Mitigating check valve slamming and subsequentwater hammer events for PPFS using MOC

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Abstract The method of characteristic (MOC) was adopted to analyze the check valve-induced water hammer behaviors for a Parallel Pumps Feedwater System (PPFS) during the alternate startup process. The motion of check valve disc was simulated using inertial valve model. Transient parameters including the pressure oscillation, local flow velocity and slamming of the check valve disc etc. have been obtained. The results showed that severe slamming between the valve disc and valve seat occurred during the alternate startup of parallel pumps. The induced maximum pressure vibration amplitude is up to 5.0 MPa. The scheme of appending a damping torque to slow down the check valve closing speed was also performed to mitigate of water hammer. It has been numerically approved to be an effective approach.

Key words Water hammer, Check valve slamming, Method of characteristic, Inertia-type valve model

1 Introduction

The water hammer, a harmful risk to nuclear power plant (NPP) systems, is an important issue[1,2]. A severe accident of check valve-induced water hammer occurred in 1985 in Unit 1 reactor of the San Onofre Nuclear Generation Station[3]. The reactor was forced to shutdown because of the water hammer-induced pipe breakage and abruption of pipe support systems.

Studies on water hammer problem have been conducted increasingly with measurement techniques or numerical simulations. Based on findings in the experimental studies, and according to acceptable physical mechanisms, the terms of valve-induced, void-induced and condensation-induced water hammers were proposed by relevant authors[2-6]. In the numerical simulations, among the models to solve the water hammer problems, the method of characteristics (MOC) is the most popular in modeling the valve-induced water hammer transient because of its feasibility and advantage in solving complex system problem. The MOC is also adopted by commercial computer codes, e.g. AFT-Flowmaster, which was developed for one dimensional flow hydraulic analysis. Studies on accurate evaluation of valve-induced and cavitation-induced water hammer by using NPP safety analysis code Relap5 etc. are also being done[7,8].

While the physical mechanism of water hammer is clear, cares should be taken in applying the results obtained on certain test facilities to a particular system, because the water hammer characteristics is greatly influenced by the test facility geometry, layout and thermal physical property of working fluid.

In this paper, the MOC is adopted to analyze the valve-induced water hammer behaviors of a Parallel Pumps Feedwater System (PPFS) during the alternate startup of parallel pump. The motion of check valve is evaluated based on the inertia valve model. The transient pressure wave propagation, local flow velocity, check valve phenomena and other parameters

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are analyzed in detail, aimed at preventing water hammer problems from happening.

2 The parallel pumps feedwater system

The PPFS in the present study is comprised of two parallel mixed-flow pumps connected with two check valves, a pressure container, a flow regulating valve and pipe lines (Fig.1). Check valves are widely applied in NPP systems because of the simple structure and high automatic performance. The schematic diagram of the present check valve is shown in Fig.2. The valve opening angle, θ, is the angle between the valve disc and the horizontal axis. To ensure a certain amount of natural circulation flow rate, θ is kept at about 70° (nearly closed position of the valve) under natural stationary condition. The valve fully opens at θ= 0°, and fully closed when θ is equal 100° (not 90°, like general check valve).

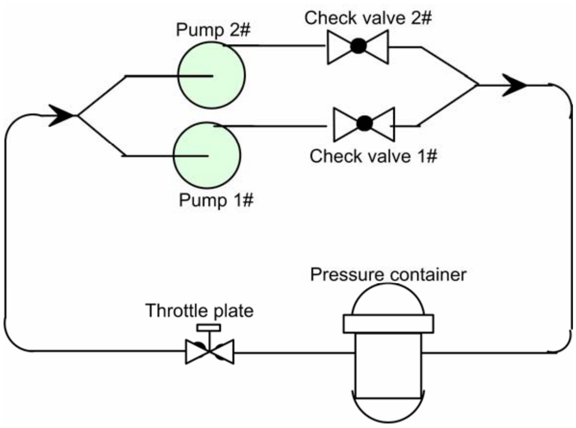

Fig.1 Sketch map of PPFS.

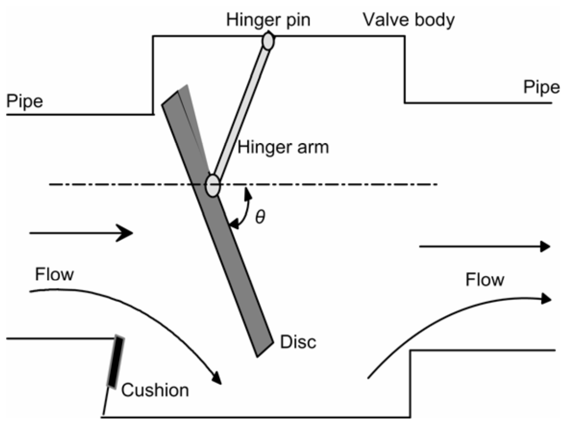

Fig.2 Sketch diagram of check valve.

Under normal working condition, one parallel pump is in operation and the other in standby/backup status. Starting a pump causes inverse flow in the other pump branch, and severe slamming occurs between the check valve disc and the valve seat. Therefore, serious water hammer phenomena may occur.

The traditional MOC was adopted to evaluate the water hammer behaviors during the alternate startup of parallel pumps. Basic principle of MOC, characteristic equations and corresponding boundary condition equations for divergent/convergent branch flow, container, regulation valve and other components, can be referred in Refs.[9,10].

3 Motion evaluation of the check valve disc

Simulating the check valve disc motion is a key point for water hammer analysis[7]. Simple valve models adopted in AFT and Relap5 are not suitable to simulate the check valves under investigation because their minimum/maximum valve flapper angle positions are different from those in the simple modes. Furthermore, there is very limited user experience in applying the valve component models[11]. Instead, the inertia-type check valve model was adopted in this work. A two-order ordinary differential equation deduced from Newton's second law was used to simulate the motion of the valve disc.

$$I\frac{d^2\theta}{dt^2} = M_{\rm RF} + M_{\rm P} + M_{\rm H} + M_{\rm B} - M_{\rm V} - M_{\rm F} + M_{\rm R} \tag{1}$$

where I is pump rotary inertia, kg·m^2; $M_{\rm RF}$ is the instantaneous rebound torque caused by the slamming between valve disc and valve seat, N·m; $M_{\rm P}$ is the torque caused by the pressure difference between the upstream and downstream of the valve disc, N·m; $M_{\rm H}$ is the hydraulic torque, N·m; $M_{\rm R}$ is the buoyancy force torque, N·m; $M_{\rm V}$ is the gravity torque, N·m; $M_{\rm F}$ is the friction torque caused by the rotation of valve disc around the rotating shaft, N·m; $M_{\rm R}$ is the friction torque caused by the fluid flow through the surface of the valve disc, N·m. The items of $M_{\rm RF}$ $M_{\rm F}$ and $M_{\rm R}$ are not considered in some previous studies[11-13].

The slamming force between the valve disc and seat can be calculated by[10]

$$P = \omega_{\rm m}\sqrt{\frac{fE}{\delta}I} \tag{2}$$

where P is the mean slamming force, $\omega_{\rm m}$ is the valve disc angular velocity before collision, rad/s; f is the contact area of seat cushion, m^2; E is the elastic ratio of cushion material, GPa; δ is the thickness of the valve base cushion, m.

The slamming torque can be calculated by

$$M_{\mathrm{RF}} = P \cdot L_{\mathrm{RF}} \tag{3}$$

where L_{RF} is the rebound force arm length, m.

For evaluating the rebound torque, the collision duration, a sensitive parameter, should be carefully compared with computational time step. The rebound torque is only valid within the collision duration. The geometry and material of the valve influence the rebound torque value and collision duration evidently

$$M_{\mathrm{P}} = (P_1 \cdot S_1 \cdot L_1 - P_2 \cdot S_2 \cdot L_2) \tag{4}$$

where, P_1 and P_2 are upstream and downstream pressure of valve disc, Pa; S_1 and S_2 are action area of P_1 , P_2 on two opposite sides of valve disc, m^2; L_1 and L_2 are corresponding arm length, m.

M_{P} is considerable when the pressure drop through the valve is much higher when the valve disc is in nearly closed position under certain mass flow condition.

$$M_{\mathrm{H}} = C_{\mathrm{T}} D^3 \frac{1}{2} \rho_{\mathrm{c}} (V - L\omega) \tag{5}$$

where C_{T} is coefficient; D is valve disc diameter, m; ρ_{c} is fluid density, kg/m^3; ω is angular velocity, rad/s; L is arm length, m; and V is fluid flow velocity, m/s.

C_{r}, which is usually illustrated as $C_{\mathrm{T}}=f(\theta)$, can be determined experimentally.

M_{F} is quantitatively proportional to the square value of valve disc rotary speed, and the action direction is opposite to the valve disc's rotation.

$$M_{\mathrm{F}} = C_{\mathrm{K}} \cdot \omega |\omega| \tag{6}$$

where C_{K} is coefficient and ω is valve disc angular velocity, rad/s.

M_{R} is related to relative velocity difference between the fluid flow and valve rotation.

$$M_{\mathrm{R}} = K \cdot \rho_{\mathrm{c}} (V - \omega L_{\mathrm{m}}) |(V - \omega L_{\mathrm{m}})| \cdot S \cdot L_{\mathrm{m}} \tag{7}$$

where K is flow friction coefficient, S is disc surface area, m^2; and Lm is the arm length (the distance between the valve geometry center to rotary shaft), m.

Each torque component can be calculated using upstream and downstream flow parameters of the check valve. Then, Eq.1, a two-order ordinary differential equation that expresses the valve disc motion, can be discretized and explicitly solved. It should be noted that the flow area between the valve flapper and the valve seat behaves an orifice whose area changes in time as a function of inertia valve geometry. The energy loss efficient also changes with the varying flow area from 1 to 0.5[11].

4 Results and discussion

The water hammer events during the alternate startup of parallel pumps were numerically simulated, with water as working fluid, initial conditions of 3.0 MPa system pressure and 30℃ fluid temperature.

The startup series is as follows: Pump 1 starts first and fails in 10 s on instantaneous start of Pump 2. Fig.3a shows the transient volume flow rate of two parallel pumps. The flow rate of Pump 1 increases rapidly from zero to 1500 m^3/h within 2 s after startup. The startup of Pump 1 induces a small inverse volume flow in Pump 2 branch since Valve 2 is not fully closed under natural stationary state. Ten seconds later, Pump 1 is shutdown and Pump 2 starts promptly. The flow rate of Pump 1 begins to decrease and the flow rate of Pump 2 increases. This induces a considerable inverse flow (up to 400 m^3/h) in Pump 1. This inverse flow induces severe slamming of Valve 1. In Fig.3b, the transient opening angle θ of Valve 1 and 2 indicates the motion of valve disc. Valve 1 is fully opened within 0.7 s after the startup of Pump 1. The startup of Pump 1 can cause inverse flow in Pump 2 branch which induces several times of slamming of Valve 2 before full closure. This slamming vanishes gradually with reduced vibration amplitude of upstream and downstream pressure of Valve 2 (Fig.3d). Ten seconds later, when Pump 2 starts after Pump 1 is shutdown, Valve 2 is fully opened from closure condition within 0.65 s and Valve 1 is fully closed from fully opening state within 0.5 s followed by severe slamming and pressure oscillation (Fig.3c). From Fig.3c and Fig.3d, the pressure oscillation amplitude at Valve 1 is up to 5.0 MPa, a damaging pressure for the valves and pipelines. We note that the minimum upstream pressure of Valve 1 is only 0.5

MPa corresponding saturated temperature 151℃. Higher emperatures may cause flash evaporation and subsequent cavitation-induced water hammer.

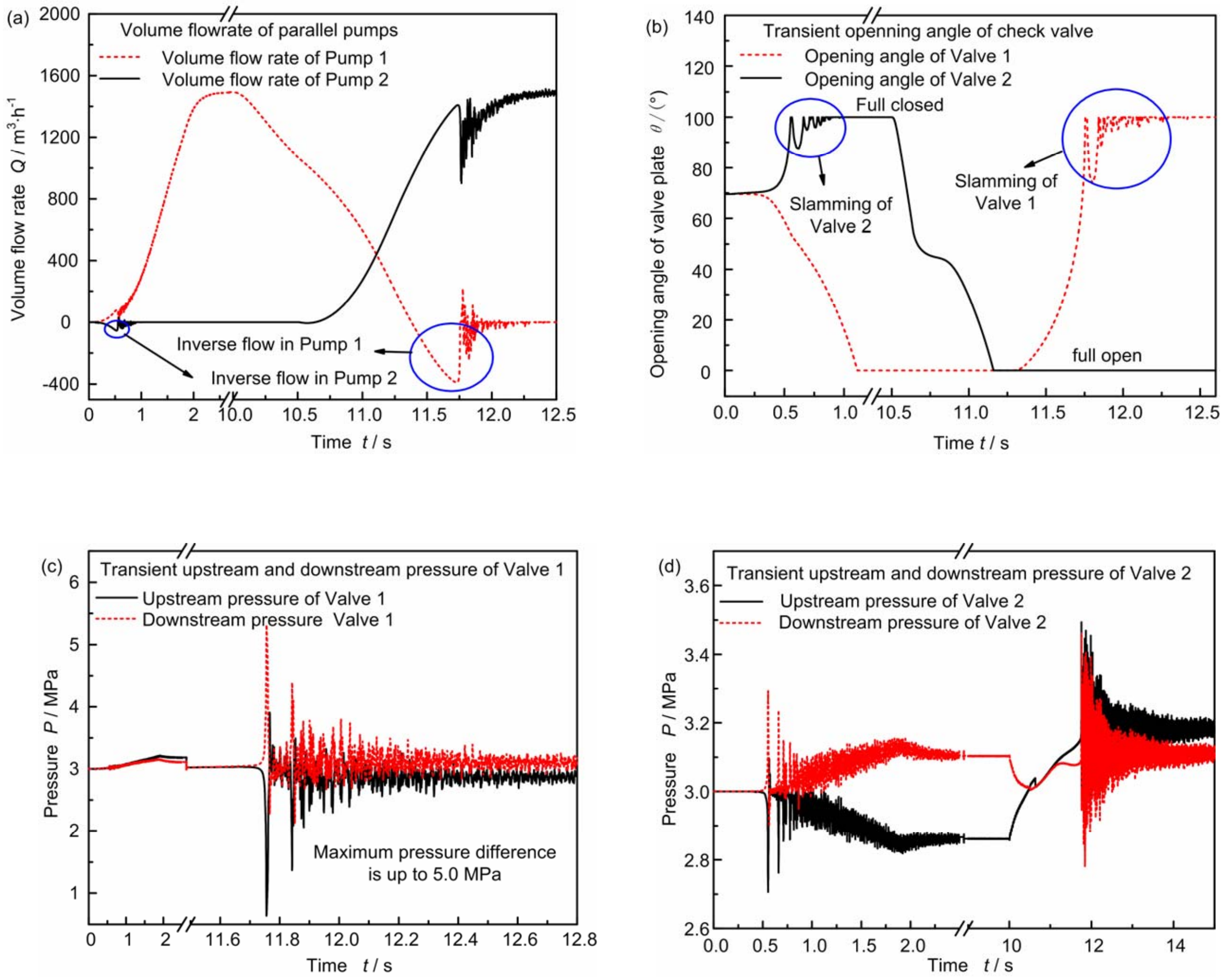

Fig.3 Water hammer behaviors of PPFS. (a) Volume flow rate intwo parallel pumps, (b) Opening angle of Valve 1 and Valve 2, (c) Upstream and downstream pressure of Valve 1 and (d) Upstream and downstream pressure of Valve 2.

To mitigate the slamming, pressure pulse and other water hammer subsequence were evaluated. For eliminating valve-induced water hammer, Thorley[12] proposed that the check valve slamming can be alleviated by minimizing the reverse flow rate just before the valve closes, and one of the ways is to reduce the flapper closing speed (lengthen the valve closure time) when the flapper near its full closure position. Based on this idea, a suppositional spring damping was used to reduce the closing speed of the flapper. The optimum spring damping can supply the additional damping torque MS=30ω(θ–90°), near the region of flapper closure position, 90°<θ<100°. The computational results after optimization are shown in Fig.4. Fig.4a shows the transient volume flow rate of two parallel pumps which indicates the flow oscillation has been greatly eliminated compared with Fig.3a. From Fig.4b, the damping torque reduces the valve closing speed effectively and slamming phenomena can be eliminated. The upstream and downstream pressure of Valve 1 is shown in Fig.4c. In Fig.5, the results before/after optimization are given. No violent flow instability is found in Pump 1 branch after optimization (Fig.5a), and the sequent maximum pressure oscillation amplitude decreased from 5.0 MPa to 1.6 MPa (Fig.5b).

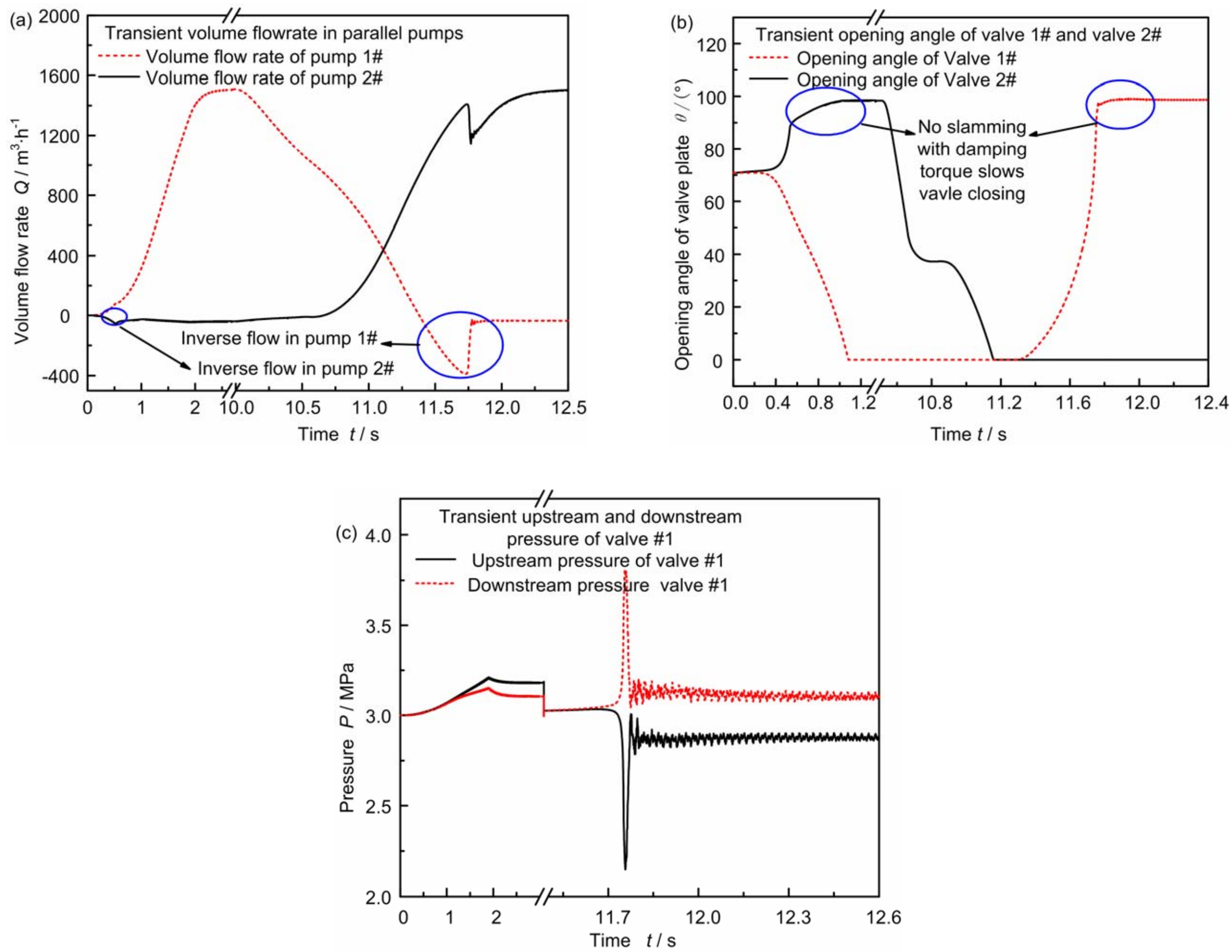

Fig.4 Computational results after optimization. (a) Volume flow rate in two parallel pumps, (b) Opening angle of Valve 1 and Valve 2 and (c) Upstream and downstream pressure in of Valve 1.

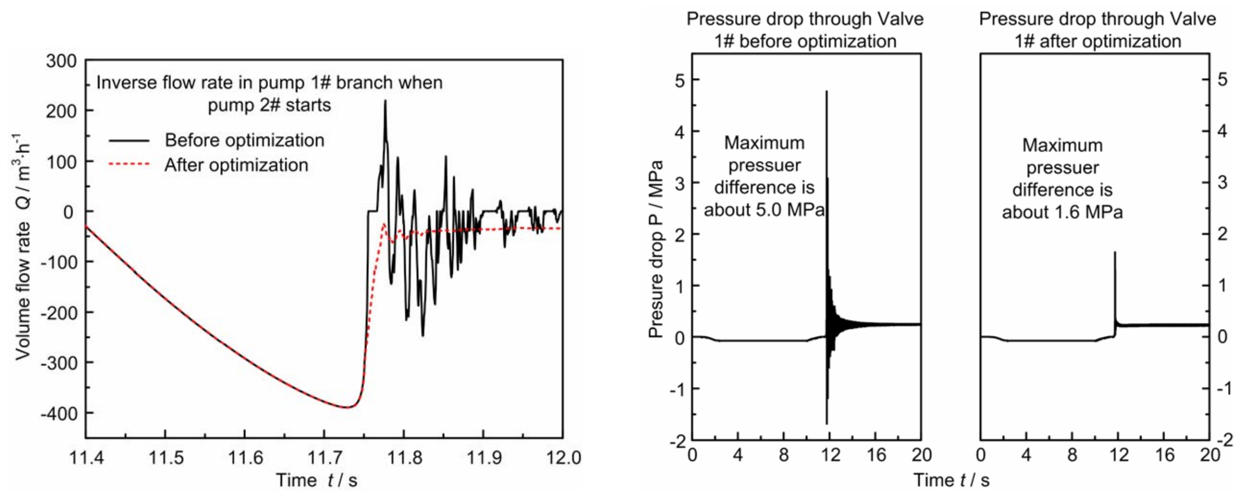

Fig.5 comparison on results before/after optimization. (a) Inverse flow rate in Pump 1 branch and (b) Pressure oscillation amplitude in Valve 1.

5 Concluding remarks

The conventional MOC supplied with inertia-type valve model were adopted to evaluate the water hammer events of a PPFS during the alternate startup process. Severe valve slamming phenomena has been captured and the induced maximum pressure vibration amplitude is up to 5.0 MPa. For the purpose of mitigating water hammer sequence, an optimum spring damping was adopted to slow the valve disc closing speed and it has been approved to be an effective approach. The present computational results are expected to be instructive for the optimum design of the PPFS to mitigate the damage potential induced by valve induced water hammer.

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