International Journal of Fluid Machinery and Systems
Vol. 9, No. 1, January-March 2016

DOI: 10.5293/IJFMS.2016.9.1.028
ISSN (Online): 1882-9554

Original Paper

Robust Design of Main Control Valve for Hydraulic Pile Hammer Flexible Control System

Guo Yong[1], Hu Jun Ping[2], Zhang Long Yan[1]

[1]Hunan Provincial Key Laboratory of Health Maintenance for Mechanical Equipment, Hunan university of science and technology, Xiangtan,411201, P.R.China, hnkjdx_guoy@163.com, eaglexmn@163.com
[2]Mechanical Engineering Department, Central South University, Changsha, 410083, hujunpingok@sina.com

Abstract

The flexible control system for hydraulic pile hammer using main control valve is present to the requirement of rapidly reversing with high frequency. To ensure the working reliability of hydraulic pile hammer, the reversing performance of the main control valve should commutate robustness to various interfere factors. Through simulation model built in Simulink/Stateflow and experiment, the effects of relative parameters to reverse performance of main control are analyzed and the main interfere factors for reversing performance are acquired. Treating reverse required time as design objects, some structure parameters as control factors, control pressure, input flow and gaps between spool and valve body as interfere factors, the robust design of the main control valve is done. The combination of factors with the strongest anti-jamming capability is acquired which ensured the reliability and anti-jamming capability of the main control valve. It also provides guidance on design and application of the main control valve used in large flow control with interferes.

Keywords: hydraulic pile hammer, main control valve, reversing performance; pile driving, Stateflow, robust design

1. Introduction

With the advantages of high adjusted blow energy and energy transfer efficiency, low noise and pollution **[1]**, hydraulic pile hammer has been an environmentally friendly substitute for diesel hammer in pile foundation more and more abroad. As a key part of hydraulic pile hammer, the control system is also widely studied. Patrick **[2]** presented the method to control blow energy and efficiency through electrically operated directional valve. James **[3]** adopted pure hydraulic control system realized underwater pile driving, however, the blow energy and efficiency can't be adjusted. Iskander **[4]** described the control and design of single acting electro-hydraulic pile hammer. Lu **[5]** presented the pile hammer with hydro-pneumatic control, but the blow energy can only be adjusted by steps of stroke. Working at high frequencies requires hydraulic pile hammer to reverse smoothly and rapidly **[6-7]**. Control methods using stroke switch or untouched switch make the impact of reversing very high, which lead to the high failure rate and the low reliability of the system **[8-11]**.Flexible hydraulic control system overcomes the rigid impacts of solenoid valves reversing, which makes the reversing more smooth and rapid. It also makes blow energy and frequency adjusted stepless, which increase the adaptability of the equipment.

The function of main control valve as the key part is to realize the pilot control of hydraulic pile hammer avoiding rigid impact in reversing. To ensure hydraulic pile hammer work reliability, the reverse performance of the main control valve should have robustness to various interfere factors **[12-13]**. However, designers can't decide which factors make the performance more robustness, and the design parameters are typically chosen freely. Hence, it is important to consider the effects of parameters to main control valve's reversing performance using a method of robust design.

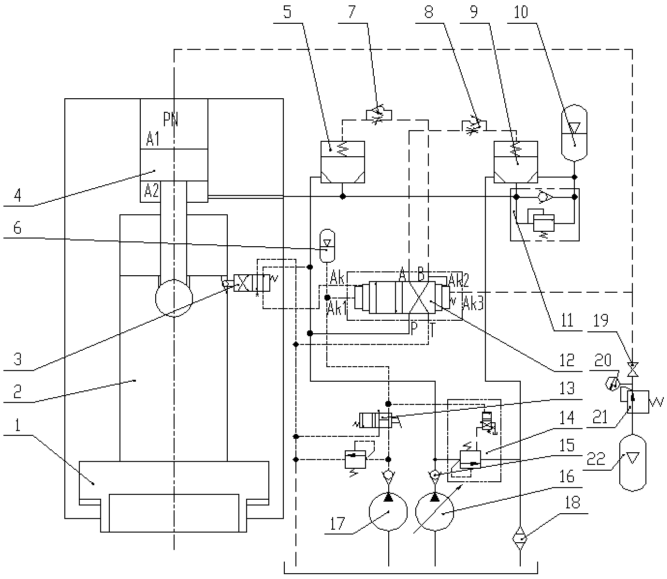
In the paper, the flexible control system for hydraulic pile hammer using the main control valve is present which ensure hydraulic pile hammer work reliability with rapidly reversing. According to the working principle of main control valve, the dynamic mathematical model of the main control valve reversing is established, and the simulation model using Simulink/Stateflow is presented. Experiments are conducted to verify the simulation model. The effects of relative parameters to reversing performance of main control are analyzed. Under the analysis of main factors, treating reverse required time as design objects, inner chamber diameter, feedback hole diameter and length as control factors, control pressure, input flow and gaps between spool and valve body as interfere factors, the robust design of main control valve is done to find the combination of factors with the strongest anti-jamming capability. It provides guidance on design and application of the main control valve used in large flow control with interferes.

Received January 12 2015; revised April 27 2015; accepted for publication May 19 2015: Review conducted by Zhang Yang-jun. (Paper number O15001C)
Corresponding author: Guo Yong, hnkjdx_guoy@163.com

2. Structure and Working Principle of Main Control Valve

2.1 Working Principle

The scheme of flexible control system for hydraulic pile hammer is shown as **Fig.1**. In initial condition, the hammer 2 is on the lowest position, the stroke valve 3 is on the right, and main control valve 12 is also on the right under control of nitrogen pressure and spring force. When the manual control valve 13 reversed, the left area AK1 of main control valve spool has a force generated by control pressure p_k, the ring area AK between spool and valve body has a force generated by system pressure p_s, meanwhile, the right area AK3 of main control valve spool has a force generated by nitrogen pressure p_N which is equal to the pressure of cylinder's upper chamber, and the ring area AK2 between spool and valve body connects to the oil tank. Under these forces, the main control valve 12 changes to the left. The control chamber for logic valve 5 connects to the oil tank through the ports B and T of main control valve which makes the logic valve 5 opened. The control chamber for logic valve 9 connects to high pressure oil through the ports A and P of main control valve which makes the logic valve 9 closed. The throttle valve A and B makes logic valve 9 closed before logic valve 5 opened which avoids the error movement of hydraulic pile hammer. The hammer 2 moves up until touched stroke valve 3 which makes stroke valve reversed.



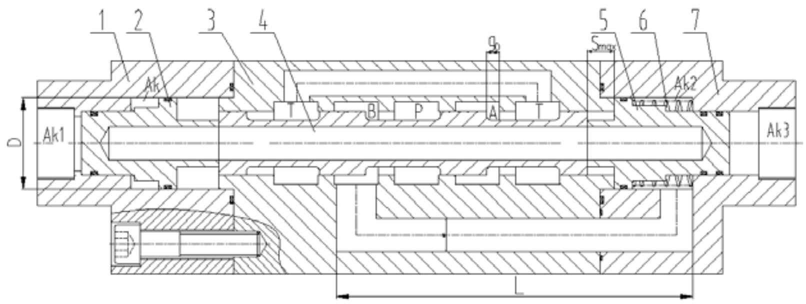
1.anvil 2.hammer 3. stroke valve 4.impact cylinder 5. logic valve of return 6. accumulator of control oil 7. throttle valve A 8.throttle valve B 9. logic valve of stroke 10. accumulator of return oil 11.reducing valve for one-way 12. main control valve 13.manual control valve 14. pressure adjust valve for pile driving 15.check valve 16. variable displacement pump 17.control pump 18.fileter of return oil 19. disconnecting valve 20. pressure gauge 21.relief valve for air 22. tank of nitrogen

Fig.1 Scheme of flexible control system for hydraulic pile hammer

At this time, the left ring area AK and the right ring area AK2 changes to connect with oil tank and the hammer 2 continually moves up to compress nitrogen in the upper chamber of impact cylinder 4, which leads the nitrogen pressure increased. When p_N is greater than p_k, main control valve 12 reverses to the right, the control chamber for logic valve 5 connects to high pressure oil through the ports A and P of main control valve which makes the logic valve 5 closed, and the control chamber for logic valve 9 connects to return oil through the ports B and T of main control valve which makes the logic valve 9 opened, which makes the lower chamber of cylinder 4 connects to the oil tank. The hammer 2 moves down under nitrogen pressure and gravity, and the nitrogen pressure p_N decreases. When the hammer 2 touched the stroke valve 3, the left ring area AK connects to high pressure oil which makes the main control valve reversed and a cycle of impact hammer movement finished. The application of reducing valve for one-way 11 realizes oil supply and protection for buffer chamber of impact cylinder 2.

2.2 Structure Description

The structure of main control valve is more complex than common four-way directional valve. It has two hydraulic control ports （AK and AK1）on the left side of spool, a air control port(AK3)and a port(AK2) connects to the feedback hole from port B on the right side of spool. The variation pressure on these four ports realizes the reversing of the main control valve as **Fig.2** shown. In order to manufacture the spool easily, the spool is divided to three segment contained left spool 2, spool 4 and right spool 5. The ring chamber between right valve support 7 and right spool 5 connects to the port B of valve body 3 through feedback hole, which makes the valve have the memory of the spool movement. Once the valve is on the left under the signal of stroke valve, it remains the state until another reversing signal transferred. In the initial condition, spools are on the left under the force of spring 6. The overlap displacement is S_{max}-g_0, which makes the reversing avoid interfere of feedback pressure with high frequency. When AK connects to high pressure oil and AK1 connects to control pressure oil, the spools move to the right. When the nitrogen pressure p_N on AK3 is higher than the control pressure p_k on AK1, the spools move to the left until the hammer reached the lowest position. The energy and frequency can be adjusted stepless through the adjustment of control pressure p_k.

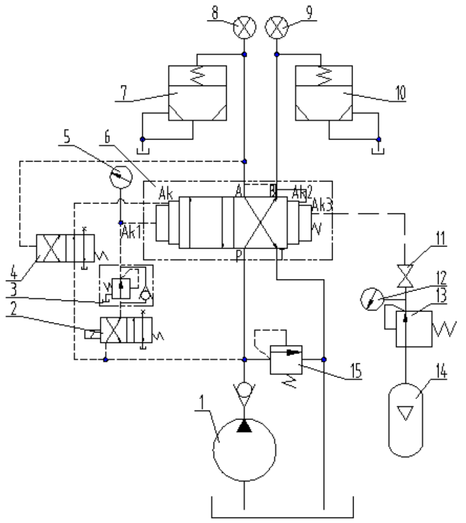


1.left valve support 2.left spool 3.valve body 4. spool 5.right spool 6.spring 7.right valve support

Fig.2 Structure of main control valve

3. Experiment for Reversing Performance of Main Control Valve

In order to test reversing performance in smooth and rapid for main control valve, the most popular way is to test the displacement of spool versus time. However, there are inconveniences of displacement sensor installation for both ends of the spool are connected with control port. Therefore, the test scheme of reversing performance by pressure collection based on flexible control system for pile hammer is designed as **Fig.3** shown, which uses pressure change of logic valve's controlling chamber to judge whether the reverse has been finished. Neglecting the influence of pipes, the time that the pressure of logic valve’s control chamber suddenly changed can be considered as the same time that main control valve completes reversing. That is to say, the time generated pressures suddenly change is equal to the time the main control valve reversed.



1.pump 2. solenoid valve 3. reducing valve with on direction 4.hydraulic control valve 5.12 pressure guage
6. main control valve 7.10 logic valve 8.9 pressure sensor 11. disconnecting valve 12. pressure guage
13.reducing valve for air 14. tank of nitrogen 15. relief valve

Fig.3 Experiment scheme for reversing performance of hydraulic pile hammer

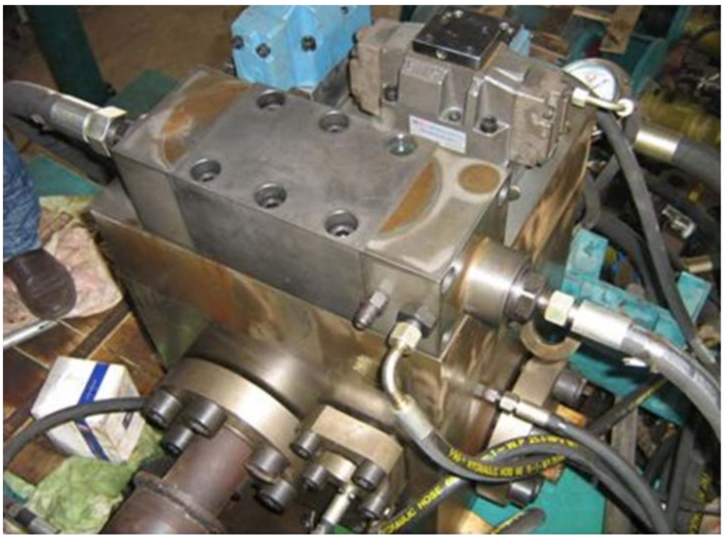


Fig.4 Picture of test

The test of the valve is shown as **Fig.4**. In parameters setting, regulate the pressure on the right area AK3 to 17 bar. It is realized by regulating pressure reducing valve 13 and disconnecting valve 11, and the pressure can be read on pressure gauge 12. Make system pressure equals to 210 bar by regulating relief valve 15. Energizing solenoid valve, regulating reducing valve with on direction 3 to ensure the pressure on left area AK1 is 22bar. In operation, the solenoid valve will be de-energized after energizing for 1s and remain the status. p_{L1} and p_{L2} tested by sensors 9 and 8 are the pressure of control chamber for logic valve 10 and 7, respectively. The pressures versus time in two seconds are shown as **Fig.5**.

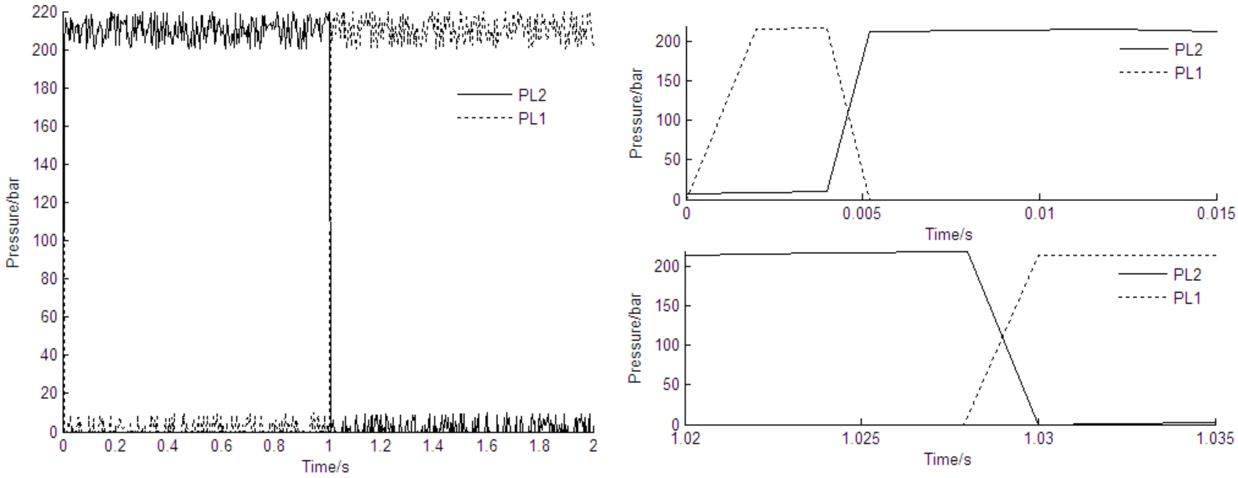


Fig.5 Test pressure of p_{L1} and p_{L2}

The test data indicate that when the main control valve moves rightward to the left side in 0-1s, p_{L1} reaches 210bar on 2ms and begin to decrease on 4ms which identifies the complement of left reversing movement. The pressure fluctuates around 0 bar after about 5ms. p_{L2} begins to increase at about 4ms and fluctuates around 210 bar after about 5ms.When the main controlling valve moves leftward to the right side in 1-2s, p_{L1} begins to increase at about 1.028s which identifies the complement of right reversing movement, then the pressure fluctuate around 210bar after about 1.03s. p_{L2} begins to decrease at about 1.028s and fluctuates around 0 bar after about 1.03s

4. Mathematical Model and Simulation of Main Control Valve Reversing

4.1 Mathematical Model

According to Newton's theory, the equation of motion for spool of main control valve is shown as follows

$$m\ddot{s} = P_{AK}A_1 + P_K A_0 - P_{AK2}A_1 - P_N A_0 - K(s+x_0) - f - F_s - F_t \quad (1)$$

$$f = \frac{\mu\pi d_{sp}L_f}{rc}\dot{s} \quad (2)$$

$$F_s = 0.43\pi\Delta p d_{sp}\left|s - g_0\right| \quad (3)$$

$$F_t = C_d\pi d_{sp}L_f\sqrt{2\rho\Delta p}\,\dot{s} \quad (4)$$

Where, m is the mass of the spool, s is the displacement of spool, P_K is the pressure on the left area AK, A_1 is the ring area, D is the inner chamber diameter of valve body, d_{sp} is the diameter of the spool, P_{AK2} is the pressure on the right ring area AK2, P_N is the pressure on the right area AK3, K is the spring stiffness, x_0 is the initial compression length of the spring, f is the friction between spool and valve body, F_s is the steady state flow force, F_t is the transient state flow force, μ is the dynamic viscosity of hydraulic oil, L_f is the length of spool shoulder, r_c is the gap between spool and valve body, C_d is the flow coefficient, ρ is the density of hydraulic oil, g_0 is underlap displacement.

Neglecting the effect of pipe, according to continuity equation on oil supply port of chamber AK, we get

$$Q_{AK} = \frac{1}{4}C_{d1}\pi d_{L1}{}^2\sqrt{\frac{2}{\rho}\left|P_{k1} - P_{AK}\right|} \quad (5)$$

Where, $P_{k1} = \begin{cases} P_s & g_0 > s \ge 0\, and \quad \dot{s} \ge 0 \\ & 0 \end{cases}$, P_S is the system pressure, d_{L1} is the diameter of oil-supply port.

$$Q_{AK} = -\mathrm{sgn}1(s)\frac{V_1}{E}\dot{P}_{AK} + \mathrm{sgn}2(\dot{s})A_1\dot{s} \quad (6)$$

Where, $\operatorname{sgn}1(s)=\begin{cases}1 & g_0>s\geq 0\\ -1 & others\end{cases}$, $\operatorname{sgn}2(\dot{s})=\begin{cases}1 & \dot{s}\geq 0\\ -1 & others\end{cases}$, V_1 is the volume of chamber Ak, E is the elastic modulus.

According to continuity equation on oil-supply port of chamber Ak2, we get

$$Q_{AK2}=\frac{1}{4}C_{d2}\pi d_{r1}^{2}\sqrt{\frac{2}{\rho}\left|P_{AK2}-P_{k2}\right|} \tag{7}$$

Where, $P_{k2}=\begin{cases}P_s & g_0>s\geq 0\\ & 0\end{cases}$

$$Q_{AK2}=\operatorname{sgn}1(s)\frac{V_1+V_p}{E}\dot{P}_{AK2}+\operatorname{sgn}2(\dot{s})A_1\dot{s} \tag{8}$$

Where, d_{r1} is the diameter of feedback hole, $V_p=\frac{\pi d_{r1}^{2}}{4}L$, L is the length of the chamber.

Using continuity equation on control chamber of the logic valve 7, we get

$$Q_A=C_d\left|\omega\right|\sqrt{\frac{2}{\rho}\left|P_A-P_{L1}\right|} \tag{9}$$

Where, $w=\pi d_{sp}(s-g_0)$, P_A is the pressure on port A of the main controlling valve, P_{L1} is the pressure on control chamber of logic valve 7.

$$P_A=\begin{cases}P_s & g_0\leq s\leq s_{\max}\\ 0 & \end{cases},\quad Q_A=-\operatorname{sgn}1(s)\frac{V_{L1}}{E}\dot{P}_{L1} \tag{10}$$

Where, V_{L1} is the volume of control chamber of logic valve 7.

Using continuity equation on control chamber of the logic valve 10, we get

$$Q_B=C_d\left|\omega\right|\sqrt{\frac{2}{\rho}\left|P_{AK2}-P_{L2}\right|} \tag{11}$$

$$Q_B=\operatorname{sgn}1(s)\frac{V_{L2}}{E}\dot{P}_{L2} \tag{12}$$

Where, V_{L2} is the volume of control chamber of logic valve 9.

4.2 Simulation Model

There are four motion states in a movement cycle of the main control valve, which is transformed in the certain condition. The key trouble to realize the simulation of main control reversing in Simulink is to solve the states transform. In order to solve the problem, Stateflow is used to embed with Simulink, which makes states control easily realized [14-15]. The chart of motion states transforms is established as **Fig.6** shown.

Combining the equation of motion for a spool of the main control valve, a simulation model is established in Simulink as **Fig.7** shown.

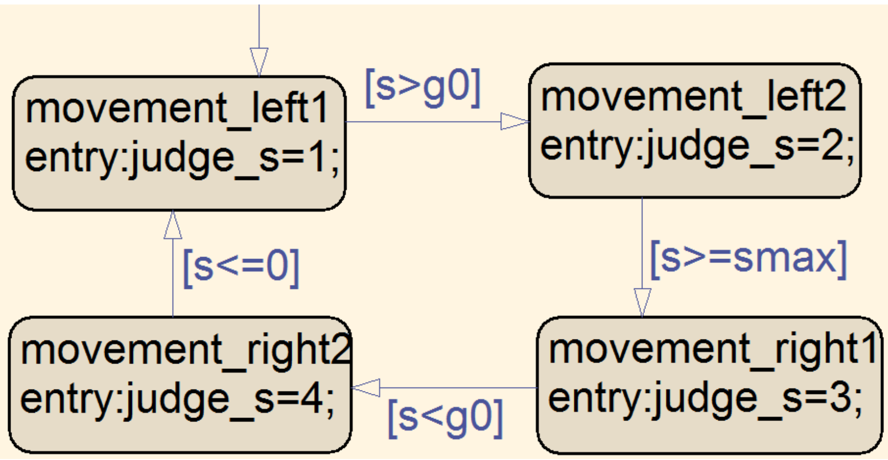


Fig.6 States transfer for spool movement of main control valve

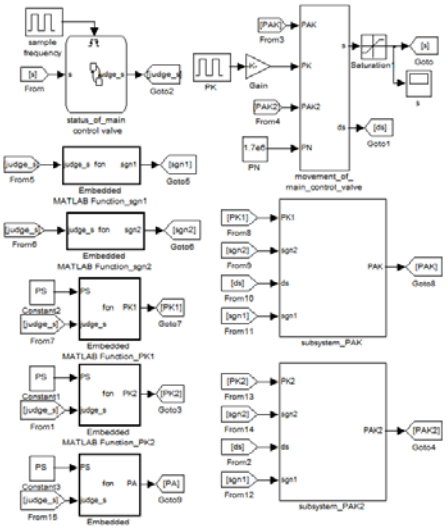


Fig.7 Simulation model for main control valve reversing

4.3 Simulation Results

Use rectangle wave to simulate control pressure change on the left area Ak1. The simulation parameters are given in **Table 1**. Using ode15s(stiff/NDF) algorithm to solve, the curves of p_{L1} and p_{L21} to time are obtained as **Fig.8** shown.

Table 1 Simulation parameters

Parameter	Value	Parameter	Value
density ρ/[Kg/m^3]	850	spool diameter d_{spool}/mm	35
underlap displacement g_0/mm	7	system pressure P_s/bar	210
elastic modus of oil E/bar	17000	volume of ring chamber V_1/cm^3	15
nitrogen pressure P_N/bar	17	Spring stiffness K/[KN/m]	9
dynamic viscosity μ/$Pa \cdot s$	0.0391	feedback oil length L/mm	80
feedback oil diameter d_{r1}/mm	16	gap between spool and valve body r_c/mm	0.036
spool mass m/Kg	3.53	Maximum displacement S_{max}/mm	15
control chamber volume of logic valves V_1, V_2/cm^3	38.4		

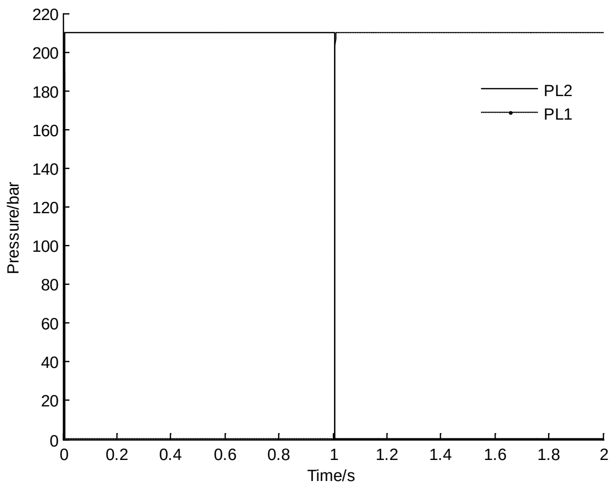


Fig 8 Simulation curve of p_{L1} and p_{L2} versus time

As indicated in the simulation data of **Fig.8**, when the main control valve moves rightward to left side in 0- 1s, p_{L1} reaches 210 bar at 2.1ms, then it begins to decrease at 4.5ms and keep stable on 210 bar after 5.4ms. p_{L2} begins to increase at 3.9ms and keep stable on 210 bar after 5.2ms. When the main control valve moves leftward to right side in 1-2s, p_{L1} begins to increase at 1.0042ms and keep stable on 210 bar after 1.0586s. p_{L2} begins to decrease at 1.0036ms and keep stable on 0 bar after 1.054s. The simulation results are accord with the testing results in 0-1s. However, the reversing time has almost 24ms difference between simulation and test results in 1-2s. There is a delay for about 20-30ms of solenoid valve movement **[16].** Neglecting the delay of solenoid valve, the reversing of main control valve can be simulated by the model.

5. Effect Factors on Reversing Performance

In order to enable the main control valve realize the rapidly control of the flexible impact system, a maximum flow rate of 400-600L/min is demand. The maximum flow rate is determined by the spool diameter d_{sp}, the underlap displacement g_0 and the maximum displacement S_{max}. In order to supply enough flow for rapid reverse of flexible control system, these factors which influence the reversing performance of the main control valve are ignored. By simulating the right movement of main control valve, the effect on rapidly reversing of inner chamber D, spring stiffness K, feedback hole diameter d_{r1} and feedback hole length L are analyzed through the variation of p_{L1}.

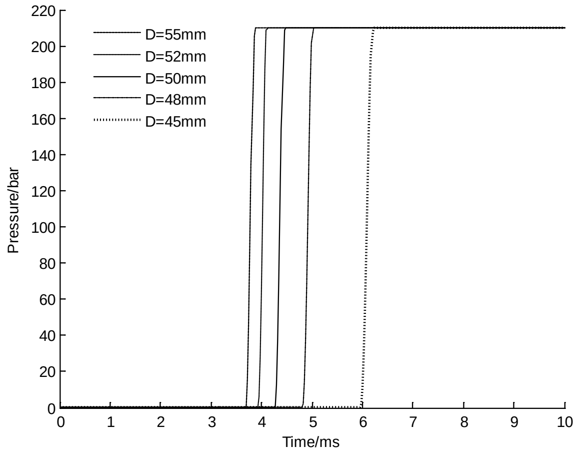


Fig.9 Effect of inner chamber diameter D

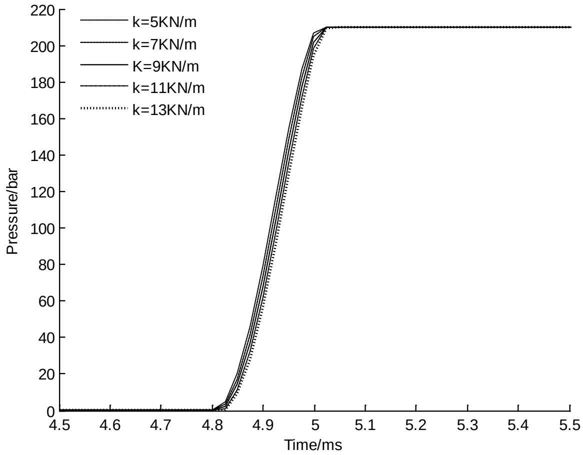


Fig.10 Effect of spring stiffness K

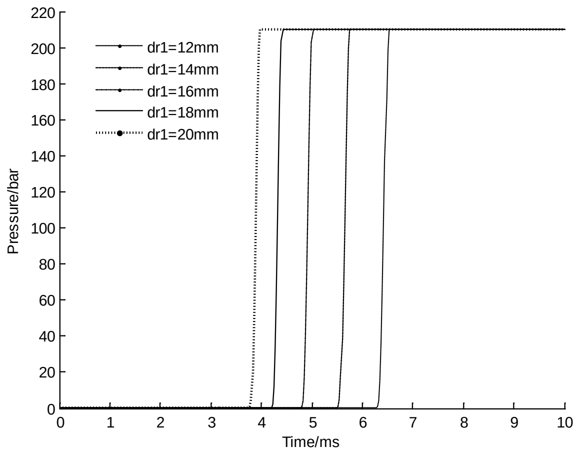


Fig .11 Effect of feedback hole diameter d_{r1}

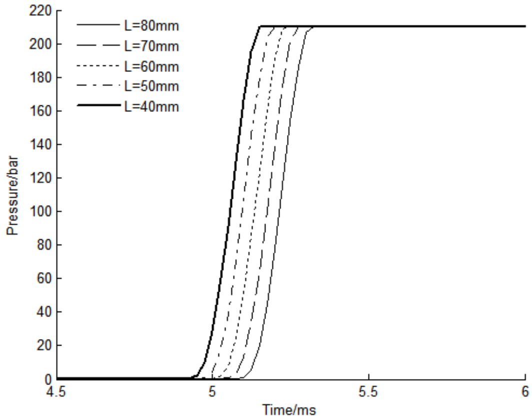


Fig.12 Effect of feedback hole length L

Fig.9 shows the effect of inner chamber diameter D on main control valve reversing. With the increase of D, the change of pressure p_{L1} is brought forward and the time used for reversing reduces. However, the decrement declines with the increase of diameter D. The diameter D has a great effect on main control reversing. There is a reduction of 0.2ms when the value of D changed from 52mm to 55mm.

Fig.10 shows the effect of spring stiffness K on main control valve reversing. With the decrease of K, the change of pressure p_{L1} is brought forward and the time used for reversing reduces. However, the effect on main control reversing is small.

Fig.11 and **Fig.12** shows the affects of feedback hole diameter d_{r1} and length L on main control valve reversing, respectively. According to the figures, d_{r1} and L effects the reserving of main control valve apparently. The time used for reversing decreases with the increase of d_{r1} and the decrease of L.

6. Robust Design

In above analysis, the inner diameter D, the feedback hole diameter d_{r1} and the length L are the main factors affect main control valve reversing. Therefore, the set of control factors is

$$x = (x_1, x_2, x_3) = (D, d_{r1}, L) \tag{13}$$

During the regulation of blow energy and frequency, the pressure P_K on left area Ak1 and the supply oil flow Q are changed frequently, and they also affects the reversing of main control valve that can be regarded as interfere factors. In addition, the gap r_c between spool and valve body also changes frequently due to the manufacturing tolerance. It is also a interference factor for the valve reversing. Therefore, the set of noise factors is

$$Z = (Z_1, Z_2, Z_3) = (P_K, Q, r_c) \tag{14}$$

The control factors adopt three levels, as shown in **Table 2**. The factors are tested according to arrange of $L_9(3^4)$ orthogonal table, which is showed in **Table 3**. Noise factors adopt three levels, as shown in **Table 4**. Adopting $L_9(3^4)$ orthogonal table to conduct outer table design, as shown in **Table 5**, combining the test scheme of outer and inner table, conducting 81 simulation test, recording the reversing time t_{ij} of each test(j is the times of test conducted as inner table , i is the times of test conducted as outer table).

Table 2 Levels of the control factors

Level	Factor		
	D/mm	d_{r1}/mm	L/mm
1	50	16	50
2	52	18	60
3	55	20	70

Table 3 Inner table and values of signal to noise ratio SN

Number	A(D)	B(d_{r1})	C(L)	e	SN/dB
1	1	1	1	1	19.158
2	1	2	2	2	19.136
3	1	3	3	3	19.9
4	2	1	2	3	19.72
5	2	2	3	1	20.1
6	2	3	1	2	20.63

7	3	1	3	2	20.54
8	3	2	1	3	20.63
9	3	3	2	1	20.54
T1	58.194	59.396	60.396	59.776	
T2	60.45	59.888	59.418	60.328	T=3614.174
T3	61.71	61.07	60.54	60.25	C_T =180.354
S_γ	2.115488	0.493496	0.248456	0.059496	S_T =2.916936

Table 4 Levels of noise factors

Level	Factor		
	P_K/bar	Q(L/min)	r_c/mm
1	22	160	0.036
2	30	200	0.025
3	36	240	0.009

Table 5 Outer table

Number	P_K/bar	Q(L/min)	Δr_c/mm
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	3
6	2	3	1
7	3	1	3
8	3	2	1
9	3	3	2

In statistical analysis of t_{ij}**[17]**, we can get

$$u_i = \frac{\sum_{k=1}^{k=j} t_{ik}}{j} \tag{15}$$

Where, u_i is the average value of the reversing time in the ith test according to inner table.

$$\sigma_i^2 = \frac{\sum_{k=1}^{k=j} (t_{ik} - \bar{u}_i)^2}{j-1} \tag{16}$$

Where, σ_i is the variance of the reversing time in the ith test according to inner table.

Signal to Noise Ratio (SN) is given as follows

$$SN = 10\lg \frac{u_i^2}{\sigma^2} \tag{17}$$

$$T_{i\gamma} = \sum SN_{i\gamma} \tag{18}$$

Where, $T_{i\gamma}$ is the sum of SN for the factor γ under i level.

$$T = \sum SN \tag{19}$$

Where, T is the sum of SN.

$$S_{T\gamma} = \frac{\sum_{i=1}^{3} T_{i\gamma}^2}{3} - \frac{T^2}{9} \tag{20}$$

Where, $S_{T\gamma}$ is the quadratic sum of the *SN* for factor γ.

$$S_T = \sum S_{T\gamma} \tag{21}$$

Where, S_T is the total quadratic sum of the *SN* .

$$C_T = \frac{T^2}{9} \tag{22}$$

Where, C_T is a correction term.

Statistical analysis results are shown in **table 3**. And the variance analysis results are shown in **table 6**. Both statistical quantities of factor D and d_{r1} are greater than $F_4^2(0.05)$, which means they are significant factors. While the statistical quantities of L is less than $F_4^2(0.05)$. By comparison, the impact of factor D is most significant, factor d_{r1} takes the second place, factor L ranks the least. To reduce the influences of noise factors, the control factors combination generating less signal-noise ratio should be selected. Hence, scheme 2 has the best anti-interference to noise factors, the optimum parameter combination resulted from statistical analysis are D=50mm, d_{r1}=18mm, and L=60mm.

Table 6 Analysis of variance

Resource	$S_{T\gamma}$	f_γ	V_γ	F_γ	S_γ'	ρ_γ
A	2.115488	2	1.057744	35.55681	35.49731	73.81905
B	0.493496	2	0.246748	8.294608	8.235112	17.12547
C	0.248456	2	0.124228	4.176012	4.116516	8.560572
e	0.059496	4	0.029748		0.237984	0.494904
S_T	2.916936	8			48.08693	

7. Conclusion

1) Combining working characteristics of hydraulic pile hammer, the working principal and structure of the main control valve are analyzed, which realizes flexible control of the pile driving and stepless control of blow energy and frequency.

2）According to the mathematical model of the test scheme for main control valve's reversing performance, the simulation model is established using Simulink and Stateflow. As shown in the results of simulation and experiment, if the motion delay of the solenoid valve is ignored, the simulation results are quite well corresponded to the experiment results.

3）The time required for the main control valve's reversing decreases as the inner chamber diameter D and feedback hole diameter d_{r1} increases, and increases with the increase of spring stiffness K and feedback hole length L.

4）Considering the interferes of manufacturing error r_c, blow energy and frequency adjusting parameters of P_K and Q, the analysis of the main control valve has done with parameters in practice. The robust design has done to avoid the interference of the noise factors. As shown in the results, inner chamber diameter D has the most significant influence on main control valve's reversing, the feedback hole diameter d_{r1} takes the second place. The parameter combination with the strongest capability of anti-interfere is the scheme in which D=50mm, d_{r1}=18mm, and L=60mm.

Acknowledgments

The authors thank Hunan Changhe machinery CO. LTD and the Science and Technology Department of Hunan province for th e financial support of this work under Grant no. 2008JT1014

Nomenclature

C_T	correction term	C_d	flow coefficient
D	inner chamber diameter of valve body	dsp	diameter of the spool
d_{r1}	feedback hole diameter	E	elastic modulus of hydraulic oil
F_s	steady-state flow force	F_t	transient-state flow force
f	friction between spool and valve body	g_0	underlap displacement
K	spring stiffness	L	feedback hole length
L_f	length of spool shoulder	m	mass of the spool
P_{AK}	pressure on the left ring area AK	P_{AK2}	pressure on the right ring area AK2
P_N	pressure on the circle area AK3	P_A	pressure on port A
P_s	system pressure	r_c	gap between spool and valve body

s	displacement of spool	SN	signal to noise ratio
$S_{T\gamma}$	quadratic sum of the SN for factor γ	$T_{i\gamma}$	sum of SN for the factor γ under i level
T	sum of *SN*	V_1	volume of chamber Ak
x_0	initial compression length of the spring	μ	dynamic viscosity of hydraulic oil
u_i	average value of the reversing time in the *i*th test	σ_i	variance of the reversing time in the *i*th test

References

[1] Donald, L. B., Jams, W. W., and Kathleen, I. F., 1997, “Pile driving equipment,” Engineering and Construction Army Corps of Engineers, Washington, UFC 3-220-02P.
[2] Patrick, D. B., Mark, T., and David, H., 2008, US Patent No. 7404449B2.
[3] James, E. A., 2009, US Patent No. 20100012336A1.
[4] Iskander, M., 2011, Behavior of Pipe Piles in Sand, Berlin.
[5] Lu, J.Z., Sui, Z., Yang,Y. H., et al, 2004, “Hydro-pneumatic driving system of new type hydraulic pile hammer,” Journal of Jilin University of Technology(Natural Science Edition), Vol. 34, No. 4, pp. 597-601.
[6] Hu, J. P., Guo,Y., Song, G. W., et al. 2010, “Numerical analysis for impact model of pile hammer,” China Journal of Highway and Transport, Vol. 23, No. 2, pp. 122-126.
[7] Yang, Y. H., Wang, X. L., Lu, J.Z., et al, 2005, “Study on performance evaluation system and testing methods of hydraulic pile hammer,” Transactions of The Chinese Society of Agricultural Machinery, Vol. 36, No. 4, pp. 597-601.
[8] Hu, J. P., Shi, T. L., and Zhang, L., 2009, “Simulation of a new type hydro-pneumatic hydraulic pile hammer,” Computer Simulation, Vol. 26, No. 8, pp. 274-277.
[9] Yang G. P., 2006, “Research of a full hydraulic pressure hydraulic impactor with strike energy and frequency adjusted independently,” Journal of Hunan University of Science & Technology(Natural Science Edition),Vol. 21, No. 1, pp. 25-29.
[10] Angadi, S. V., Jackson, a., Song, Y. C, et al, 2009, “Reliability and life study of hydraulic solenoid valve,” Engineering Failure Analysis, Vol. 16, No. 1, pp. 874-887.
[11] Sumio, S., Keita, D., Masaaki, T., Gaku, H. J., Takuya, I., 2012, “Effects of the lift valve opening area on water hammer pump performance and flow behavior in the valve chamber,” International Journal of Fluid Machinery and Systems, Vol. 5, No.3, pp. 117-125.
[12] Li, P. Y., 2002, “Dynamic redesign of a flow control servo valve using a pressure control pilot,” Journal of Dynamic Systems Measurement and Control, Vol. 124, No. 1, pp.428-434.
[13] Om, P. Y., Nanua, S., Ratna, B. C., et al, 2003, “A fuzzy logic based approach to reliability improvement estimation during product development,” Reliability Engineering & System Safety, Vol. 80, No.1, pp. 63-74.
[14] Aditya, A., Gyula, S., and Gabor, K., 2004, “Semantic translation of Simulink/Stateflow models to hybrid automata using graph transformations,” Electronic Notes in Theoretical Computer Science, Vol. 109, No. 14, pp. 43-56.
[15] Jia, Q. L., Yun, D. L., and Luan, Y. F., 2006, System simulation and design based on MATLAB7.x\Simulink\Stateflow, Northwest industrial university press, Xian.
[16] Cheng, D. X., 2004, Handbook of mechanic design (volume4), Chemical industry press, Beijing.
[17] Wu, C.E., and Hamada, M., 2000, Experiments: Planning, analysis, and parameter design and optimization, John Wiley & Sons, New York.