a buffer device, hydraulic shock will still occur. The problem of hydraulic shock seriously affects the leakage of hydraulic oil and decreases system reliability. Now the closed hydraulic system of a certain type of boom tower crane is taken as the research object, and the joint modeling of machinery and hydraulic is completed in AMESim. Without changing the basic hydraulic components, the hydraulic system of the whole machine just adjusts the ratio of the pulley block or changes the hydraulic speed regulation method to study the hydraulic shock of the servo-sensitive hydraulic system. AMESim simulation results show that in this load-sensitive system, the double-rate hoisting of the pulley block is compared with the single-rate hoisting of the pulley block, the maximum dynamic load generated by the load off the ground is increased by 5.4%; Speed regulation of the variable pump and variable motor combination is compared with the speed control of only variable motor, the maximum dynamic load of the hydraulic system is reduced by 7.6%.

1. Introduction

Boom tower cranes have super-large lifting moments, super-high lifting heights, high lifting speeds, and small tail slewing radius, which have unparalleled advantages in the construction of high-rise buildings or narrow construction sites. When the working condition of the boom tower crane is switched, the pressure of the tower crane hydraulic system will rise temporarily. Even at a slower speed, due to the huge inertial dynamic load, the pressure in the oil return chamber of the motor will instantly rise to a high level [1], and the oil intake chamber of the motor will not absorb enough oil, which will cause a pressure shock to the system. The occurrence of hydraulic shock will seriously affect the performance and life of each key component in the system and increase the failure rate of the system [2].

In view of the hydraulic shock generated during the lifting process of a large inertia load system, there have been many researches on the hydraulic shock by adding or replacing different hydraulic components to the system. Qing Xiao et al. proposed a parallel rotary buffer valve in a two-chamber hydraulic motor, which can buffer the instantaneous high pressure after the impact pressure is higher than the set pressure of the buffer valve [3]. Jingfang Zhang et al. proposed an accumulator with a throttle valve as a buffer device for the load-sensitive hydraulic system of a coal shearer. Through analysis and optimization of the accumulator parameters, the hydraulic shock is reduced to a certain extent [4]. Linyi Gu et al. used load-sensitive modulation valves instead of throttle valves to improve the starting and braking characteristics of the load [1], and so on.

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This paper conducts simulation research on the closed load sensitive system of large scale boom tower cranes, and explores the maximum additional dynamic load generated by the mechanical system when using different hydraulic speed regulation method and single-rate or double-rate lifting, and the resulting hydraulic system pressure overshoot and the resulting pressure shock. Combined with STL2400 full hydraulic boom tower crane produced by Fushun Yongmao Construction Machinery Co., Ltd, using AMESim software as a platform, a simulation model of the lifting mechanism including hydraulic closed system and boom and other mechanical structures is built to simulate the lifting process of the crane under maximum load. During the lifting process, study the hydraulic impact of switching work condition to the system during different hydraulic speed regulation method modes and different load ratios during lifting, thus providing a theoretical basis for the improvement of this type of tower crane hydraulic system.

2. Modeling of lifting mechanism

During the lifting process of the crane, the motor in the hydraulic drive circuit rotates and is transmitted to the drum through the coupling and the reducer, drum rotates to realize the retracting of the steel rope, which in turn drive the load up and down. If the pulley is added to the hook, a double-rate lifting pulley group is formed to realize the function of saving effort. The hydraulic driving device and the hoisting drum are installed on the tower crane balance arm platform, and the fixed pulley in the working device is installed on the boom jib. When the lifting mechanism is converted under different working conditions, additional inertial dynamic load will be generated. The inertial dynamic load will cause the change of the internal tension of the wire rope, causing the boom to vibrate at a high frequency and small amplitude around its hinge point with the balance arm, resulting in pressure shock inside the hydraulic system. The above factors affect each other, and in combination affect the generation and attenuation of dynamic load effects. It shows that the hydraulic shock brought by the conversion of the working condition of the hoisting mechanism is the result of a series of complex machine-hydraulic coupling effects [5]. Using mechanical and hydraulic joint modeling can better simulate the hydraulic impact and some dynamic characteristics brought by the boom crane when it is lifted.

The lifting circuit of STL2400 boom tower crane is composed of two Rexroth swash plate axial piston pumps A4VG and four Rexroth swash plate axial plunger motors A6VM connected in parallel. Four motors are used to drive the load together, and the load torque is evenly divided by the four motors. This avoids hydraulic system leakage, increased oil temperature rise, and damage to hydraulic components caused by excessively high working pressure when only one motor drives a large load torque [6].

In order to facilitate the description of the working principle of this closed lifting hydraulic circuit, after omitting part of the control oil circuit, it is simplified into a closed circuit with a single variable pump driving a single variable motor as shown in figure 1.

In the servo-variable pump, the spool acts upon receiving the displacement signal input by the electrical-mechanical converter, changes the flow area of the control side orifice and then changes the flow into the two chambers of the displacement control cylinder, and drives the piston to extend, through the calculation of the function formula, the displacement of the piston extension is converted into the displacement control signal of the main pump. The established model uses a spring damping element to replace the feedback rod in the pump to achieve the feedback function. This displacement signal can make the valve sleeve return to zero position, the control cylinder is also in a "floating" state [7], the pump displacement is fixed, and the feedback adjustment function is realized. The control piston of the servo-sensitive pump is built with a "piston element with rod cavity and a return spring", the function of the return spring is to ensure that the control piston returns automatically when there is no control signal input or the control pressure oil is cut off due to overload, to make the displacement of the main pump 0. Based on the above principles, a hydraulic lifting system model with two pumps driving four motors is built.
1. Variable pump
2. Variable motor
3. Pump servo variable mechanism
4. Motor servo variable mechanism
5. Charge pump
6. Charge relief valve
7. Pressure shut-off valve
8. High pressure relief valve
9. Flush valve

Figure 1. Simplified schematic diagram of closed hydraulic drive system.

The torque and speed output by the motor are firstly decelerated by four Brevini reducers, and then driven by the meshing transmission of the small gear of the motor output shaft and the large ring gear of the reel to achieve two-stage deceleration.

The main parameters of the system are given in table 1.

Table 1. The main parameters of the system.

| Name                              | Parameter          |
|-----------------------------------|--------------------|
| Control pressure of motor         | 1.3~2.3MPa         |
| Maximum speed of motor            | 3400rot/min        |
| System maximum lifting pressure   | 31MPa              |
| Rated pressure of pump            | 40MPa              |
| Geometric displacement of pump    | 180mL              |
| Maximum output flow of pump       | 450L/min           |
| Geometric displacement of motor   | 200mL              |
| Rated pressure of motor           | 40MPa              |
| Reduction ratio of Brevini reducer| 20.4               |
| Reduction ratio of motor and drum | 170/23             |

STL2400 lifting pulley group is a single ratio or double ratio pulley block composed of guide pulley, force measurement pulley, fixed pulley and movable pulley. Generally, the pulley group adopts double ratio. After the wire rope passes through the fixed pulley and the movable pulley, the rope end is fixed to the tip of the arm. In some occasions where light load and high-speed lifting are required, the pulley block can adopt a single ratio. The lifting rope is directly connected to the hook after passing through the arm-tip pulley, and a complete lifting mechanism model of the boom tower crane is built according to the above principle as shown in figure 2.
Figure 2. AMESim model of complete hoisting mechanism.

The simulation takes the crane lifting weight as the designed maximum lifting weight of 100t, and the boom is at the minimum swing angle of 17°. The complete simulation working conditions include six working stages: accelerated lifting, constant speed lifting, brake hovering, accelerated descending, constant speed descending, decelerating descending until it stops. The real speed regulation process is very short during lifting [8]. In order to visually observe the changes of the simulation curves of various variables, the duration of the speed regulation process in each working stage is specially set to 5s. Take the signal gain of 0.5, the pump and motor variable displacement mechanism control signal source in the model is set according to figure 3.

Figure 3. Lifting mechanism control signal.
As shown in figure 4, the model simulation verifies that the simulation curve in the startup phase has obvious fluctuations, which is caused by the additional dynamic load caused by working condition conversion in the startup phase.

Figure 4. Motor oil outlet pressure.

3. Simulation analysis

3.1. Influence of single-rate or double-rate on hydraulic dynamic characteristics during lifting

In practice, the boom tower crane changes the position of the hoisting rope to realize the conversion of the pulley block from double-rate to single-rate. Figure 5 shows the AMESim model of the mechanical part of the working device when using single-rate.

Figure 5. Model of single-rate lifting working device.
Set up a comparison simulation of single-rate and double-rate lifting, where the variable pump control current in the single-rate comparison group is half of the double-rate comparison group, as shown in figure 6(a), to ensure that the loads in the two comparison groups move at the same speed. As shown in figure 6(b).

**Figure 6.** Pump control current and load lifting speed at different rates: (a) variable pump control current; (b) lifting speed curve comparison.

The simulation results are shown in figure 7.

**Figure 7.** Simulation comparison under different magnification conditions: (a) contrast curve of inertial load during lifting; (b) pressure curve of high-pressure side of hydraulic system.

The dynamic performance indicators of the hoisting system at different rate are shown in table 2.
Table 2. Dynamic performance index of lifting system under single-rate or double-rate.

| Working condition    | Magnification | Maximum dynamic load (kN) | Maximum overshoot of dynamic load (%) | Peak pressure (MPa) | Maximum overshoot of pressure (%) | Adjusted time (s) |
|----------------------|---------------|---------------------------|---------------------------------------|---------------------|-----------------------------------|------------------|
| Lift off the ground  | Single-rate   | 77.0                      | 78.0%                                 | 10.5                | 50.0%                             | 2.27             |
|                      | Double-rate   | 81.2                      | 82.9%                                 | 7.0                 | 45.8%                             | 2.16             |
| Lifts twice after hovering | Single-rate | 5.0                        | 5.1%                                  | 8.8                 | 25.7%                             | 1.37             |
|                      | Double-rate   | 6.9                        | 7.0%                                  | 5.9                 | 22.9%                             | 1.16             |

When hoisting with double-rate, the maximum dynamic load generated in the stage of the load leaving the ground and the stage of second lift after hovering are both slightly larger than the single-rate hoisting conditions, increasing by 5.4% and 3.8% respectively. The single-rate of the system under dynamic load is slightly better than the double-rate, but on the vibration waveform, the single-rate ratio lags the double-rate, and the adjustment time required to return to a stable state is also longer. As far as the system pressure is concerned, when the single-rate is raised, the system steady-state pressure is increased by 46% compared to the double-rate. The system is under high-pressure working conditions for a long time, which is not conducive to the heat dissipation of the closed system and will cause oil leakage problems.

3.2. Influence of speed regulation method on dynamic characteristics

In this example, the variable pump-variable motor combination speed regulation method is used to simulate the load lifting phase, and the simulation results are compared with the results of using a single variable pump to drive the load lifting. In order to ensure that the loads in the control group have the same lifting speed, the control signals of the variable pump and variable motor are set according to figure 8(a). The load speed curve obtained in this way and the load speed curve obtained using only pump variables are placed in figure 8(b).

![Figure 8](image1.png)

(a) Control current of the pump (b) Load lifting speed (m/s)

**Figure 8.** Comparison of control signals and load lifting speed of different variable modes: (a) control current of the pump; (b) contrast curve of lifting speed.

Figure 9 shows the simulation comparison curves under different speed adjustment modes.
Figure 9. Simulation comparison under different speed adjustment modes: (a) contrast curve of inertial load during lifting; (b) pressure curve of high-pressure side of hydraulic system.

Table 3 gives the dynamic performance indexes of the lifting system under different speed control modes.

Table 3. Dynamic performance of hoisting system under Different speed regulation modes.

| Speed mode          | Maximum dynamic load (kN) | Maximum overshoot of dynamic load | Peak pressure (MPa) | Maximum overshoot of pressure | Adjusted time (s) |
|---------------------|---------------------------|----------------------------------|---------------------|-------------------------------|------------------|
| Adjust speed with pump | 53.5                      | 54.6%                            | 7.1                 | 47.9%                         | 1.7              |
| Adjust speed with motor | 49.7                      | 50.7%                            | 13.3                | 26.7%                         | 0.9              |

In this hydraulic lifting mechanism, the variable speed ratio of the variable pump-variable motor combination is only with variable pump speed control, the maximum dynamic load of the hydraulic system is reduced by 7.6%, and the adjustment time is shortened by 89%. However, the high-pressure side pressure (10.5MPa) under steady state increased by 118.8% compared with the single pump speed control group (4.8MPa), and the peak pressure increased by 87%. It can be seen that using variable pump speed regulation under the same load condition than the variable pump-variable motor combination speed regulation, the steady-state working pressure is reduced by about 50%, and some closed-system heating can be reduced. Therefore, on the premise of meeting the speed requirements, variable pump speed regulation method should be selected.

4. Conclusion

This article takes the full hydraulic boom tower crane lifting mechanism with a maximum lifting torque of 24000N·m as the research object, and uses AMESim simulation software to deal with the hydraulic shock caused by the inertial dynamic load during the conversion of the working condition of the lifting mechanism. Simulation is a means to analyze the influencing factors of the dynamic characteristics of the hoisting mechanism. The inertial dynamic load generated during the working condition change of the lifting mechanism will cause the pressure impact of the hydraulic system, and the pressure overshoot will reach 50.3% of the steady state working pressure.
Under the same lifting load conditions, the maximum dynamic load of single rate is reduced by 5.2% compared with double rate, which has no obvious effect on improving the force of the mechanism. However, the steady-state pressure has increased by 46%, which will bring the problem of increased system temperature and increased leakage. Therefore, when high-speed or light-load is not required, single rate is not recommended, but double rate should be used;

The hydraulic circuit of the hoisting mechanism is a closed hydraulic system that uses variable pump and variable motor speed regulation. Under the same load condition, a single variable pump speed control is compared with variable pump-variable motor combination speed control. The steady-state working pressure is reduced by nearly 50%, which can effectively reduce the closed system heating. Therefore, under the premise of meeting the speed requirements, the variable pump should be used for speed regulation first.

The above data can provide useful conclusions for the structure and hydraulic system improvement of this type of tower crane. In future research, the dynamic characteristics of the switching of working conditions during hoisting can be further optimized through comprehensive consideration of various parts in the hoisting mechanism. The load-sensitive closed hydraulic system model of the lifting mechanism of the boom tower crane in the article can provide a reference for the optimal design and research of the system in the future.

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