Analysis and Optimization of Pressure shock and Cylinder Stroke Deviation in Open Circuit Hydraulic System of Concrete Pump

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Keywords: Concrete pump, Hydraulic system, Cylinder stroke, Pressure shock

ABSTRACT

This research mainly investigates the pressure shock and the cylinder stroke deviation in the open circuit hydraulic system of the concrete pump. According to the actual open circuit hydraulic system of concrete pump, a simulation model is developed with the software AMESim. Through the analysis of the simulation results, the cause of the above problems is the unmatched switching time of the pumping circuit and distributing circuit. Based on the simulation model, a lot of simulations have been done to find out the relationships between the restrictor diameters and the pressure shock and the cylinder stroke deviation. On the basis of these laws, several kinds of proper collocations are sought out. The experimental results validate that the best collocation can reduce the pressure shock and the cylinder stroke deviation by 20%. The lifespan and the efficiency are bettered accordingly.

INTRODUCTION

Concrete pump trucks have been widely used in infrastructural construction which greatly reduce the labor intensity of construction workers and improve the construction efficiency. The operation pressure is generally high in the concrete pump hydraulic system. Pressure shock is a serious problem and it is even worse for the open circuit concrete pump hydraulic system [Stefan Jacobsen, 2008]. The pressure shock would generate vibration and noise, which do much harm to the hydraulic system and components of the concrete pump truck. The pressure shock is generally caused by two reasons. First, when the two pumping cylinders operate alternatively, the main pump displacement is fixed, so when the directional valve works in the transitional and median position, the main pump cannot be unloaded. It leads to the pressure shock during the valve switching [JIE Lin-feng, 2010]. Second, when the S-pipe is switching, the lower pressure cylinder is loaded instantly, and the same as the main pump. Therefore, the sudden change of the load can also cause the pressure shock [Chunlei Song, 2005]. Further, the types of concrete are different and the velocity of the cylinder piston is also varied in many working conditions, so it results in the cylinder stroke deviation. In extreme cases, the piston even hits the cylinder head or cannot travel enough. This will reduce the lifetime of the cylinder [FU Lei,2000].

In order to lower the pressure shock, some companies use the closed circuit hydraulic system including the *SN* control method or free-flow hydraulics [ChenYing, 2006][The Putzmeister Group, 2010]. However, the closed circuit hydraulic system is complex and costly. There are also other problems like overheating oil and air suction in this circuit. So the open circuit hydraulic system with its benefits such as low cost and easy maintenance is still playing an important role in the market. Therefore the researches on the open circuit concrete pump hydraulic system have practical significance [Hongyu

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Wei, 2005][Chunlei Song, 2005].

In this research, the cause of the pressure shock and cylinder stroke deviation in the open circuit concrete pump hydraulic system is found out through simulation. Moreover, after analyzing the simulation results of the different restrictor diameters in the two circuits, an experimentation validated method to solve the problems is proposed to provide guidance for designing the hydraulic system of the concrete pump truck.

SYSTEM PRINCIPLE

Figure 1 shows the open hydraulic circuit of the concrete pump hydraulic system. The pumping cylinder 4 is extending while the pumping cylinder 5 is retracting simultaneously. There is a trigger port in either of the pumping cylinder. When the piston of the pumping cylinder surpasses the trigger port, the pressure differential valve works. Then the hydraulic pilot directional valve 14 is switched to the left working position. Then the main hydraulic directional valve is switched to the right working position. Therefore, the distributing cylinders can switch the S-pipe. Meanwhile, they send the switching signal to the hydraulic pilot directional valve 2 and make it work at the left position. And then, the main hydraulic directional valve 3 is switched to the right working position, which makes the pumping cylinder 4 retract and the pumping cylinder 5 extend. This circle of working process makes the concrete pumped continuously.

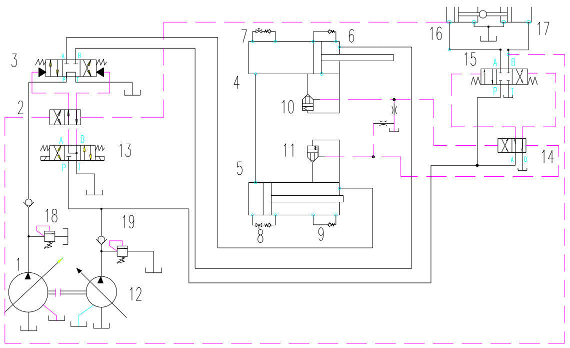

Fig. 1 The hydraulic circuit
1- the variable displacement pump 2,14-hydraulic pilot directional valve 3,15- main hydraulic directional valve 4,5-pumping cylinder 6,7,8,9-U-tube restrictor 10,11- pressure differential valve 12- constant pressure pump 13- electromagnetic directional valve 16,17-distributing cylinder 18,19- relief valve

THE CURRENT PROBLEMS AND BAD EFFECTS

There is high pressure shock in the hydraulic system when the pumping cylinders and distributing cylinders are reversing. It will be even worse when the working pressure is high. Figure 2(a) shows the measured results of the system pressure and the cylinder displacement of the concrete pumping truck. As we can see from it, when the working pressure is 17MPa, the peak pressure reaches 24MPa, 40% higher than the working pressure.

The high pressure shock reduces of the life span of the system. There is no perfect theory to calculate the life time under high pressure shock precisely. But it can be estimated using the modification of Langer Equation. The basic form of the equation is as follows [Wang Xueyan,1993]

$$N = [\frac{E \ln \frac{1}{1-\psi}}{4(nK\sigma_a - \beta\varepsilon_d\sigma_F)}]^2 \qquad (1)$$

In order to use Equation (1) to calculate the fatigue life time under the pressure shock, the basic Langer formula need to be modified. In Eq.(1), the item $nK\sigma_a$ represents the common peak fatigue stress belonging to the static load predicament, so the dynamic load coefficient K_d can be added to calculate the fatigue life time under the pressure shock. On the other hand, the item $\beta\varepsilon_d\sigma_F$ is suitable for the common fatigue load carrying capacity. A limit factor of the impact fatigue K_e is also needed when considering the effect on the material fatigue induced by the impact load. $K_e = \sigma_e / \sigma_F$, σ_e is the limit amplitude of the impact fatigue.

Therefore, the Equation (1) can be transformed to

$$N = [\frac{E \ln \frac{1}{1-\psi}}{4(nK_d K\sigma_a - \beta\varepsilon_d K_e\sigma_F)}]^2 \qquad (2)$$

Seen from Equation (2), the higher pressure shock is, the fewer numbers of the cycles and the shorter life span is.

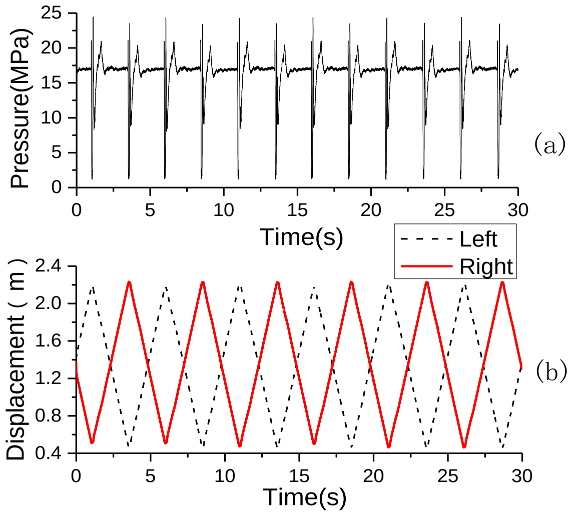

Fig. 2 System pressure（a）and cylinder displacement（b）

The stroke of pumping cylinders varies in different working conditions. When the flowrate is low, the pumping speed is low and the stroke of cylinders becomes short. As shown in Fig.2(b), when the flowrate is 150L/min, the piston displacement of the pumping cylinders is between 0.4m and 2.2m. The stroke of the two cylinders is less than 1.8m. However, the cushioning zone of the cylinder's rear end is from0.19m to 0.26m, so it can be seen that in the retract stroke of the cylinders, the piston cannot reach the cushioning zone.

This reduces the pumping efficiency because of the short cylinder stroke. The efficiency of one cycle can be calculated with the Equation (3).

$$e = \frac{l}{L} \times 100\% \tag{3}$$

It is inferred from Eq.(3) that the long stroke is good for improving the pumping efficiency.

In the actual work process, it is expected that the pressure shock is reduced as low as possible and the stroke of cylinders is increased as long as possible under the limit that the piston cannot hit the cylinder head.

SIMULATION MODEL AND ANALYSIS

1. Simulation Model

AMESim is an excellent multidisciplinary simulation software, which is used to simulate the open circuit concrete pump hydraulic system in this research. The simulation model mainly contains the following components, the variable displacement pump, the constant pressure pump, all the valves, pumping cylinders, distributing cylinders and distributing mechanism. Figure 3 shows the simulation model of the hydraulic system of concrete pump.

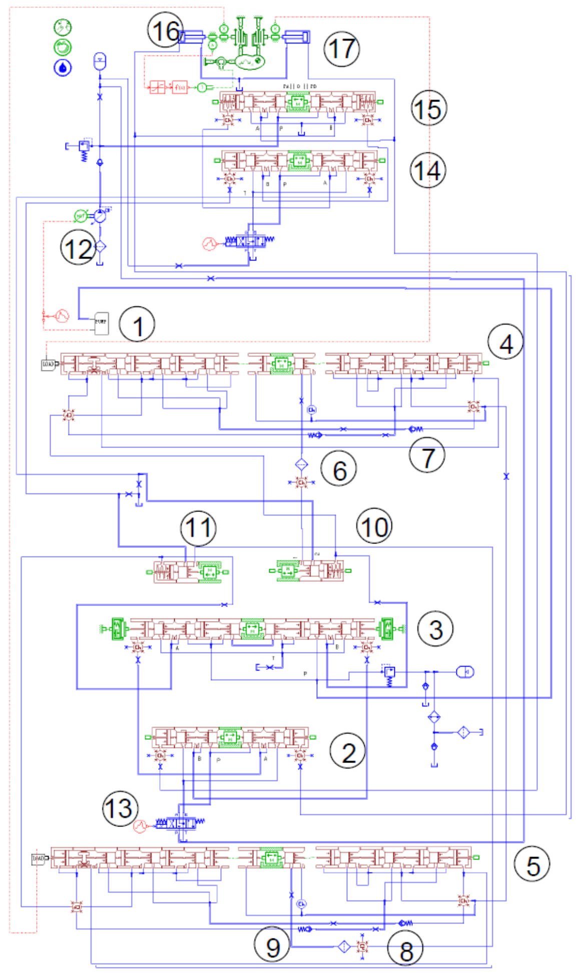

Fig. 3 Simulation model of concrete pump hydraulic system
1- the variable displacement pump 2、14-hydraulic pilot directional valve 3、15- main hydraulic directional valve 4、5-pumping cylinder 6、7、8、9-U-tube restrictor 10、11- pressure differential valve 12- constant pressure pump 13- electromagnetic directional valve、16、17-distributing cylinder

2. Analysis of switching time sequence

Figure 4 shows the simulation results when the working flowrate is 300L/min. It contains the following curves, the pressures in the control chambers of the four directional valves, the pressure of the variable displacement pump and the displacements of the pumping cylinder and the distributing cylinder.

The switching time sequence of the hydraulic system shown in Fig. 1 can be obtained from Fig. 4. When the piston of pumping cylinder 4 surpasses the trigger port, the pressure differential valve 10 works (At the time point A in Fig. 4). The pressure oil flows into the left chamber of the hydraulic pilot directional valve 14, so that the pressure in the left chamber increases and the valve 14 switches. After the pilot valve 14 switches, the main hydraulic directional valve 15 begins to switch. Then the high pressure oil flows to the rear chamber of the distributing cylinder

17 and makes it extend (At the time point B in Fig. 4). At the same time, the distributing cylinder 16 retracts. There is also pressure oil coming from the rear chamber of the distributing cylinder 17 at this time, which flows into the left control chamber of the hydraulic pilot directional valve 2 and leads to the increase of the pressure in the chamber. It is about 125ms later than the time when the pressure increases in the left chamber of the hydraulic pilot directional valve 14. After the pilot valve 2 switches, the main hydraulic directional valve 3 begins to switch. Then the main hydraulic directional valve 3 switches, the pumping cylinders reverse (At the time point C).

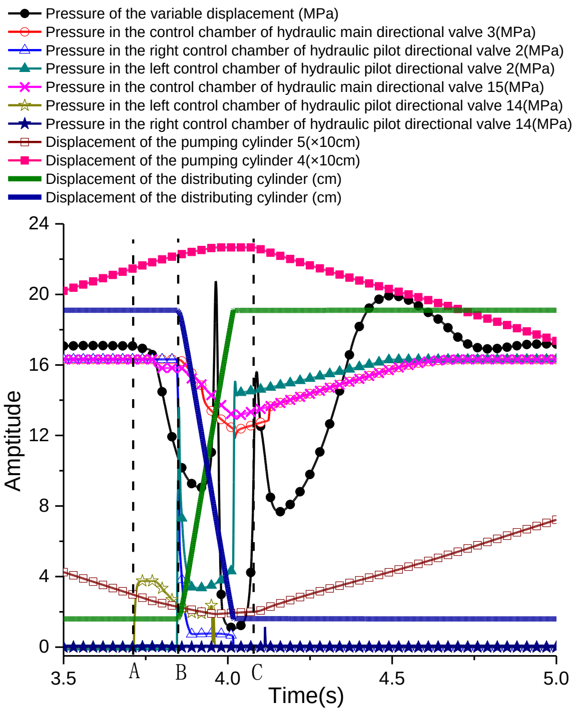

Fig. 4 The pumping pressure , the pressure in control chambers of the valves and displacement of cylinders

According to the above analysis, we can see that the valves in pumping circuit switch much later than the valves in the distributing circuit. The switching time for the whole hydraulic system is constant if the pressures of the constant pressure pump and all the control chambers of the valves do not change. Thus, with the different flowrate, the pumping speed and the pumping cylinder stroke are varied. The system switching time sequence can be optimized through adjusting each valve's switching time.

Generally, the control chambers of the hydraulic control directional valves are equipped with restrictors. Changing the restrictor diameters can adjust the control pressure. Thus, the switching time of the valve can be regulated.

3. Influence of the restrictors on the pressure shock and cylinder stroke

Typically, the restrictor is a short orifice(0.5< l/d≤4) or a thin-wall orifice. The flow rate can be described as follows,

$$Q = C_q \cdot A \cdot \sqrt{\frac{2}{\rho}\Delta p} \tag{4}$$

Define the pressure before the restrictor as p_1, the pressure after the restrictor as p_2, the diameter of the restrictor as d. Thus, the Equation (4) can be written as

$$p_2 = p_1 - \frac{8\rho Q^2}{\pi^2 \alpha^2} \cdot \frac{1}{d^4} \tag{5}$$

Where p_2 is the pressure in the control chamber of the valves.

Then the simulation model is utilized to simulate the pressure shock and the pumping cylinder stroke in different restrictors. The simulation condition sets as the typical actual working condition, which is the working pressure of 12MPa and the flowrate of 300L/min. Restrictor diameters in five positions which are located in the control chamber of the four directional valves (valve 2,3,14,15 shown in Fig. 1) and the U-tube have been simulated. At the same time, some experiments are done for comparison. The actual pressure of pumping concrete is 10~14MPa and the actual flowrate is 290L/min.

1) Restrictor in the control chamber of main hydraulic directional valve 3

It can be indicated from Figure 5Fig. 5 that the simulation results have the same trend with the experimental ones.

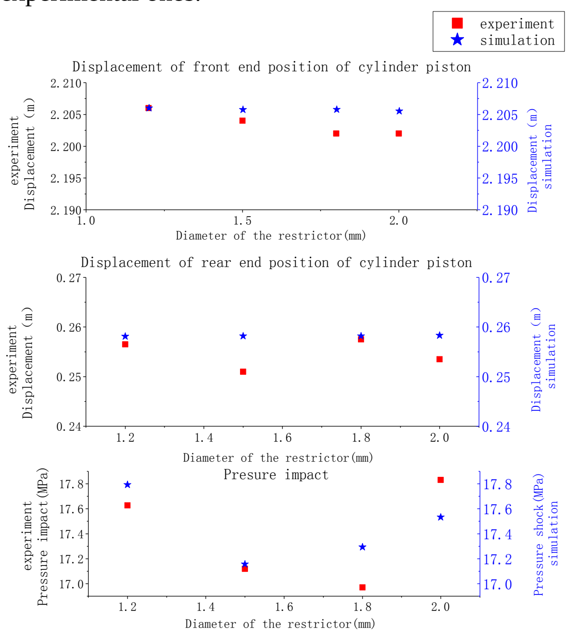

Fig. 5 Piston displacement of the cylinder and the pressure shock with different restrictor diameters used in valve 3

Fig. 5 shows the change of the piston

displacement of the cylinder and the pressure shock with the change of the restrictor diameters. With the restrictor diameter increasing from 1.2mm to 2.0mm, the piston displacement of the front end position of the cylinder decreases in a little range. As for the displacement of the rear end position, it is hardly changed. Generally, the restrictor here has little effect on the cylinder stroke.

The pressure shock decreases by 0.6Mpa approximately when the restrictor diameter changing from 1.2mm to 1.5mm, and in the diameter of 1.8mm, it has little change, then it increases in the diameter of 2.0mm.

According to the above analysis, the restrictor in this position has some effect on the pressure shock although it is not very obvious. Therefore, the diameter can be chosen from 1.2mm to 1.8mm to decrease the pressure shock.

2) Restrictor in the control chamber of hydraulic pilot directional valve 2

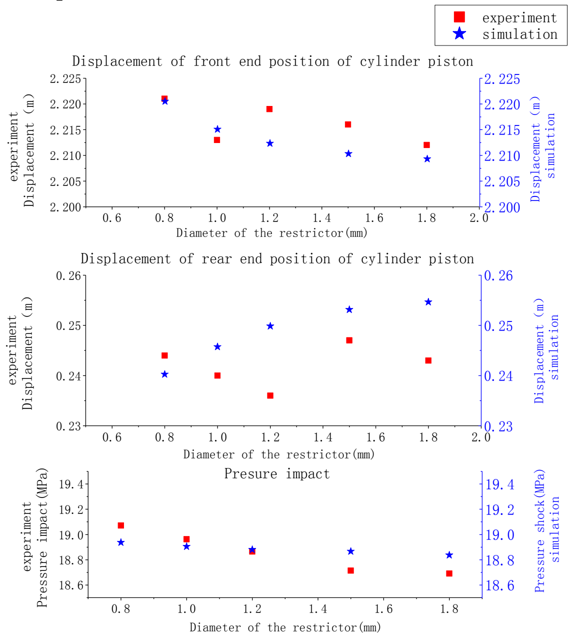

Fig. 6 Piston displacement of the cylinder and the pressure shock with different restrictor diameters used in valve 2

As indicated in Figure 6, with the restrictor diameter increasing from 0.8mm to 1.8mm, the piston displacement of the front end position of the cylinder decreases by 10mm and the piston displacement of the rear end position increases by 15mm approximately. Therefore the cylinder stroke decreases by approximately 25mm.

The pressure shock decreases gradually from 18.9MPa to 18.8MPa.

According to the above discussion, it is obvious that the restrictor diameter in this position has little effect on the pressure shock. The pressure only reduces 0.1MPa when the restrictor diameter changing from 0.8mm to 1.8mm. But it affects the cylinder stroke more, reducing the cylinder stroke 25mm. Taking a comprehensive consideration, the restrictor diameter here should be small, like 0.8mm.

3) Restrictor in the control chamber of main hydraulic directional valve 15

Seen from Figure 7, with the restrictor diameter increasing from 0.8mm to 2.0mm, the piston displacement of the front end position of the cylinder decreases from 2.26m to 2.21m and the piston displacement of the rear end position increases from 0.215m to 0.25m. Therefore, the cylinder stroke decreases 0.04m when the restrictor diameter increases from 0.8mm to 2.0mm.

The pressure shock increases from 18.5MPa to 21MPa when the diameter increases from 1.2mm to 2.0mm, although it decreases from 19MPa to 19.5MPa when the diameter is between 0.8mm and 1.2mm.

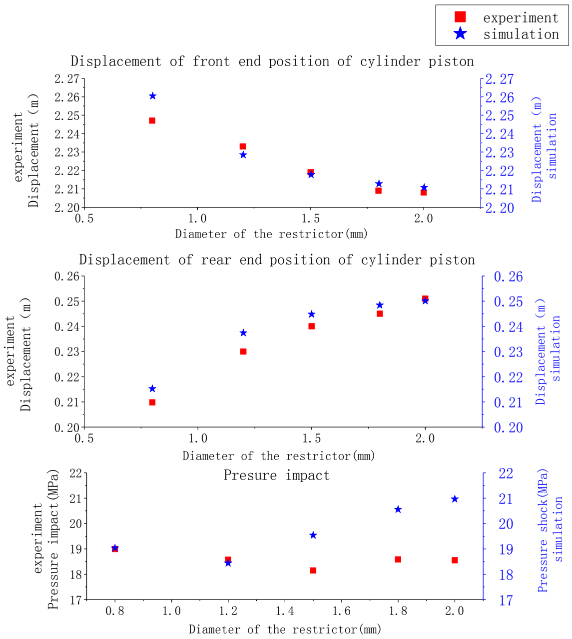

Fig. 7 Piston displacement of the cylinder and the pressure shock with different restrictor diameters used in valve 15

Therefore, the restrictor diameter here should be from 0.8mm to 1.2mm based on the above analysis. But actually, if the diameter is very small, the main hydraulic directional valve 15 switches very slowly. It will make the distributing cylinder reverse slowly and late. Then part of the concrete in the pumping pipeline will flow back and the pumping efficiency is reduced. The reversing time of the distributing cylinder should be less than 300ms [Ebinger Will, 1996], So it would be better not to choose the restrictor diameter too small here.

4) Restrictor in the control chamber of hydraulic pilot directional valve 14

Figure 8 shows that the piston displacement of the front end position of the cylinder decreases by

10mm when the restrictor diameter is changing form 0.8mm to 2.5mm and the displacement of the rear end position increases from 0.233m to 0.241m with the restrictor diameter varying from 0.8mm to 2.5mm.

The pressure shock increases from 14.5MPa at the diameter of 0.8mm to 16.5MPa at the diameter of 2.5mm.

The changing trends are almost the same as the restrictor in valve 15. Considering the pumping efficiency, the restrictor diameter in this position should be small, not too small. A range from 0.8mm to 1.2mm will be preferred.

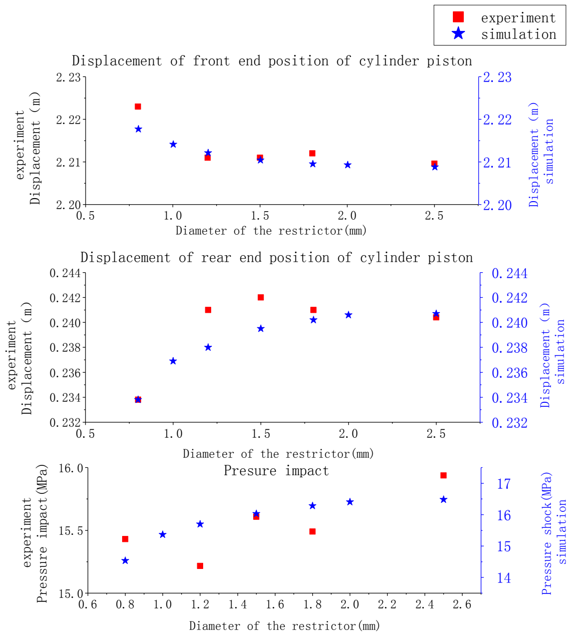

Fig. 8 Piston displacement of the cylinder and the pressure shock with different restrictor diameters used to valve 14

5) The U-tube restrictor 6,7,8 and 9

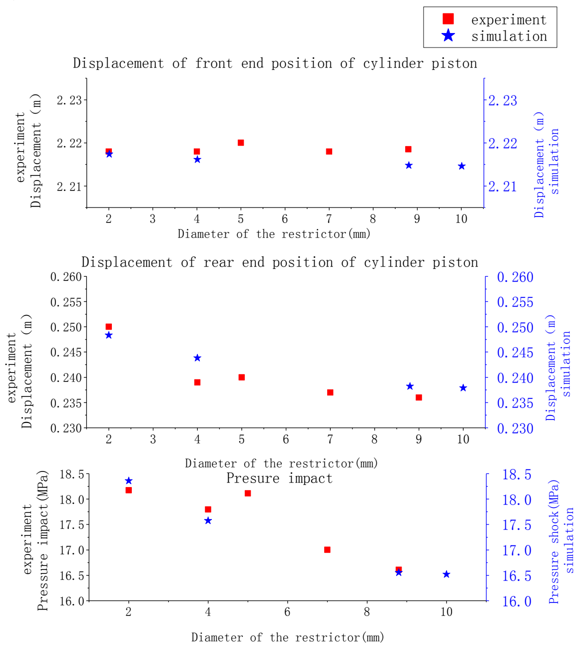

Fig. 9 Piston displacement of the cylinder and the pressure shock with different restrictor diameters used to U-tube

Seen from Figure 9, the piston displacement of the front end position of the cylinder has a slight change when the diameter changes and the displacement of the rear end position decreases more and more slowly from 0.248m to 0.238m with the restrictor diameter varying from 2mm to 10mm. The cylinder stroke decreases by 10mm approximately when the restrictor diameter increases.

The pressure shock also decreases from 18.25MPa to 16.5MPa when the restrictor diameter increases from 2mm to 10mm.

The restrictor diameter in this position should be chosen as large as possible. But the diameter is restricted by the structure of the cylinder. The maximum diameter is 8.8mm.

Overall, when the restrictor diameter used in valve 14 and 15 of the distributing circuit decreases, the switch processes of the two valves are late and slow, and the system has lower pressure shock and longer cylinder stroke. It has been proved that the switching time in the distributing circuit is much earlier than the time in the pumping circuit.

OPTIMIZATION COLLOCATIONS OF RESTRICTOR DIAMETERS AND EXPERIMENT VALIDATION

On the basis of the analysis of the simulation results, several kinds of restrictor diameter collocations which are shown in Table 1 are chosen to seek for and validate the proper one. The first collocation is the original one.

Table 1 restrictor diameter collocations

No.	1	2	3	4	5	6	7
V3	1.2	1.2	1.2	1.2	1.2	1.2	1.3
V2	0.8	1.2	2	0.8	0.8	0.8	1.2
V15	1.5	1.5	1.5	1.5	1	1.5	1.5
V14	0.8	0.8	0.8	1.2	0.8	0.8	0.8
U-tube restrictor 6,9	8.8	8.8	8.8	8.8	8.8	8.8	8.8
U-tube restrictor 7,8	8.8	8.8	8.8	8.8	8.8	4	8.8

Where: Vi——Control chamber of valve i, i denotes 3,2,15,14

The experimental device is shown in Figure 10. It mainly contains pump source, pumping cylinder, concrete mixer and concrete pumping pipeline.

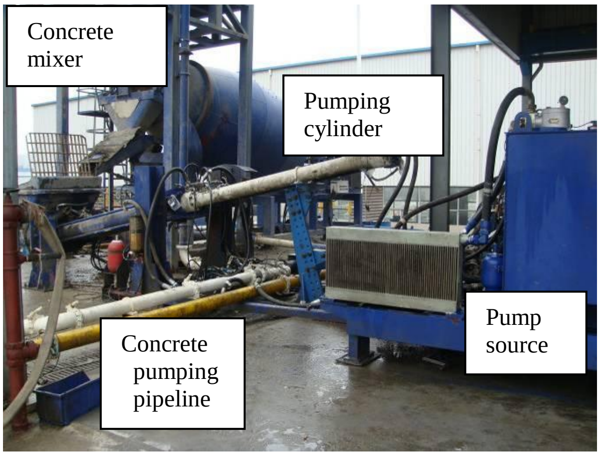

Fig. 10 experimental pumping facilities

The flowrate used in the experiment is 150 L/min、 200 L/min、 270 L/min、 300L/min and 380L/min, respectively.

1. Stroke of the pumping cylinders

The strokes of the pumping cylinders in the different flowrate are shown in Figure 11. It can be seen that the No.5, and No.2 collocations of the restrictor diameters are better than others, which are all beyond 1.9m. And No.5 collocation has the longest stroke of the pumping cylinders.

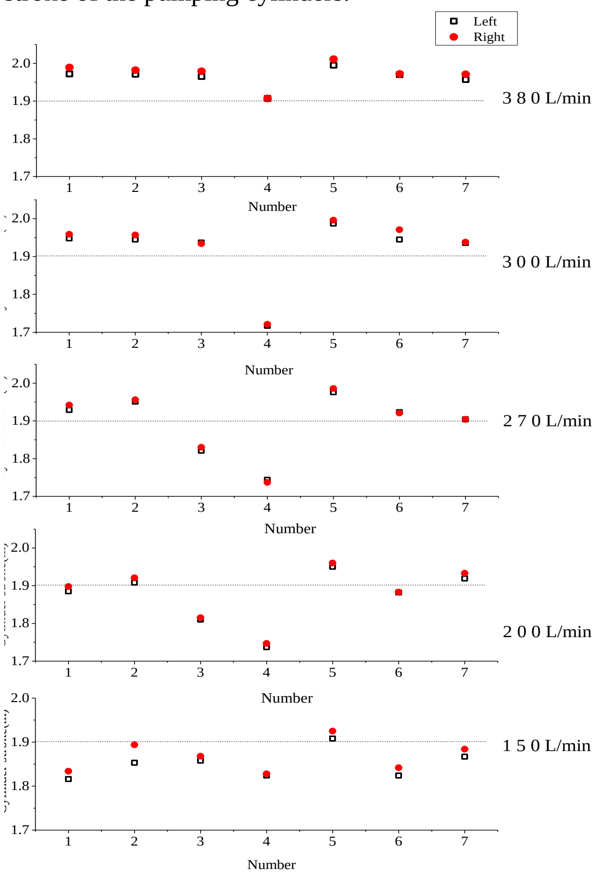

Fig. 11 Pumping cylinder strokes of the different collocations in the different flowrate

2. Pressure shock

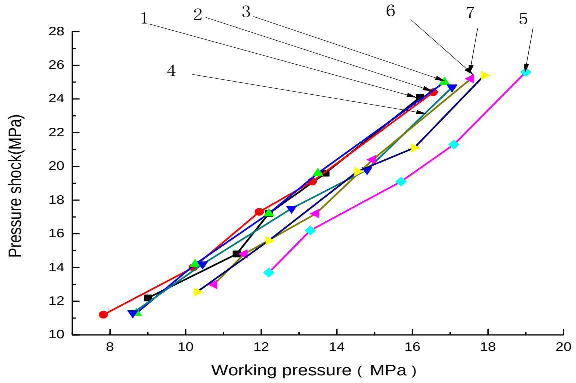

Fig. 12 Pressure shock in different working pressure

The pressure shocks in the different working pressure are shown in Figure 12. It indicates that the No.5 collocations of the restrictor diameters are better than others.

Therefore, the experiment result indicates that the No.5 collocation of the restrictor diameter is the best one. Compared with the original collocation(No.1), the cylinder stroke is lengthened at least 0.022m even 0.091m in the flowrate of 150L/min. The pressure shock reduces more than 20%.

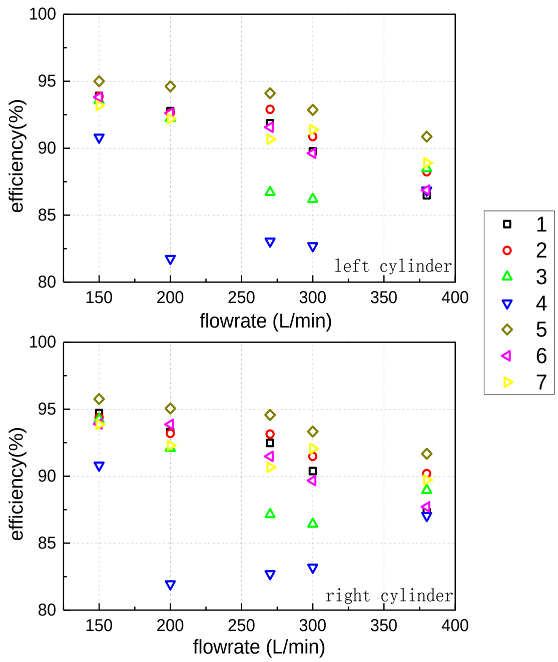

Fig. 13 Pumping efficiency

The pumping efficiency can be seen from Figure 13. The No.5 collocation of the restrictor diameter is the most efficient which is 1.1~4 percentage points more than the original collocation(No.1).

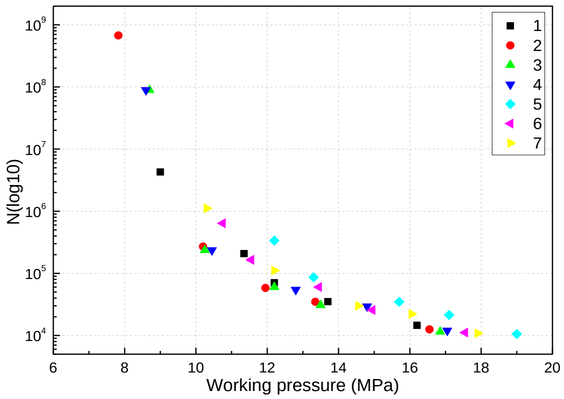

Fig. 14 Lifespan in different working pressure

The lifespan in different working pressure can be seen from Figure 14. In the same working pressure, the lifespan of the No.5 collocation is almost twice as many as the original collocation(No.1).

CONCLUSIONS

This research establishes the simulation model of the open circuit concrete pump hydraulic system with the software AMESim. Through the simulation of the switching time sequence, the reason that gives rise to the pressure shock and the deviation of the cylinder stroke is that the switching time in the pumping circuit is much later than it in the distributing circuit. Then the influence of the different restrictor diameters of the five positions on the pressure shock and the cylinder stroke are simulated. Some collocations of the restrictor diameters are chosen based on the simulation results. After the experimental comparison and validation, an appropriate collocation of the restrictor diameter is ascertained, which can lengthen the cylinder stroke 0.022m to 0.091m and reduce the pressure shock by 20%, so the pumping efficiency and the lifespan improved.

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NOMENCLATURE

A ——cross sectional area

C_q ——flow coefficient

E ——Young's modulus

e ——the pumping efficiency of one cycle

K ——effective stress concentration factor

L ——the nominal cylinder stroke

l ——the actual cylinder stroke

N ——the number of cycles

n ——safety factor

Δp ——differential pressure of the restrictor

Q ——flowrate

β ——surface machining factor

ε_d ——dimensional factor

ρ ——density

σ_a ——virtual stress amplitude(the conversion of full strain)

σ_F ——stress amplitude corresponding to elastic strain

ψ ——reduction of area

混凝土泵送開式液壓系統壓力衝擊與油缸行程偏差分析與優化

謝海波，陳健，劉豐，劉志斌，楊華勇，林樹勇，張勁

流體動力與機電系統國家重點實驗室

中聯重工科技發展股份有限公司

摘 要

本文針對混凝土開式泵送液壓系統中油缸行程偏差大和壓力衝擊大的問題進行了研究。建立了泵送系統的 AMESim 仿真模型，找出問題的原因是泵送與分配回路換向時間不匹配。根據各位置阻尼影響油缸行程偏差和壓力衝擊的變化規律，得出合理的阻尼配置并通過實驗驗證，使壓力衝擊與油缸行程偏差改善了 20%以上，提高了泵送效率和使用壽命。