Numerical Simulation of Transient Characteristics of Water Hammer before Proportional Directional Valve of Ship Steering System

Yankai Qiu

Huazhong Univ Sci & Technol,FESTO Pneumatic Center.1037 Luoyu Road, Wuhan, China. E-mail: qiuyankai@126.com

Baoren Li

Huazhong Univ Sci & Technol, FESTO Pneumatic Center. 1037 Luoyu Road, Wuhan, China. E-mail: lbr@hust.edu.cn

Gang Yang

Huazhong Univ Sci & Technol, FESTO Pneumatic Center. 1037 Luoyu Road, Wuhan, China. E-mail: ygxing_73@sohu.com

Abstract

With the development of hydraulic technology, hydraulic steering gears have been widely used on the ship. However, the fluid-borne noise of hydraulic steering gear system is one of its fatal flaws, which not only affects working and resting environment of the crew, but also lead to various accidents. Water hammer which is caused by opening or closing the proportional directional valve abruptly is a vital noise source in the ship steering system, so reducing the effect of water hammer is important to improve work environment and the performance of the ship steering system. The wave equations of non-constant flowing of the pipeline of the ship steering system were established in the paper. The numerical simulate of the transient characteristics of water hammer which was caused by closing the proportional directional valve abruptly was done by the method of characteristics and finite difference method ((MOC-FDM) using Fortran language. The simulation took into account not only the steady-state friction loss of the pipeline, but also the dynamic friction loss of the pipeline. The effects of reducing water hammer by using the single accumulator and multi-accumulator with different inflation pressures and nominal volumes were compared in the paper, too. The results show that the capacity to reduce the water hammer or hydraulic shock is related to the nature frequency and the ratio of damping of the system.

KEYWORDS: water hammer, ship steering system, numerical simulation, transient characteristics

1. Introduction

Water hammer (hydraulic shock or transient flow), is the term used to describe the pressure fluctuations in pipeline when the liquid is stared or stopped quickly. With the development of hydraulic technology, the mechanical steering gears is gradually replaced by the electro-hydraulic steering gears which include pump-control steering gears (PCSG) and valve-control steering gears (VCSG) in ships. The VCSG has been widely used in ships because of the low cost and simple compositions. However, water hammer caused by valve closure will reduce the servo precision of the system and generate great noise, which is a great challenge to the development of VCSG.

The calculation methods of water hammer mainly include analytical method, graphical method, numerical simulation method. Water hammer is a complex phenomenon with non-constant fluctuations. The cavitation caused by negative pressure will change the original characteristics of the pressure wave fundamentally. So it is difficult to accurately describe the process of water hammer by the former two methods which must greatly simplify the problem. With the emergence of high-speed computers and the development of numerical methods, a new age for water hammer analysis is dawning. Numerical methods of water hammer mainly include method of characteristics (MOC), finite volume method (FV), plane wave method. The equations of water hammer are essentially hyperbolic partial differential equations, which can be translated into ordinary differential equations by MOC, which make it easy to simulate in computes and is the most widely used numerical method for water hammer problems. It is necessary all the components of the system have the same time step which usually use grid interpolation methods (GIM) in the simulation by MOC (Cai 1990). GIM are mainly include spatial interpolation methods and time interpolation methods (Trikha 1975 & Goldberg etal.1983). Time interpolation methods are include forward time interpolation and backward time interpolation. The former can greatly reduce the time step without the limitations of time, while the latter has well efficient and has advantages in solving non-linear problems. In addition, some scholars have proposed mixed-interpolation method (Lai, 1989 & Karney et al. 1997). spline interpolation method (Sibertheros et al. 1991) and other new methods.

The inertia and viscous properties of journal fluid of the pipeline has an important impact on the transient characteristics. As the oscillation of pressure wave is closely related to the inertia properties of fluid, while the friction in the pipeline is mainly caused by the viscous properties of fluid. The friction model of pipeline mainly include steady-state friction model (Streeter *et al.* 1985) experienced friction model (Brunone *et al.* 1991) and friction model based on physical properties (Vardy et al. 1995).

Using accumulator to prevent pressure surge and valve vibration of the condensate pump system is a good method (Jiang, 2010). An accumulator of convenient size can reduce the duration and amplitude of the pressure fluctuations (Rabie, 2007). In the paper, the accurate model taking account to the characteristic of entrance of the accumulator is established. In addition, the models of single-accumulator and multi-accumulator suppressing the hydraulic shock are established respectively. The numerical simulation of single-accumulator, multi-accumulator which use MOC-FDM take account to both steady-state friction and unsteady-state friction.

2. Mathematical model

2.1 Model of single-accumulator taking account to the characteristics of entrance



Figure 1: Suppressing water hammer with single-accumulator



Figure 2: Structure of the bladder accumulator

The bladder accumulator, which mainly consist of bladder, shell, poppet valve and so on is widely used in suppressing the hydraulic shock as it is sensitive to the pressure fluctuations. In practice, the pipeline L_1 is usually used to connect the accumulator to the hydraulic system. The force balance equation of L_1 follows:

$$P_1 - P_{1m} = R_{1H}Q_{1m} + L_{1H}(\mathrm{d}Q_{1m}/dt)$$
⁽¹⁾

The force balance equation of L₂ follows:

$$P_{2m} - P_2 = R_{2H}Q_2 + L_{2H}(\mathrm{dQ}_2/\mathrm{d}t)$$
⁽²⁾

The force balance equation of the fluid in the accumulator follows:

$$P_2 - P_a = L_{acH} dQ_2 / dt + R_{acH} Q_2$$
(3)

Where P_i = the pressure of the accumulator, Q_i = the flow of the accumulator, $R_{iH} = \frac{128\mu l_i}{\pi d_i^4}$ = the fluid resistance of the accumulator, $L_{iH} = \frac{m_i}{A_i^2} = \frac{\rho l_i}{A_i}$ = the fluid induction of the accumulator, μ = oil viscosity, ρ = oil density, l_i = the length of the pipeline L_i , A_i is cross-sectional area of the pipeline L_i , m_i = quality of the of the oil of pipeline L_i , i=1,2,ac. In the joint 1m-2m of the pipeline, the force equation follows:

$$P_{1m} - P_{2m} = \Delta P_m = \xi \rho v_1^2 / 2$$
(4)

Where $\xi =$ local resistance coefficient, $v_1 =$ the flow velocity in the joint of the pipeline.Compared to the overall pressure drop, the pressure drop in the joint of the pipeline is very small, after ignoring the local resistance coefficient ξ , the equation (4) follows:

$$P_{1m} = P_{2m} \tag{5}$$

The compressibility of the oil is very small compared with air, which has little effect on the system. After ignoring it:

$$Q_1 = Q_{1m} = Q_{2m} = Q_2 \tag{6}$$

When the accumulator is considered as "gas spring-damper model", the force balance equation follows:

$$P_{a}A_{ac} = K_{e}\frac{V_{a}}{A_{ac}} + C_{a}\frac{dV_{a}}{dt}\frac{1}{A_{ac}}$$
⁽⁷⁾

Where K_e = stiffness of gas spring, C_a = gas damping coefficient.

In the equation (7), the second term is very small compared to the first term in the right side of equality sign, which could be ignored. Unite equations (1)-(7):

$$P_1 = \frac{m_e}{K_e} \ddot{P}_a + \frac{B_e}{K_e} \dot{P}_a + P_a \tag{8}$$

Equation (8) can be rewritten after Laplace transformation as follows:

$$\frac{P_a(s)}{P_1(s)} = \frac{1}{\frac{s^2}{w_n^2} + 2\zeta \frac{s}{w_n} + 1}$$
(9)
Where $w_n = \sqrt{\frac{k_e}{m_e}} =$ natural frequency of the accumulator, $\zeta = \frac{B_e}{2m_e w_n} =$ damping ratio
of the system, $m_e = m_{ac} + m_1 (\frac{A_{ac}}{A_1})^2 + m_2 (\frac{A_{ac}}{A_2})^2 =$ effective mass of the system,
 $B_e = 8\pi\mu (l_{ac} + l_1 (\frac{A_{ac}}{A_1})^2 + l_2 (\frac{A_{ac}}{A_2})^2) =$ equivalent damping coefficient of the system,
 $K_e = \frac{kP_0}{V_0} A_{ac}^2 =$ stiffness of the system, $k =$ air polytropic exponent.

2.2. Model of multi-accumulator suppressing water hammer



Figure 3: Suppressing water hammer with multi-accumulator

When there are lots of small accumulators before the proportional direction valve, the model of the system is following:

$$\frac{P'_{a}(s)}{P_{1}(s)} = \frac{1}{\frac{s^{2}}{w_{n}^{\prime 2}} + 2\zeta' \frac{s}{w_{n}^{\prime}} + 1}$$
(10)
Where $w'_{n} = \sqrt{\frac{k'_{e}}{m'_{e}}} =$ natural frequency of the accumulators, $\zeta' = \frac{B'_{e}}{2m'_{e}w'_{n}} =$ damping ratio
of the system, $m'_{e} = N(m_{ac} + m_{1}(\frac{A_{ac}}{A_{1}})^{2} + m_{2}(\frac{A_{ac}}{A_{2}})^{2}) =$ effective mass of the system with
multi-accumulator, $B'_{e} = 8\pi\mu(l_{ac} + l_{1}(\frac{A_{ac}}{A_{1}})^{2} + l_{2}(\frac{A_{ac}}{A_{2}})^{2}) =$ equivalent damping coefficient
of the system, $K'_{e} = N\frac{kP_{0}}{V_{0}}A^{2}_{ac} =$ stiffness of the system, N = the number of accumula-

tors.

3. Friction model

When the axial velocity of the one-dimensional unsteady flow before valve is replaced by the average velocity *V*, taking the small fluid body which the thickness is δx for studying, the momentum equation of the flow is following:

$$\frac{\partial P}{\partial x} + \frac{\rho Q}{A^2} \frac{\partial Q}{\partial x} + \frac{\rho}{A} \frac{\partial Q}{\partial t} + f(Q) = 0$$
(11)

Where P=pressure of the pipeline, Q= flow of the pipeline.

The continuity equation is following:

$$\frac{\partial P}{\partial t} + \frac{Q}{A}\frac{\partial P}{\partial x} + \frac{\rho a^2}{A}\frac{\partial Q}{\partial x} = 0$$
(12)

Where $a = \sqrt{\frac{\frac{K_e}{\rho}}{1 + \frac{d}{E}\frac{K_e}{B}(1 - \frac{\mu'}{2})}}$ = propagation speed of the pressure wave, A= cross sec-

tional area of the pipeline, f(Q) = friction related to the flow Q, d is the diameter of the pipeline, E = elastic modulus of the pipe, B = pipe thickness, μ' = poisson's ratio of the pipe.

The wave equations (11) and (12) are all hyperbolic partial differential equations, which can be transformed to special ordinary differential equations by MOC as follows:

$$C^{+}:\begin{cases} Z_{0} \frac{dQ}{dt} + \frac{dP}{dt} + af(Q) = 0\\ \frac{dx}{dt} = a \end{cases}$$

$$C^{-}:\begin{cases} Z_{0} \frac{dQ}{dt} - \frac{dP}{dt} + af(Q) = 0\\ \frac{dx}{dt} = -a \end{cases}$$
(13)

On the integral along the characteristic line, the ordinary differential equations can be transformed to finite difference equations which can be processed easily in the computer as follows:

$$Z_0(Q_P - Q_A) + P_P - P_A + \frac{a\Delta t}{2}(f(Q_P) + f(Q_A)) = 0$$
(15)

$$Z_0(Q_P - Q_B) - P_P + P_B + \frac{a\Delta t}{2}(f(Q_P) + f(Q_B)) = 0$$
(16)

The viscosity of hydraulic oil is large compared to the air and water, so it is necessary to consider both steady friction and unsteady friction in the dynamical flow:

$$f(Q(t)) = f_u(Q(t)) + f_s(Q(t))$$
(17)

Where $f_d(Q(t))$ is the unsteady friction as follows:

$$f_u(Q(t)) = \frac{16\rho\nu}{d^2} \sum_{i=1}^k \int_0^t m_i e^{-n_i(4\nu/d^2)(t-t_1)} \frac{\partial V(t_1)}{\partial t} dt_1$$
(18)

In the equation (18), $\frac{\partial V(t_1)}{\partial t}$ is an unknown expression, which is difficult to calculate directly by integral method. After $\frac{\partial V(t_1)}{\partial t}$ is replaced by the iterative expression $(Q(t + \Delta t) - Q(t))/A$, the equation (23) can be rewritten as follows:

$$f_d(Q(t)) = \frac{16\rho\nu}{d^2} \sum_{i=1}^k \int_0^t m_i e^{-n_i(4\nu/d^2)(t-t_1)} (Q(t+\Delta t) - Q(t)/A) dt_1$$
(19)

Where $m_i > n_i$ could be calculated by the function which was proposed by Trikha as follows:

$$W_{app}(\tau) = \sum_{i=1}^{3} m_i e^{-n_i \tau} = 40.0 e^{-8000\tau} + 8.1 e^{-200\tau} + e^{-26.4\tau}$$
(20)

 $f_s(Q(t))$ is the steady friction, which is as follows for laminar flow:

$$f_s(\mathcal{Q}(t)) = \frac{128\rho v}{\pi d^4} \mathcal{Q}(t) \tag{21}$$

For turbulent flow, $f_s(Q(t))$ is following:

$$f_s(Q(t)) = \frac{0.213\rho v^{0.75}}{d^{4.75}} (Q(t))^{1.75}$$
(22)

Where v is the kinematic viscosity of hydraulic oil.

4. Numerical simulations

4.1 With single-accumulator

When the proportional direction value is closed abruptly, the flow before the value Q_6 will have a step change. As $\Delta Q_6 = -\frac{\pi d}{4}v_0$, the equation can be rewritten after Laplace transformation as follows:

$$Q_6(s) = -\frac{\pi d^2}{4s} v_0$$
(23)

Where d = diameter of the pipe, v_0 = flow velocity before the valve.

In general, the performance indicators of transient response of the second-order system include peak time t_P maximum overshoot M_P , adjustment time t_s , which can be found in Hu, S. S. (2001) and Yang, S. Z. & Yang, K. C. (2001).

The time when the response curve reach maximum value is defined as peak time,

$$t_p = \frac{\pi}{w_d} = \frac{\pi}{w_n \sqrt{1 - \zeta^2}}$$
(24)

When $t \ge t_s$, $|P_o(t) - P_o(\infty)| \le \Delta \cdot P_o(\infty)$, in which t_s is defined as adjustment time and Δ is the ratio of allowable amplitude of pressure fluctuations to the pressure of system. When $\Delta = 0.02$, the expression of t_s follows:

$$t_s \approx \frac{4}{\zeta w_n} = \frac{8m_e}{B_e}$$
(25)

Maximum overshoot is defined as follows:

$$M_{p} = \frac{P_{o}(t_{p}) - P_{o}(\infty)}{P_{o}(\infty)}$$
(26)

United equations (24) and (26), the equation (14) can be rewritten as follows:

$$M_{P} = e^{-\zeta \pi / \sqrt{1 - \zeta^{2}}}$$
(27)



Figure 4: Trend of M_p with the change of ζ



Figure 5: with single-accumulator before the proportional direction valve

The accumulators with nominal volume 1.6L and 2.5L were used in the paper respectively, which have the same size except the total length. The length of connecting pipe is 66mm, the length of neck pipe is 34mm, the diameter of neck pipe is 42mm, the diameter of the accumulator body is 152mm.



Figure 6: Pressure fluctuations without accumulator

As the figure 6 shows, when the proportional direction valve is closed abruptly, the kinetic energy of the fluid is transformed to the pressure energy. The maximum pressure is 15.94 MPa, while the minimum pressure is 4.41 MPa. The period of the pressure fluctuations is 0.094s. After the time about 3 second, the pressure fluctuations amplitude gradually trend to steady state because of the oil viscosity and pipeline friction. The simulation is consistent with the theoretical results, which the maximum value of the pressure is 15.95 MPa, and minimum value is 4.05 MPa, and the period of the pressure wave is 0.094s.



Figure 7: Simulation results with single accumulator 1.6L/5MPa and 1.6L/6MPa suppressing water hammer



Figure 8: Simulation results with single accumulator 2.5L/7MPa and 2.5L/8MPa suppressing water hammer

Note: V_s / P_s = the nominal volume and the inflation pressure of the accumulator.

As is shown in the figure 7 and 8, the pressure with accumulator has reduced obviously by contrast to the results in the figure 7 without accumulator. The maximum pressure is 15.57 MPa, the minimum pressure is 5.57 MPa with accumulator 1.6L/5MPa, while the maximum pressure is 14.66 MPa, the minimum pressure is 6.39 MPa with accumulator 1.6L/6MPa. The maximum pressure is 13.34 MPa, the minimum pressure is 7.16 MPa with accumulator 2.5L/7MPa, while the maximum pressure is 8.09 MPa with accumulator 2.5L/8MPa.

With the increasing of the inflation pressure under the conditions of the same nominal volume, the natural frequency become small and the ratio of damping become large, which caused the capacity to suppressing pressure surge become large.



Figure 9: Simulation results with single accumulator 1.6L/10MPa and 2.5L/10MPa suppressing water hammer

As is shown in the figure 9, the pressure surge with accumulator has reduced greatly by contrast to the results in the figure 7 without accumulator. The maximum pressure is 10.80 MPa, the minimum pressure is 9.24 MPa with accumulator 1.6L/10MPa, while the maximum pressure is 10.70 MPa, the minimum pressure is 9.48 MPa with accumulator 2.5L/10MPa.

With the increasing of the nominal volume under the conditions of the same inflation pressure, the natural frequency become small and the ratio of damping become large, which caused the capacity to suppressing the surge pressure become large, too.

4.2 With multi-accumulator



Figure 10: With multi-accumulator before the proportional direction valve

As is shown in the figure 10, the muffler is equivalent to lots of parallel accumulators from the structure and the mechanism to suppressing water hammer. However, it is impossible to install numbers of accumulators in the ships in the limitation of the cost and space of installation. So the case of multi-accumulator is replaced by the muffler both in simulation and experiment. In the paper, the length of hydraulic muffler is 0.4m, the inner diameter is 28mm, the outer diameter is 94mm, the length of the hole is 1.5mm, the diameter of the holes is 0.8mm, and the number of the holes are 368.



Figure 11: Simulation results of hydraulic mufflers with different inflation pressure



Figure 12: Simulation results of hydraulic mufflers with different nominal volume

As is shown in figure 11, with the increasing of the inflation pressure under the conditions of the same nominal volume, the capacity of suppressing pressure surge become large. As is shown in figure 12, with the increasing of the nominal volume under the conditions of the same inflation pressure, the capacity of suppressing pressure surge become large, too.

As the model of hydraulic muffler is second order system, too. The M_p is proportional to the natural frequency and is inversely proportional to the ratio of damping, while the inflation pressure and the nominal volume are all inversely proportional to the natural frequency of the system and proportional to the ratio of damping of the system.

5. Conclusions and discussions

The precise mathematical models of single accumulator and multi-accumulator were established in the paper. The systems with single accumulator and multi-accumulator are all second-order systems which the natural frequency and the ratio of damping are important parameters. The simulation presented that choosing hydraulic accumulators of convenient nominal volume and inflation pressure is important to suppressing pressure surge. The study showed that the effect to reduce the amplitude of pressure surge using accumulator is related to the nature frequency and ratio of damping of accumulators.

6. References

- /1/ Brunone, B., Golia, U. M., and Greco, M. 1991. "Some Remarks on the Momentum Equation for Fast Transients," *Proc. Int. Conf. on Hydr. Transients With Water Column Separation*, IAHR, Valencia, Spain, pp. 201–209.
- /2/ Cai Y. G. 1990. Dynamics of Hydro-Transmission of pipeline[M]. Hangzhou: Press of Zhejiang University. (Chinese)
- /3/ Goldberg, D. E., and Wylie, E. B. 1983. "Characteristics Method Using Time-Line Interpolations," J. Hydraul. Eng. 109(5), pp. 670–683.
- /4/ Hu, S. S. 2001. Principles of Automatic Control[M]. Beijing: Press of Science.(Chinese)
- /5/ Jiang J., Weng X. H., Zhang X. H. 2010. Water-hammer Calculation and Protection of Condensate Pump System . 2010 THE SECOND CHINA ENERGY SCIENTIST FORUM. 2nd China Energy Scientist Forum, OCT, 2010, Xuzhou, China, pp. 73-77.
- /6/ Karney, B. W., and Ghidaoui, M. S.1997. "Flexible Discretization Algorithm for Fixed Grid MOC in Pipeline Systems," J. Hydraul.Eng. 123(11), pp. 1004–1011.
- /7/ Lai, C. 1989. "Comprehensive Method of Characteristics Models for Flow Simulation," J. Hydraul. Eng. 114(9), pp. 1074–1095.
- /8/ Mohamed S. Ghidaoui , Ming Zhao, Duncan A. McInnis, David H. Axworthy.
 2005. A Review of Water Hammer Theory and Practice[J]. Appl. Mech. Rev. 58(1), pp.41-76.
- /9/ Rabie, M. Galal. 2007. On the application of oleo-pneumatic accumulators for the protection of hydraulic transmission lines against water hammer - A theoretical study, International Journal of Fluid Power, 8(1), pp.39-49.
- /10/ Sibertheros, I. A., Holley, E. R., and Branski, J. M. 1991. "Spline Interpolations for Water Hammer Analysis," J. Hydraul. Eng.117(10), pp. 1332–1349.
- /11/ Streeter, V. L. and Wylie, E. B. 1985. Fluid Mechanics (8th Edition),McGraw Hill New York.

- /12/ Trikha, A. K. 1975. "An Efficient Method for Simulating Frequency-Dependent Friction in Transient Liquid Flow," ASME J. Fluids Eng. pp. 97–105.
- /13/ Vardy, A. E., and Brown, J. M. B. 1995. "Transient, Turbulent, smooth Pipe Friction," J. Hydraul. Res. 33, pp. 435–456.
- /14/ Yang, S. Z. & Yang, K. C. 2001. Mechanical engineering control foundation.Wuhan: Press of Huazhong University of Science and Technology. (Chinese)

7. Symbols

A_1	Cross-sectional area of pipeline connected to accumulator	m^2
A_2	Cross-sectional area of the neck of accumulator	m^2
A_{ac}	Cross-sectional area of the body of accumulator	m^2
a	Speed of pressure wave	<i>m / s</i>
В	Pipe thickness	т
B _e	Equivalent damping coefficient of the system with single accumulator	
B'_e	Equivalent damping coefficient of the system of multi-accumulator	
C_a	Gas damping coefficient.	
d	Diameter of the pipeline	т
Ε	Elastic modulus of the pipe	N/m^2
k	Air polytropic exponent	
K _e	Stiffness of gas spring in system with single accumulator	N / m
K'_e	Stiffness of gas spring in system with multi-accumulator	N / m
L	Fluid induction	т
l_1	Length of pipeline connecting to accumulato	т
l_2	Length of the neck of accumulator	т
l _{ac}	Length of body of accumulator	
m_1	Mass of the liquid of pipeline connecting to accumulator	Kg
m_2	Mass of the liquid of the neck of accumulator	Kg
<i>m</i> _{ac}	Mass of the liquid in the body of accumulator	Kg
m _e	Effective mass of the system with single accumulator	Kg
m'_{e}	Effective mass of the system with multi-accumulator	Kg
M_p	Maximum overshoot	
Ν	Number of accumulators	
_		

 ρ Oil density

Р	Pressure of the pipeline	Pa
Q	Flow of the pipeline	m^3 / s
R	Fluid resistance	
t_p	Peak time	S
t _s	Adjustment time	S
v_0	Flow velocity before the valve.	<i>m / s</i>
v_1	Flow velocity in the joint of the pipeline	<i>m / s</i>
v	Kinematic viscosity of hydraulic oil	m^2/s
W _n	Natural frequency with single accumulator	rad / s
w'_n	Natural frequency with multi-accumulator	rad / s
Y	Oscillation frequency	
ζ	Damping ratio of the system with single accumulator	
ζ'	Damping ratio of the system with multi-accumulator	
ξ	Local resistance coefficient	
μ	Dynamical viscosity of hydraulic oil	Pa•s
μ'	Poisson's ratio of the pipe	