

Article

Optimizing Water Hammer Protection Strategies for Water Conveyance Systems at Floating Pump Stations Experiencing Large Water Level Fluctuations

Yu Zheng ¹, Jianyong Hu ^{2,3,*}, Gaohui Li ⁴, Jinke Mao ^{2,5}, Jinfeng Hu ^{2,5} and Yubing He ^{2,6}

¹ Standard & Quality Control Research Institute, Ministry of Water Resources, Hangzhou 310012, China; solomon128@126.com

² Institute of Water Sciences, Zhejiang University of Water Resources and Electric Power, Hangzhou 310018, China; mjk4310@163.com (J.M.); 13905702216@163.com (J.H.); 19975263649@163.com (Y.H.)

³ Engineering Research Center of Digital Twin Basin of Zhejiang Province, Hangzhou 310018, China

⁴ Powerchina Huadong Engineering Corporation Limited, Hangzhou 311122, China; li_gh2@hdec.com

⁵ School of Marine Engineering Equipment, Zhejiang Ocean University, Zhoushan 316022, China

⁶ School of Water Resources and Electric Power, Hebei University of Engineering, Handan 056038, China

* Correspondence: hujy@zuwe.edu.cn; Tel.: +86-18757109986

Abstract

Under large water level fluctuations, emergency pump shutdown in floating pump stations tends to induce severe water hammer oscillations, with consequences including excessive negative pressure, potential cavitation and water-column-separation risk, and uncontrolled pump reverse rotation; this seriously threatens the operational safety of water intake systems. We seek to develop an economical and effective water hammer protection strategy. Therefore, we study a long-distance water conveyance project with a floating pump station, establish a numerical model of hydraulic transient processes, and carry out a numerical simulation of a pump-trip water hammer, enacting a combined protection strategy that integrates rules on pump outlet valve closure and air vessels. Meanwhile, we analyze the regulation effects of key parameters, including air vessel volume, connecting pipe diameter, and water–air ratio (the ratio of initial water volume to initial air volume inside the air vessel), on the pipeline pressure and pump rotational speed. Overall, with the simultaneous shutdown of three pumps at the minimum operating water level, negative pressure seriously exceeds its limits in the absence of protective measures. In optimizing the valve closure rule alone, we can effectively suppress reverse pump rotation, but this fails to effectively address the issue of negative pipeline pressure. After introducing air vessels to form a combined protection system, the indicative negative-pressure severity in the pipeline significantly improves due to the synergistic effect of valve closure (suppressing reverse rotation) and air vessels (regulating the pressure). On this basis, we identify the best-performing option among the investigated cases, on the premise that a larger volume is economically unjustified, via multi-parameter comparative analysis. This consists of a two-stage valve closure rule (a 5 s fast closure duration + 80% fast closure angle + 35 s slow closure duration) alongside an air vessel with a 15 m³ volume, a 0.3 m connecting pipe diameter, and a 1:1 water–air ratio, and a terminal air valve with a 300 mm air intake diameter. This scheme balances hydraulic safety and engineering economy; the maximum pipeline pressure, minimum pipeline pressure, and maximum pump reverse speed all meet specification requirements, enabling risk control for the investigated pump-trip water-hammer conditions. These research findings provide a theoretical basis

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and technical support for the design of water hammer protection schemes in similar water conveyance projects at floating pump stations.

Keywords: floating pump station; water hammer protection; valve closure rule; air vessel; combined protection strategy

1. Introduction

China has experienced rapid economic and social development, with a resulting increased prominence of unevenly distributed water resources, both temporally and spatially. As a result, long-distance high-lift water conveyance projects have become critical infrastructure, alleviating regional water shortages, ensuring secure urban and rural water supply, and promoting sustainable economic and social development. Such projects generally involve long conveyance distances, complex terrain, high lift, and a large system scale, and their safe and stable operation is directly related to the national economy and the livelihood of the population. However, the water hammer effect threatens pipeline safety during the operation of long-distance high-lift water conveyance systems. The superimposed effect of complex terrain and high lift further aggravates the risk of water hammer accidents, especially in pressurized pumping conveyance systems. To avoid or reduce the hazards caused by water hammer, scholars worldwide have sought protection against it, the primary purpose being to reduce the intensity and duration of water hammer pressure oscillations, ensuring safe pipeline operation within the allowable pressure range. Therefore, controlling the extreme transient pressure induced by water hammer is a key research topic in the field of hydraulics.

For high-lift pumping stations, rapid valve closure can significantly change the flow momentum in channels, trigger obvious pressure fluctuations, and potentially lead to water hammer damage. Optimizing valve closure mechanisms is an effective technical approach to mitigate pressure shocks in the system, and valve regulation strategies have thus become a key research direction in water hammer protection. Ouyang et al. [1] proposed a water hammer protection scheme with gates installed at the upstream end for ultra-long pressurized diversion tunnels. After water hammer occurs in the upstream and downstream sections during water conveyance, delaying the gate opening time by 40 min and adopting a uniform slow-opening control mode can constrain the pressure and flow fluctuations inside the tunnel within a safe range. This method effectively suppresses adverse flow patterns, such as gas–liquid mixing, large bubble formation, and water column separation, and improves the efficiency of water conveyance while reducing water hammer pressure. Gate operation strategies such as this, with precise timing and rate regulation capability, can also be applied to control transient pressure fluctuations in pipelines induced by pump shutdown. Lohrasbi and Attarnejad [2] found that valve opening and closure cause flow fluctuations in water conveyance pipelines, and numerical calculations confirmed that shortening the valve closure time generates enormous water hammer pressure, necessitating an extension of the valve closure duration. Xue et al. [3] conducted numerical analysis via the Flowmaster simulation platform, showing that a staged gate closure strategy, starting fast and later slowing down, can effectively weaken the pressure peak and severe oscillations caused by water hammer. The fast closure stage accounts for two-thirds of the total stroke, corresponding to an opening adjustment range of 0.6–0.3, with an optimal opening threshold of 0.4. This optimized scheme can effectively guarantee the operational safety and steady-state performance of the pipeline system. Lescovich [4] revealed the correlation between the length of the water conveyance pipeline and the gate closure duration: the longer the pipeline, the more prominent the pressure wave

effect caused by water inertia during rapid gate closure; accordingly, the gate closure time needs to be extended. Wang and Liu [5] also verified this conclusion; they investigated multiple gate control schemes in a bidirectional-flow-passage pumping station, including single-side gate opening and simultaneous opening of both sides. The flow velocity distribution in the inlet area of the flow passage was the most uniform when keeping the inlet-side gate fully open and regulating flow only through the outlet-side gate opening. Romuald et al. [6] studied valve closure rules and proved that staged valve closure can reduce the impact of water hammer compared with linear closure rules. Liu et al. [7] investigated a two-stage valve closure rule for minimizing pump-trip water hammer; with the same total valve closure duration, balancing between a fast closure time and the angle was conducive to improving the safety of pump pipeline systems. In complex pump-valve systems, the operation time interval has a significant impact on the hydraulic transient process of switching operating states; appropriate delay can not only reduce the transient pressure during startup but also prevent reverse rotation of the pump [8,9]. Han et al. [10] used computational fluid dynamics to numerically simulate ball valves with different closure times and rules; they found that a faster valve closure speed in the early stage led to greater water hammer pressure, whereas extending valve closure time could effectively reduce its maximum. This rule also confirms the research conclusions of the aforementioned scholars, explaining the interaction between valves and water hammer, which provides theoretical guidance for optimizing valve operation. Toumi and Sekiou [11] showed that the optimal valve closure law depends on the ratio of slow-closure time to pipeline reflection period; above 0.50, the optimal exponent becomes a damped periodic function and its range narrows with longer closure, aiding dynamic selection of closure laws.

Nevertheless, in long-distance water conveyance projects, valve regulation alone is often insufficient to achieve full-line water hammer control, and it must usually be combined with other protection devices such as air vessels, surge tanks, and air valves. As an emergency water supply source, air vessels [12,13] perform vital functions in protecting against water hammer. When pumps stop operating, the gas in the air vessel expands rapidly to replenish necessary water and limits the negative pressure inside the pipeline. When the pipeline pressure rises and forces water to flow into the air vessel, the gas inside the vessel is compressed, thereby preventing an increase in the pipeline pressure. Thus, air vessels are an effective means to prevent both positive and negative water hammer. Their effect is particularly prominent in high-lift water diversion projects with low flow rates, significantly reducing the impact of water hammer on pipelines. Air valves [14] can effectively remove accumulated air masses in the system and eliminate flow shocks, preventing positive pressure from damaging the water conveyance system. Meanwhile, when the pipeline pressure drops sharply, air valves can suck air in quickly to cut off the water hammer propagation path, thus avoiding pipeline damage caused by water column separation. Surge tanks [15] utilize the free water surface in a nearby tower to absorb pressure waves generated by pipeline transients and convert them into surge fluctuations. Pressure relief valves open automatically to discharge part of the flow when the water pressure in the pipeline exceeds the maximum allowable value, rapidly reducing the positive pressure behind the valve to ensure pipeline safety. Bettaieb and Taieb [16] conducted a transient analysis using a MATLAB 2014b program based on the linear integral method of characteristics; they showed that combining check valves with an optimized closure rate and non-return valves can effectively control transient pressure. Ali et al. [17] simulated water hammer in water supply networks using the WHAMO software and verified the protective effect of open buffer tanks paired with check valves. Khokhlov et al. [18] innovatively proposed a combined scheme of bypass devices and proportional control butterfly valves to protect hydraulic equipment. Himr et al. [19] numerically simulated a hydraulic model to reveal a better-than-expected protective effect of air vessels

against pump-trip water hammer. Lyu et al. [20] proposed a combined protection scheme involving air vessels and safety valves, which significantly reduced the air vessel volume. Chen et al. [21] verified the superiority of combining air vessels with air valves through numerical simulation. In addition, the authors studied opening and closure rules of over-pressure relief valves; they should be opened quickly to achieve rapid discharge and suppress maximum pressure rise but closed as slowly as possible to avoid large pressure waves. Hamed and Abdulaziz [22] experimentally and numerically optimized air vessel performance, identifying orifice diameter as the most critical parameter affecting water hammer head and vessel size, thus providing a reference for balancing safety and economy. Ilić et al. [23] analyzed the hydraulic transient process of water conveyance systems with air vessels, proposing a combined protection measure that involved installing an air vessel behind the pump and arranging an air valve at the end of the pipeline. Compared with using air valves or vessels alone, this measure can greatly eliminate water hammer pressure and effectively reduce the air vessel volume.

In summary, existing studies have accumulated abundant achievements—optimizing valve opening and closure rules, conducting performance analyses of various water hammer protection devices, and enacting multi-measure combined protection—laying a solid theoretical and technical foundation for risk prevention and control of water hammer in long-distance water conveyance projects. However, most of these studies focus on conventional onshore pumping stations and fixed pipelines, with less attention paid to water-lifting projects at floating pump stations. There is a lack of in-depth analysis of the synergistic regulation mechanisms of valve closure rules and air vessels, as well as their overall protective effect on floating pump stations. Therefore, in this paper, we study a water-lifting project at a floating pump station to propose a combined protection strategy involving valve closure rules and air vessels whilst considering actual engineering conditions. Different from previous studies focusing on conventional onshore pumping stations, this paper targets the unique characteristics of floating pump stations with large water level fluctuations, including variable working head and rocker-arm pipeline layout. Based on the method of characteristics, we establish a refined numerical model of hydraulic transient processes for floating intake systems, and integrate two-stage valve closure, air vessels and terminal air valves to form a multi-level combined protection system. On this basis, we conduct multi-criteria hydraulic optimization covering three core indicators: maximum pipeline pressure head, minimum pipeline pressure head and maximum pump reverse speed, while taking engineering economy into account by reducing the volume of air vessels to cut down construction and operation costs. We systematically analyze the control effect of our combined protection scheme on water hammer pressure along the line, and demonstrate the necessity and feasibility of the scheme. Finally, we formulate the best-performing hydraulic safety regulation strategy among the investigated cases. The results of this research may provide a reference for the design and optimization of water hammer protection in similar water-lifting projects at floating pump stations.

2. Methods

2.1. Mobile Water Intake

Floating water intake technology [24] features a long development history, widespread application, and a mature technical system. Compared with other methods of water intake, mobile intake pumping stations have the following advantages:

- (1) Wide applicability: Suitable for rivers and lakes; the floating vessel automatically rises and falls with the water level, keeping the suction port 1–2 m below the water surface, thereby reducing sediment concentration and debris entrainment in the intake water.

- (2) Few construction restrictions: They can be arranged regardless of whether the water intake meets navigation conditions.
- (3) Good structural stability: They comply with unsinkable design principles for three-compartment intake pumping stations.
- (4) Significant cost and construction period advantages: Compared with conventional civil pump houses, no cofferdam construction, complex underwater works, or large-scale earthwork operations are needed; the construction period is short, the project cost is low, the construction process is simple and safe, and disturbance to the surrounding environment is minimal.
- (5) Outstanding operation and maintenance convenience: They avoid the problems of fixed pumping stations buried at the bottom of water bodies, whose water inlet pipes are prone to blockage and difficult to clean and maintain.
- (6) Reliable safety guarantee: Complete and reliable supporting facilities are provided, and special lifting devices are installed to facilitate equipment maintenance and daily upkeep.

Under large water level fluctuations, the differences between floating and conventional fixed pumping stations are especially pronounced. In a fixed station, the intake elevation is fixed. When the water level varies over a wide range, the suction conditions deteriorate, the operating head changes substantially, and the pump units may operate far from their high-efficiency region; moreover, the intake pipes and structures are more likely to suffer from sedimentation and clogging. In contrast, a floating station equipped with a rocker-arm pipeline system can continuously follow the water level. The pump vessel is connected to the rocker-arm pipes through rotary sleeves, and the pipes are hinged to an onshore fixed buttress via rotary supports. As the water level fluctuates, the change in hull buoyancy drives the rocker arm to swing around the support shaft, so that the relative position between the pumps and the suction port remains essentially unchanged. This arrangement narrows the operating head range, improves pumping efficiency, and avoids frequent adjustment of intake structures.

A floating water intake structure generally consists of a hull, pumps, and a rocker-arm water pipe system connecting the pump vessel to the onshore water conveyance pipeline. Figure 1 shows the plane layout of the mobile intake pump vessel under study, and Figure 2 shows cross sections of the pumping station at the highest and lowest water levels.

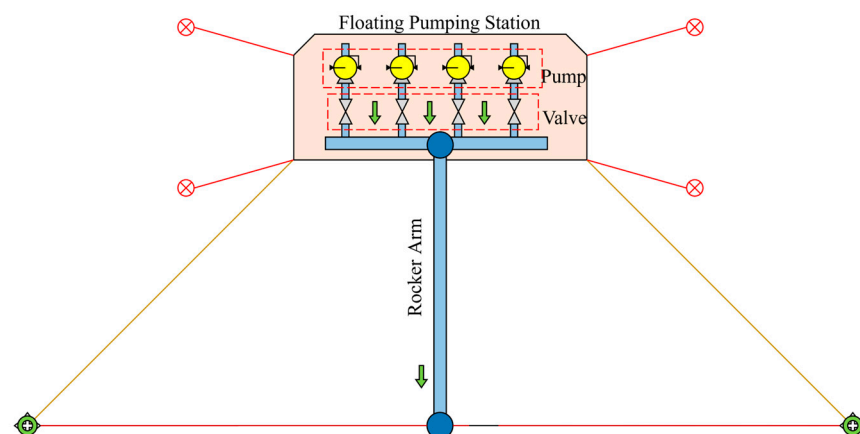


Figure 1. Plane layout of the floating pumping station.

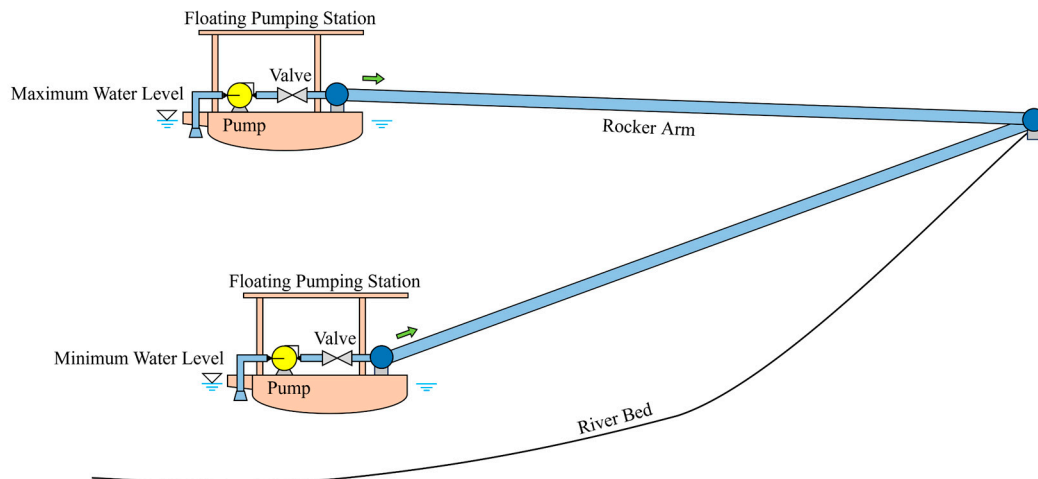


Figure 2. Cross-section of the floating pumping station.

2.2. Characteristic Compatibility Equations for Water Hammer Calculation

The basic equations describing the flow motion state in any pipeline are [2,25]:

$$\frac{Q}{A} \frac{\partial H}{\partial x} + \frac{\partial H}{\partial t} + \frac{a^2}{gA} \frac{\partial Q}{\partial x} - \frac{Q}{A} \sin \beta = 0 \quad (1)$$

$$g \frac{\partial H}{\partial x} + \frac{Q}{A} \frac{\partial (Q/A)}{\partial x} + \frac{\partial (Q/A)}{\partial t} + \frac{fQ|Q|}{2DA^2} = 0 \quad (2)$$

where H is the piezometric head, m; Q is the flow rate, m³/s; D is the pipeline diameter, m; A is the pipeline's cross-sectional area, m²; t is a time variable, s; a is the water hammer wave velocity, m/s; g is gravitational acceleration, m/s²; f is the friction coefficient; β is the angle between the pipe axis and the horizontal plane, °.

Equations (1) and (2) can be simplified into standard hyperbolic partial differential equations by neglecting the convective acceleration terms ($V \frac{\partial V}{\partial x}$ and $V \frac{\partial H}{\partial x}$), as the water hammer wave velocity a is significantly larger than the flow velocity V . These are then transformed into equivalent characteristic compatibility equations for calculating pipeline water hammer using the method of characteristics.

For a discretized pipe segment A-B with length ΔL , the following characteristic compatibility equations can be established for the transient head ($H_A(t)$, $H_B(t)$) and transient flow rate ($Q_A(t)$, $Q_B(t)$) at adjacent nodes A and B at time t [2,25]:

$$C^+: H_B(t) = C_P - R_P Q_B(t) \quad (3)$$

$$C^-: H_A(t) = C_M + R_M Q_A(t) \quad (4)$$

where $C_P = H_A(t - \Delta t) + \frac{a}{gA} Q_A(t - \Delta t)$; $R_P = \frac{a}{gA} + R|Q_A(t - \Delta t)|$; $C_M = H_B(t - \Delta t) - \frac{a}{gA} Q_B(t - \Delta t)$; $R_M = \frac{a}{gA} + R|Q_B(t - \Delta t)|$; Δt is the calculation time step; ΔL is the pipe segment length of the characteristic line grid, $\Delta L = a\Delta t$ (Courant condition); the pipeline of length L is discretized into k segments, $k = L/\Delta L$; R is the head loss coefficient, $R = \Delta h/Q^2$; and other symbols have the same meanings as above.

Hydraulic transient calculation generally starts from the initial stable operation state, i.e., $t = 0.0$ at this time. Therefore, when $(t - \Delta t) < 0$ in the equation, we let $(t - \Delta t) = 0$ (i.e., take the initial value). Equations (3) and (4) each have only two unknowns. By solving them jointly with the boundary conditions of nodes A and B, respectively, the transient parameters of nodes A and B can be obtained.

To ensure numerical accuracy and Courant compatibility, the pipeline system is discretized using the method of characteristics (MOC). The wave speed in all pipe sections is

assumed to be $a = 1200$ m/s. A uniform time step of $\Delta t = 0.001$ s is selected, yielding a spatial step of $\Delta x = a\Delta t = 1.2$ m. Consequently, the number of computational reaches for each pipe section is calculated as $N_i = L_i/\Delta x$. The shortest pipe section (5 m) is divided into 4 reaches, while the longest (1400 m) is divided into 1167 reaches. The total simulation duration is set to 200 s to ensure the transient oscillations fully decay. The convergence tolerance for the iterative boundary equations is set to 10^{-6} . The Darcy–Weisbach friction formulation is adopted. Branch junctions are treated using the standard equal-head and flow-conservation boundary conditions ($H_{up} = H_{down}$, $\sum Q_{in} = \sum Q_{out}$). A $\pm 10\%$ variation in wave speed results in a maximum pressure deviation of less than 5%, which remains within the safety margin.

2.3. Control Equations for Pump Nodes

2.3.1. Processing of Complete Characteristic Curves

The complete characteristic curves of the pump are transformed as follows [9,25]:

$$W_H(x, y) = \frac{h}{y^2(\alpha^2 + q^2)} \quad (5)$$

$$W_B(x, y) = \frac{\beta + k_1 M_{1r}' y}{\alpha^2 + q^2} \quad (6)$$

$$x(y) = \arctan\left(\frac{q}{\alpha}\right) + k_2 h \quad (\alpha \geq 0) \quad (7)$$

$$x(y) = \pi + \arctan\left(\frac{q}{\alpha}\right) + k_2 h \quad (\alpha < 0) \quad (8)$$

where h , β , a , and q are the dimensionless values of head, torque, rotational speed and flow rate, respectively; y represents guide vane opening; M_{1r}' is the unit's torque under the rated working condition, kN·m; k_1 and k_2 are coefficients, taken as $k_1 = 1.1$ and $k_2 = 0.5$.

To accurately simulate the transient behavior of the pump under extreme conditions (including zero-flow, reverse-flow, and reverse-rotation), the complete four-quadrant characteristic curves are adopted. The characteristic curves are transformed into continuous Suter curves using Equations (5)–(8). For the interpolation of $WH(x, y)$ and $WM(x, y)$, piecewise linear interpolation with a sufficiently small step size is employed to ensure numerical stability.

It should be noted that the Suter transformation introduces mathematical singularities when $q \rightarrow 0$ or $\alpha \rightarrow 0$. To prevent numerical divergence during the iteration of the method of characteristics (MOC), a small perturbation (e.g., $\epsilon = 10^{-4}$) is applied to the denominator, or a linear extrapolation based on the nearest boundary points is activated within this region.

Regarding the dimensionless torque β , the sign convention is defined as follows: $\beta > 0$ corresponds to normal pumping (forward rotation, forward flow); $\beta < 0$ corresponds to energy dissipation (forward rotation, reverse flow, or reverse rotation, reverse flow). This convention ensures the rotational moment balance equation correctly captures the deceleration and reverse acceleration processes during a pump trip.

2.3.2. Runner Boundary Head Balance Equation

Let the node numbers of the upper and lower boundaries of the runner be 1 and 2. According to Equations (3) and (4), the equation for runner boundary head balance can be obtained as [9,25]:

$$h = \frac{(C_{P1} - C_{M2})}{H_r} + \frac{q(R_{q1} + R_{M2})Q_r}{H_r} \quad (9)$$

where H_r and Q_r are the working head and flow rate of the runner under rated working conditions; other symbols have the same meanings as above.

2.3.3. Unit Rotational Moment Balance Equation

The equation for runner boundary head balance can be obtained as [9,25]:

$$\alpha = \alpha_0 + \frac{(\beta - \beta_0 + \beta_g - \beta_{g0})\Delta t}{2T_a} \quad (10)$$

where T_a is the unit inertia time constant. In standard SI units, $T_a = I\omega_r/M_r$, where I is the rotational moment of inertia ($I = GD^2/4g$, with $g = 9.81 \text{ m/s}^2$), and ω_r is the rated angular velocity. The empirical coefficient 374.7 in the original formula corresponds to a non-SI unit system (GD^2 in $\text{kgf} \cdot \text{m}^2$, n_r in r/min , and M_r in $\text{kgf} \cdot \text{m}$). To ensure dimensional consistency in this study, all parameters are converted to SI units, and the coefficient is accordingly adjusted to 38.2 (i.e., $T_a = GD^2 / n_r / 38.2 M_r$); GD^2 is the flywheel effect of the unit; n_r and M_r are the rated rotational speed and rated dynamic torque of the unit, respectively; the dimensionless torque β follows a strict sign convention: $\beta > 0$ represents normal pumping (forward rotation, forward flow), while $\beta < 0$ represents braking or turbine modes (reverse flow or reverse rotation), ensuring Equation (10) correctly captures the reverse acceleration during pump trip; β_g is the dimensionless value of the unit's rotational resistance torque; α_0 , β_0 and β_{g0} are the values of α , β , and β_g at the previous calculation time step, respectively; other symbols have the same meanings as above.

By solving Equations (5)–(10) jointly and combining them with the given closure rule $y = y(t)$, we can obtain the transient parameters h , β , α , and q of the pump–turbine node under various working conditions.

2.4. Control Equations for Pump Outlet Valve Nodes

In general, the flow equation of the outlet valve is [7,25]:

$$Q_p = \tau C_r A_r \sqrt{2g\Delta H_p} \quad (11)$$

where τ is the flow coefficient, which is a nonlinear function of valve opening and is generally expressed as discrete data or curves. Considering that the flow direction may change during the hydraulic transient process, combining this with Equations (3) and (4) gives [7,25]:

$$Q_p = \frac{C_p - C_m}{R_p + R_m + \Delta H_r |Q_p| / (Q_r \tau)^2} \quad (12)$$

Equation (12) can be solved iteratively to obtain the flow rate Q_p at the next time step.

2.5. Control Equations for Air Vessel Nodes

Assume that the air in a closed air vessel satisfies the ideal gas-state equation. Since the transient process is rapid but still permits minor heat exchange with the environment, air expansion or compression is treated as a polytropic process rather than purely adiabatic.

The main equations are as follows [21]:

$$H_A \cdot V^n = C \quad (13)$$

$$H_A = H_p - k Q_s |Q_s| - Z + H_B \quad (14)$$

$$\frac{dZ}{dt} = \frac{1}{A_C} Q_s \quad (15)$$

$$Q_s = Q_U - Q_D \quad (16)$$

$$C+: H_P = C_P - B_P Q_U \quad (17)$$

$$C-: H_P = C_M + B_M Q_D \quad (18)$$

where H_A is the absolute pressure of gas in the air vessel; H_B is the local atmospheric pressure, related to local elevation; V is the gas volume in the air vessel; n is the gas-state equation exponent representing this polytropic process; $1 < n < 1.4$, and 1.2 is commonly used in design [26,27]; C is a constant related to the initial state of gas in the air vessel; H_P is the pressure at the connection node between the air vessel and the pipeline, i.e., gauge pressure; Q_S is the flow rate into the air vessel; k is the hydraulic loss coefficient at the connection node between the air vessel and the pipeline, which takes different values for inflow and outflow according to the sign of Q_S ; Z is the water level in the air vessel; A_C is the cross-sectional area of the air vessel; Q_U is the flow rate at the upstream node of the air vessel; Q_D is the flow rate at the downstream node of the air vessel; other symbols have the same meanings as above.

Equations (13)–(18) are used to solve the pressure head and flow rate of the air vessel node, and the variations in gas volume, pressure, and water level in the air vessel can be obtained.

2.6. Control Equations for Air Valve Nodes

Air valves mitigate pressure surges by admitting air when the pipeline pressure drops below atmospheric pressure and exhausting air when the pressure rises. In this paper, the hydraulic transient calculation is based on one-dimensional flow theory, and the air valve model mainly focuses on the influence of intake/exhaust orifice diameter on air flow rate, while simplifying detailed dynamic characteristics such as valve disk opening/closing time and flow state evolution. This simplified model is suitable for comparative analysis of engineering protection schemes. The air flow rate through the air valve depends on the differential pressure across the valve, modeled by subcritical or critical (choked) flow equations [21]:

(1) Inflow Stage ($p_i < p_{atm}$):

When the pipeline internal pressure p_i drops below ambient atmospheric pressure p_{atm} , air enters the pipeline. The air mass flow rate $\dot{m}_{in}$ is expressed as:

$$\dot{m}_{in} = C_{in} A_{in} p_{atm} \sqrt{\frac{7}{RT_0} \left[\left(\frac{p_i}{p_{atm}} \right)^{1.428} - \left(\frac{p_i}{p_{atm}} \right)^{1.714} \right]} \quad \text{for } 0.53 \leq \frac{p_i}{p_{atm}} < 1.0 \text{ (Subcritical)} \quad (19)$$

$$\dot{m}_{in} = C_{in} A_{in} p_{atm} \sqrt{\frac{\gamma}{RT_0} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}}} \quad \text{for } \frac{p_i}{p_{atm}} < 0.53 \text{ (Choked)} \quad (20)$$

(2) Outflow Stage ($p_i > p_{atm}$):

When the air pocket is compressed and exhausted through the valve, the exhaust air mass flow rate $\dot{m}_{out}$ is given by:

$$\dot{m}_{out} = C_{out} A_{out} p_i \sqrt{\frac{7}{RT_0} \left[\left(\frac{p_{atm}}{p_i} \right)^{1.428} - \left(\frac{p_{atm}}{p_i} \right)^{1.714} \right]} \quad \text{for } 0.53 \leq \frac{p_{atm}}{p_i} < 1.0 \text{ (Subcritical)} \quad (21)$$

$$\dot{m}_{out} = C_{out} A_{out} p_i \sqrt{\frac{\gamma}{RT_0} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}}} \quad \text{for } \frac{p_{atm}}{p_i} < 0.53 \text{ (Choked)} \quad (22)$$

where C_{in} and C_{out} are the inflow and outflow discharge coefficients, respectively; A_{in} and A_{out} are the effective air intake and exhaust orifice areas; T_0 is absolute ambient temperature; R is the gas constant for air; and $\gamma = 1.4$ is the specific heat ratio.

The air pocket volume V_a inside the pipe at the valve location is updated by integrating the volumetric flow rate $Q_a = \dot{m}/\rho_a$ over time, coupled with boundary compatibility Equations (3) and (4) to update node head H_i^t .

In this study, the air valve model simplifies several dynamic processes to ensure computational efficiency within the 1D-MOC framework. The specific assumptions are as follows:

- (1) Intake and exhaust coefficients: The inflow and outflow discharge coefficients are set to $C_i = C_{out} = 0.6$, which is the standard engineering value recommended by related commercial software (typical values are around 0.6) and adopted in the comprehensive sensitivity analysis conducted by Tasca et al. [28]. Their study modeled the air flow through the air valve using the isentropic air flow model, considering discharge coefficients of 0.6 for both admission and expulsion.
- (2) Valve response time and orifice evolution: The valve disk opening/closing time is neglected ($t_{response} \approx 0$), assuming the effective orifice area reaches its maximum value instantaneously once the pressure differential triggers the valve.
- (3) Thermodynamic assumptions: The air inflow/outflow through the valve is treated as an isothermal process ($n = 1.0$).
- (4) Initial air state: The initial air pocket volume at the valve location is assumed to be zero, with the air pocket initialized at standard atmospheric pressure.

The sensitivity of air valve parameters has been investigated by Tasca et al. [28]. Their comprehensive sensitivity analysis demonstrated that the transient magnitude is non-linearly related to the air valve outflow capacity, and that the air valve location significantly influences the transient magnitude and the timing of air pocket collapse. These findings confirm that the discharge coefficients and valve location are the dominant parameters affecting the negative-pressure envelope, while the simplified instantaneous response and isothermal assumptions provide a conservative and reliable prediction for the downstream negative pressure control.

2.7. Numerical Model Validation

The numerical model established in this study incorporates all relevant boundary conditions, including upstream and downstream reservoirs, pumps, valves, air vessels, and air valves, and also accounts for frictional resistance and other influencing factors in the water supply pipeline. To verify the accuracy of the model, the experimental parameters of pump-trip water hammer from the physical model test conducted by Zhai [29] were adopted as a benchmark, and numerical simulations were carried out using the proposed model. The experimental system comprises a pump with a rated flow rate of 100 m³/h, a head of 20 m, a rated power of 3 kW, and a rated speed of 2900 r/min, a pipeline of 410 m in length and 0.1 m in inner diameter (DN100, galvanized steel pipe), a Manning roughness coefficient of 0.012, and a water hammer wave velocity of 1000 m/s. The air vessel has a bottom area of 0.5 m², a height of 2 m, a center pipe length of 1.2 m, and a bottom connecting pipe diameter of 0.05 m. The pump outlet valve is set to close linearly over a period of 8 s. A comparison between the experimental and simulation results is presented in Figure 3.

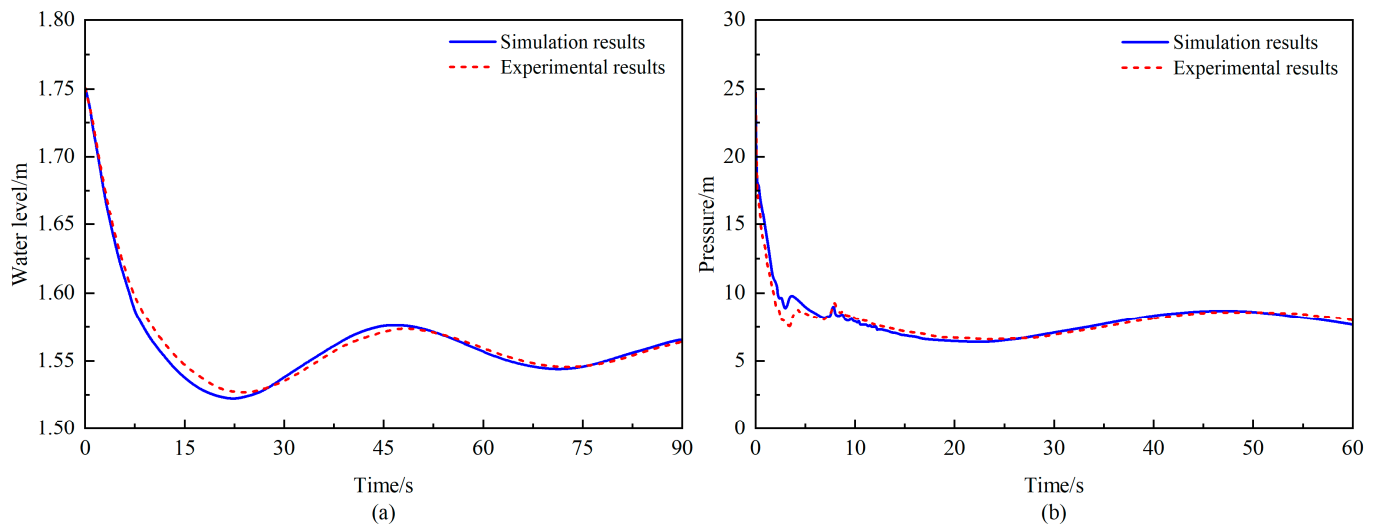


Figure 3. Comparison of simulated and experimental results: (a) water level in the air vessel; (b) pressure at the pump outlet.

As shown in Figure 3, the simulation results of the proposed numerical model exhibit good agreement with the physical model test results of Zhai et al. in terms of overall wave-forms and extreme value characteristics. Specifically, the relative errors for the maximum and minimum water levels in the air vessel are 0.11% and 0.33%, respectively, while those for the maximum and minimum pressures at the pump outlet are 0.50% and 2.88%, respectively. Although the relative error of the minimum pump outlet pressure (2.88%) is slightly higher than the others, it remains well within acceptable engineering limits, which can be attributed to the highly nonlinear characteristics of pressure fluctuations during water hammer events and systematic errors in experimental measurements. The overall small relative errors of these key characteristic parameters fully validate the high accuracy and reliability of the proposed model, demonstrating its capability to accurately capture the transient response of the system during pump-trip events. Therefore, the model can be reliably applied to subsequent hydraulic transient simulations.

3. Case Study

3.1. Project Overview

Our research object is the first-stage pumping station system of a long-distance emergency water supply project in Yunnan Province. The water supply system consists of an intake sump, pipelines, a high-level water tank, and a pumping station, as shown in Figure 4. The maximum operating water level of the intake sump is 1787 m, and the minimum is 1766 m. The water tank lies at the pump station outlet. The pressure steel pipe facilitates submerged discharge into the water tank, with the centerline elevation of the outlet pipe at 1785 m. The operating water level of the water tank is 1951.64 m, and the designed water supply flow rate is 0.927 m³/s. Given that the intake sump's water level variation reaches 21 m, a floating water intake scheme is adopted; i.e., the pumping station is floating. Four horizontal single-stage double-suction centrifugal pumps (three in operation and one on standby) are installed in the pump house. The characteristic curves and parameters of a single pump are shown in Figure 5 and Table 1. The parameters of pipelines and key nodes are shown in Tables 2 and 3. The pump's water inlet adopts a single-pipe arrangement for each unit, and the outlet adopts the same arrangement for four units. The length of the outlet's branch pipe is about 30 m, with a diameter of DN400; the diameter of the outlet's main pipe is DN800. The latter pipe is connected to the shore via a rocker arm joint, with a length of about 610 m.

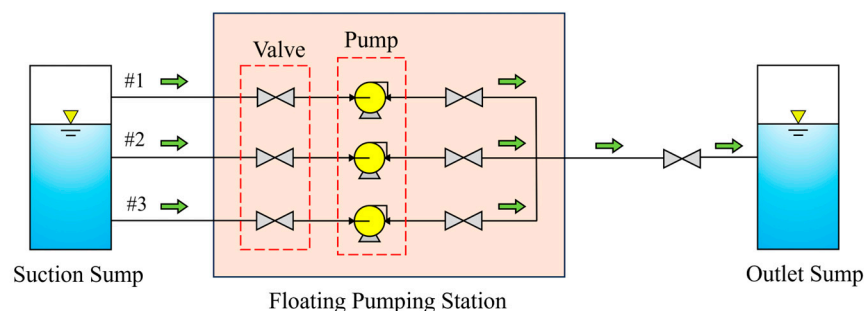


Figure 4. Schematic diagram of the water supply system.

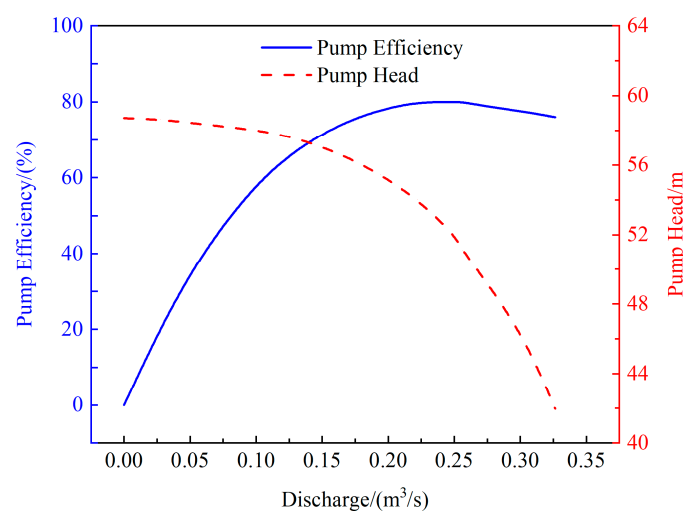


Figure 5. Pump performance curve.

Table 1. Pump parameters.

Design Head Range (m)	Rated Discharge (m³/s)	Rated Speed (r/min)	Moment of Inertia (kg·m²)
170.15–210	0.309	1480	190

Note: The flywheel effect is denoted as GD^2 (unit: $\text{kgf} \cdot \text{m}^2$ or $\text{N} \cdot \text{m}^2$). The rotational moment of inertia I (unit: $\text{kg} \cdot \text{m}^2$) is related to GD^2 by $I = GD^2/4g$, where g is the gravitational acceleration (9.81 m/s^2). In this study, all parameters are converted to standard SI units.

Table 2. Pipeline segmentation and parameters.

Pipe Section	Length (m)	Diameter (m)	Pipe Material	Wave Velocity (m/s)	Friction Factor
1	5	0.5	Steel	1200	0.0226
2	5	0.5	Steel	1200	0.0226
3	5	0.4	Steel	1200	0.0243
4	30	0.4	Steel	1200	0.0243
5	610	0.8	Steel	1200	0.0193
6	815	0.8	Steel	1200	0.0193
7	650	0.8	Steel	1200	0.0193
8	755	0.8	Steel	1200	0.0193
9	140	0.8	Steel	1200	0.0193
10	135	0.8	Steel	1200	0.0193
11	110	0.8	Steel	1200	0.0193
12	100	0.8	Steel	1200	0.0193
13	240	0.8	Steel	1200	0.0193

14	105	0.8	Steel	1200	0.0193
15	5	0.8	Steel	1200	0.0193
16	5	0.8	Steel	1200	0.0193

Note: The pipeline sections are numbered sequentially starting from the suction sump. Section 1 is adjacent to the suction sump, and Section 16 is adjacent to the water outlet. Sections 1–4 belong to the branch pipeline of a single pump; there are 3 parallel pump branches in operation with identical parameters. Sections 5–16 form the combined main pipeline after confluence.

Table 3. Elevations and chainages of key nodes in the water conveyance system.

Node	Elevation (m)	Chainage (m)
Water Inlet (Min. Water Level)	1764.00	0 + 000
Water Inlet (Max. Water Level)	1787.00	0 + 000
Upstream Pump Valve (Min. Water Level)	1768.49	0 + 005
Upstream Pump Valve (Max. Water Level)	1789.49	0 + 005
Pump (Min. Water Level)	1768.49	0 + 010
Pump (Max. Water Level)	1789.49	0 + 010
Downstream Pump Valve (Min. Water Level)	1768.49	0 + 015
Downstream Pump Valve (Max. Water Level)	1789.49	0 + 015
Terminal Valve	1950.00	3 + 710
Water Outlet	1951.00	3 + 715

Note: Min. = Minimum, Max. = Maximum. All water levels in the table refer to the operating water level of the suction sump.

3.2. Calculation Control Conditions

According to relevant national standards and specifications, such as “Standard for pumping station design” (GB 50265-2022) [30] and “Standard for Design of Outdoor Water Supply Engineering” (GB 50013-2018) [31], combined with the layout and operational characteristics of the pumping station’s water conveyance system, the following control conditions for water hammer calculation are determined as the basis for formulating our protection measures:

- (1) At various water levels, the pump outlet’s working valve closes when power is lost to the pump–motor units under various unfavorable conditions. The maximum reverse speed of the pump motor shall not exceed 1.2 times the rated speed, i.e., 1776 r/min, and the duration of exceeding the rated speed shall not exceed 2 min.
- (2) The instantaneous maximum pressure in the pipeline system shall not be greater than 1.3–1.5 times the working pressure. Considering the pipeline layout and pipe material requirements of this project, the maximum pressure is controlled at no more than 1.3 times the working pressure, i.e., 263.66 m.
- (3) The minimum pressure of the water conveyance system should be set according to the importance of the pumping station and annual operation time, and the minimum value should not be lower than −4 m in general.

It should be noted that the pressure limits adopted herein are derived from the design code and the actual working pressure of the system. They serve as hydraulic design constraints for protection scheme optimization. While a dedicated structural verification (e.g., wall thickness, corrosion allowance, fatigue) is recommended at the detailed design stage of a specific project, such verification is outside the scope of this hydraulic transient study.

3.3. Calculation Working Conditions

This project adopts a floating water intake scheme. Affected by the 21 m water level variation in the intake sump, the actual working head of the pumping station changes

significantly with the change in the intake water level. We take the high-level water tank's operating water level of 1951.64 m as the fixed boundary on the outlet side, combined with the extreme operating water levels of the intake sump; thus, two typical head conditions are defined: when the intake sump is at the maximum operating water level of 1787 m, the total water conveyance head of the system is the smallest, i.e., the minimum head condition; when the intake sump is at the minimum operating water level of 1766 m, the total water conveyance head of the system is the largest, i.e., the maximum head condition.

Among the investigated conditions, the maximum-head condition is the most severe for pump power failure, according to engineering experience in water hammer analysis of existing floating pump stations. Under this condition, the initial flow velocity and working pressure in the pipeline are both high. After sudden power failure, the flow velocity in the pump pipeline changes sharply, with an intense fluctuation in the pressure transient caused by flow inertia. Meanwhile, the reverse rotation is higher, driving the potential energy of the pump unit, making it more prone to adverse phenomena such as excessive reverse speed, excessive pipeline positive pressure, and excessively low negative pressure along the line. The hazard level here is significantly higher than that of the minimum head condition, making it the key research condition for analyzing water hammer protection schemes in this project. Four groups of calculation conditions are determined as shown in Table 4.

Table 4. Calculation working conditions.

Condition	Suction Sump Water Level (m)	Outlet Sump Water Level (m)	Number of Operating Pumps	Number of Shutdown Pumps
Case 1.1	1766	1951.64	3	3
Case 1.2	1787	1951.64	3	3
Case 1.3	1766	1951.64	3	1
Case 1.4	1766	1951.64	3	2

4. Results and Discussion

4.1. Analysis of Steady-State Operation Calculation

To analyze in detail the flow capacity and pipeline working pressure of the system under study, we carry out steady-state operation calculation and analysis for two typical conditions, i.e., maximum head and minimum head, with three pumps operating simultaneously. The results are shown in Figure 6.

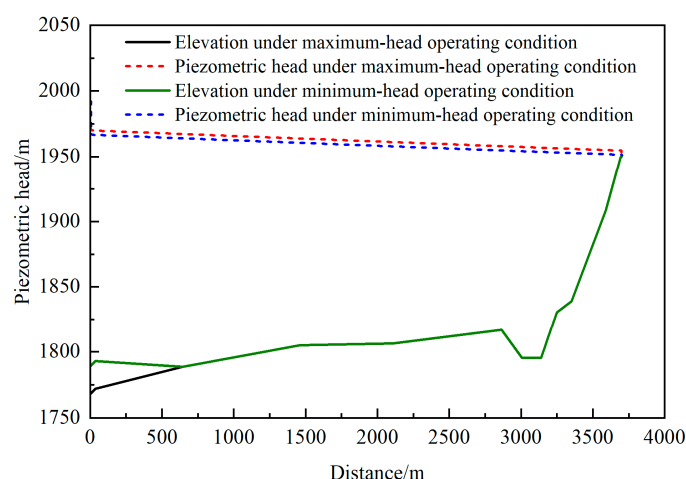


Figure 6. Piezometric head lines for steady-state operation of three pumps.

According to the steady-state calculation results, under the maximum and minimum head conditions, the water conveyance flow rates of the line are $0.926 \text{ m}^3/\text{s}$ and $0.927 \text{ m}^3/\text{s}$, respectively, meeting the water supply requirements of the conveyance system. No negative pressure occurs in the system under both conditions, satisfying the requirements for safe pipeline operation.

4.2. Analysis of Hydraulic Transient Process Under Accidental Pump Shutdown Without Pressure Regulation Measures

We will analyze in detail the impact of pressure and flow fluctuations caused by accidental pump shutdown on the operational safety of the water supply system; to this end, we perform hydraulic transient calculation and analysis for the conditions mentioned in Section 3.3 without pressure regulation measures. The results are shown in Figures 7–9.

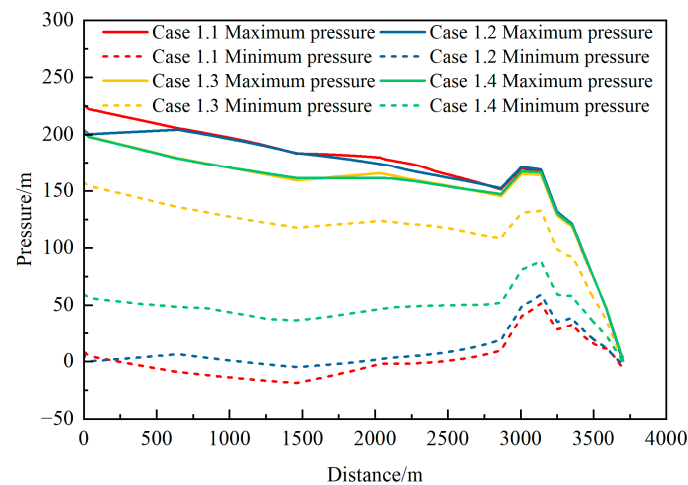


Figure 7. Variation in water pressure in the pipeline under different operating conditions.

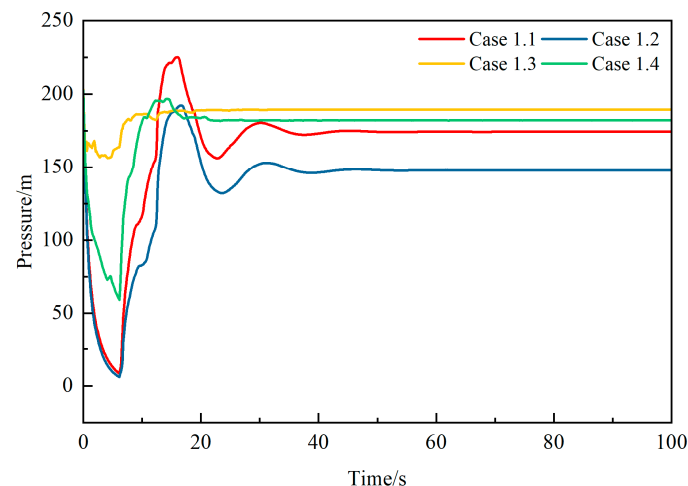


Figure 8. Variation in outlet pressure of faulty pump under different operating conditions.

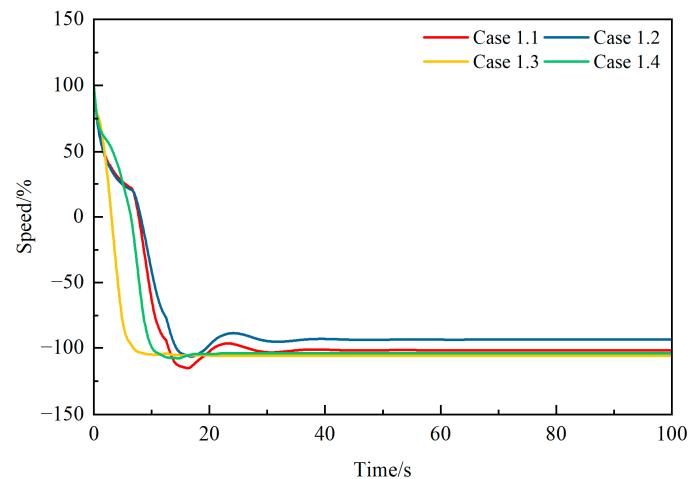


Figure 9. Variation in faulty pump speed under different operating conditions.

As shown in Figure 7, the maximum pressures of the four conditions are all lower than the positive-pressure control upper limit of 263.66 m, and all positive-pressure indicators meet the specification requirements; however, the negative-pressure levels differ significantly. All pressure values in this paper refer to gauge pressure head in meters. For Case 1.1, the maximum pressure along the water conveyance system is 225.13 m, and the minimum is as low as -18.66 m, far below the lower limit of -4 m. This negative-pressure severity is the most prominent among the investigated cases and extends over a long middle section of the pipeline, indicating a potential cavitation and water-column-separation risk. Theoretically, water may vaporize when the local pressure head falls below about -10 m gauge; therefore, the calculated minimum pressure heads in this paper should be interpreted as an indication of negative-pressure severity rather than a direct quantitative prediction of cavitation. This assessment is qualitative and based only on the calculated pressure level. The present model does not include a specialized cavitation or water-column-separation model and cannot resolve phase change, cavity growth or collapse, entrained air, or column rejoining. Accordingly, the water-column-separation discussion is a preliminary qualitative judgment based on pressure—the minimum pressure plummets to -72.77 m, with the negative pressure level seriously exceeding the standard heads below the vaporization pressure head. For Case 1.2, the maximum system pressure is 204.28 m and the minimum is -4.74 m, slightly lower than the control limit, meaning that it also fails to meet safety requirements. For Cases 1.3 and 1.4, the maximum system pressures are 203.14 m and 202.82 m, respectively, and the minimum pressures are -0.10 m and 0.29 m, respectively; both meet the control requirements, and the negative-pressure risk is essentially eliminated.

As shown in Figures 8 and 9, all four groups of conditions present typical transient characteristics of pump-trip water hammer, but the fluctuation amplitude shows obvious differences in the different conditions. After an accidental power failure, the pump loses power input, the outlet head decays rapidly, and the pressure drops to its lowest point within a few seconds; it then peaks under the reciprocating superposition of water hammer waves and gradually converges to a stable point after multiple rounds of oscillation. Correspondingly, the pump speed undergoes forward rotation deceleration, zero-crossing reverse rotation, and reverse acceleration to stabilization. Numerically, Case 1.1 has the largest pressure fluctuation amplitude and the steepest drop in pump outlet pressure, with the corresponding maximum reverse speed reaching 115.20% of the rated speed, the highest among the four groups. The pressure fluctuation and reverse speed of Case 1.2 are both reduced, with a maximum reverse speed of 106.55%. Due to the small number of shutdown pumps in Case 1.3 and Case 1.4, the flow attenuation process in the pipeline is

more gentle; there is no obvious steep drop in pump outlet pressure, and the overall transition to a steady state is smooth. The maximum reverse speeds are 105.98% and 107.88%, respectively. These speeds in the four groups do not exceed the control limit of 1.2 times the rated speed. However, after accidental pump shutdown, the pump will enter a long-term continuous reverse rotation state, which will aggravate the wear of mechanical components and have potential adverse effects on the equipment's structure and service life; a higher reverse speed makes such adverse effects more significant.

In Case 1.1, the simultaneous power failure of multiple parallel pumps triggers the most drastic system-wide changes in flow and pressure among the investigated cases, resulting in extreme pressure drops followed by severe pressure spikes. Because the system operates at maximum head, the fluid must overcome the largest elevation difference, leading to the most intense kinetic-energy conversion during reverse flow and the impact of the water column. Consequently, this condition exhibits the most severe negative pipeline pressure, the highest amplitude of pressure oscillation downstream of the pump, and the fastest pump reverse speed among the four investigated conditions. These compounding factors make its hydraulic transient process significantly more hazardous than that of the other three investigated conditions. It is therefore identified as the most severe among the investigated cases and adopted as the primary baseline for protection design; it requires focused control to achieve water-hammer protection. If the relevant protection measures can simultaneously meet the control requirements for positive pressure, negative pressure, and reverse speed under this extreme condition, the other investigated scenarios are expected to meet safe-operation standards, subject to confirmation by a full envelope analysis. Therefore, the parameter design, effect verification, and comparative analysis of subsequent water-hammer protection schemes use Case 1.1 as the benchmark calculation condition. This ensures applicability and reliability for the investigated operating conditions, but it does not replace a full envelope analysis of all possible off-design and fault combinations.

4.3. Analysis of Hydraulic Transient Process Under Accidental Pump Shutdown Using Different Pump Outlet Valve Closure Rules

When an accidental pump shutdown occurs, simple non-return valves close passively and rapidly upon flow reversal, which tends to induce severe secondary positive water hammer pressures. Additionally, adjustable valves are preferred in actual engineering projects to achieve more flexible and active flow regulation. Therefore, controlled closure strategies (linear and two-stage rules) are utilized. The variation in valve flow resistance coefficient with opening and the time-varying valve opening curves under different closure rules are shown in Figure 10 and Figure 11, respectively.

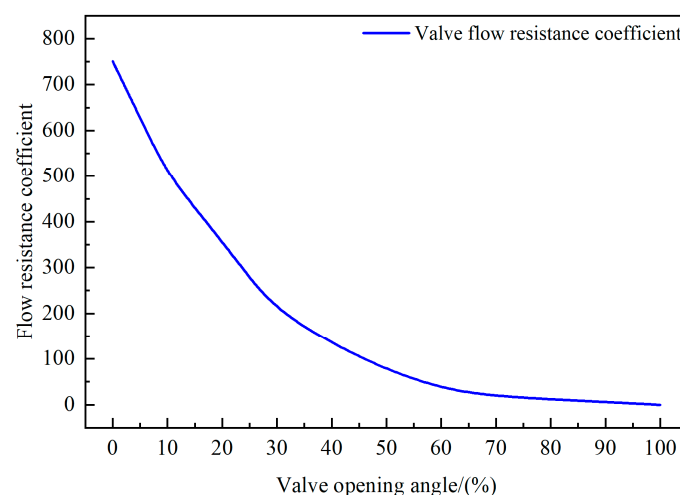


Figure 10. Variation in valve flow resistance coefficient with valve opening angle.

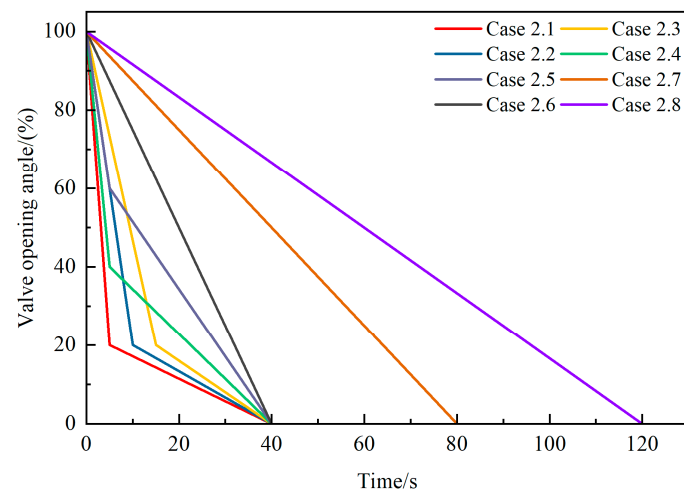


Figure 11. Variation in valve opening angle with time under different working conditions.

In this study, the pump outlet valve is a hydraulically actuated butterfly valve installed on the outlet branch of each pump unit. Its resistance characteristic is defined by the resistance coefficient ζ as a function of the relative opening y ($y = 1$ for fully open and $y = 0$ for fully closed), as listed in Table X and illustrated in Figure 10. The flow coefficient τ in Equations (11) and (12) is related to the resistance coefficient by:

$$\tau(y) = \frac{1}{\sqrt{1 + \zeta(y)}} \quad (23)$$

and the corresponding effective flow area is $CdA(y) = \tau(y)A$, where A is the cross-sectional area of the branch pipe, m^2 .

The two-stage valve closure process depicted in Figure 11 is mathematically defined by a piecewise continuous function $y(t)$. Although a slight mechanical response lag exists in real actuators between signal reception and physical movement, this delay (typically 0.1–0.5 s) is negligible compared to the total closure time. Therefore, the action delay is neglected in the numerical model ($t_{delay} = 0$), which simplifies the boundary conditions and provides a slightly conservative scenario for hydraulic transient analysis. The relative opening $y(t)$ is expressed as:

$$y(t) = \begin{cases} 1.0 - \frac{1.0 - y_{fast}}{t_{fast}} t, & 0 \leq t \leq t_{fast} \\ y_{fast} - \frac{y_{fast}}{t_{slow}} (t - t_{fast}), & t_{fast} < t \leq t_{total} \end{cases} \quad (24)$$

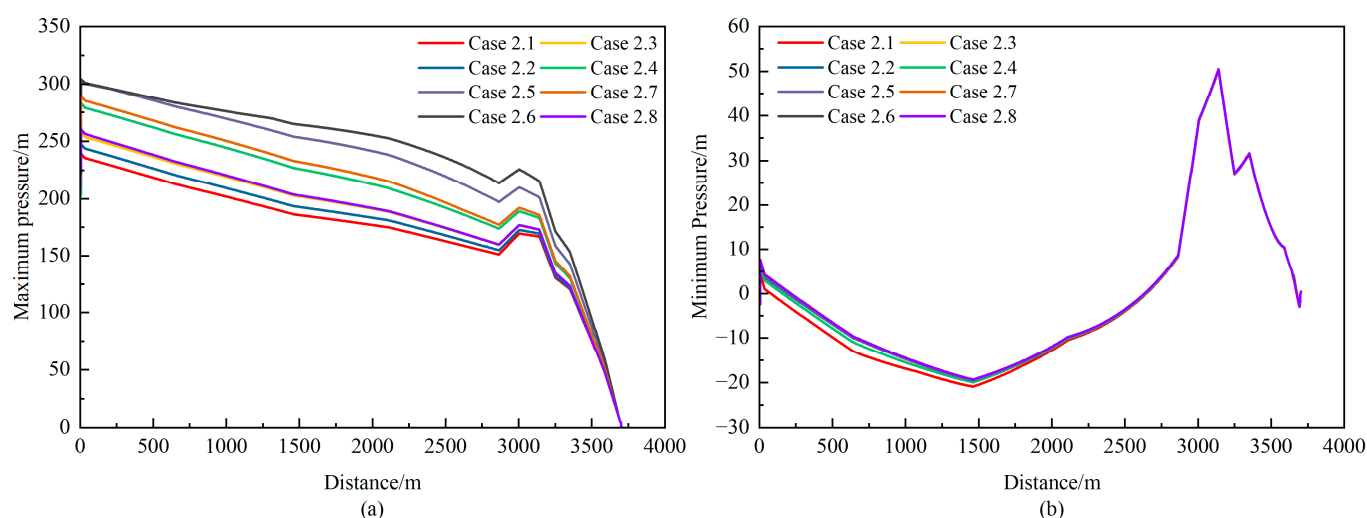
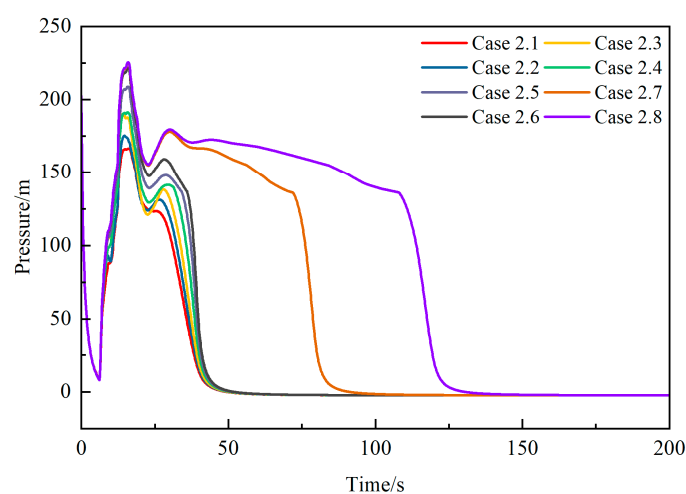
where y_{fast} is the relative opening at the end of the fast closure stage, t_{fast} is the fast closure duration, t_{slow} is the slow closure duration, and the total closure time $t_{total} = t_{fast} + t_{slow}$. Continuity is strictly enforced at $t = t_{fast}$ to avoid sudden flow area jumps in numerical iterations.

Hydraulic transient calculation and analysis are carried out for different pump outlet valve closure rules based on the most unfavorable condition, Case 1.1, with eight groups of conditions. These cover two valve closure strategies—linear closure and two-stage closure—to compare their effects on water hammer pressure protection. The calculation conditions are shown in Table 5, and the results are shown in Figures 12–14.

Table 5. Working conditions and calculation results for different pump outlet valve closure rules.

Condition	Fast Closure Time (s)	Fast Closure Angle (%)	Slow Closure Time (s)	Maximum Pres- sure (m)	Minimum Pres- sure (m)	Maximum Re- verse Speed (%)
Case 2.1	5	80	35	239.42	−20.92	−99.35
Case 2.2	10	80	30	247.64	−19.52	−101.61
Case 2.3	15	80	25	257.52	−19.44	−105.51
Case 2.4	5	60	35	283.83	−19.95	−106.37
Case 2.5	5	40	35	303.98	−19.52	−111.10
Case 2.6	40 s linear closure			304.38	−19.35	−144.35
Case 2.7	80 s linear closure			289.20	−19.34	−115.13
Case 2.8	120 s linear closure			260.37	−19.33	−115.29

Note: The negative sign of the maximum reverse speed indicates reverse rotation direction; the percentage value represents the ratio of the absolute value of rotational speed to the rated rotational speed.

**Figure 12.** Variation in water pressure in the pipeline under different valve closure rules: (a) maximum pressure; (b) minimum pressure.**Figure 13.** Variation in faulty pump outlet pressure under different valve closure rules.

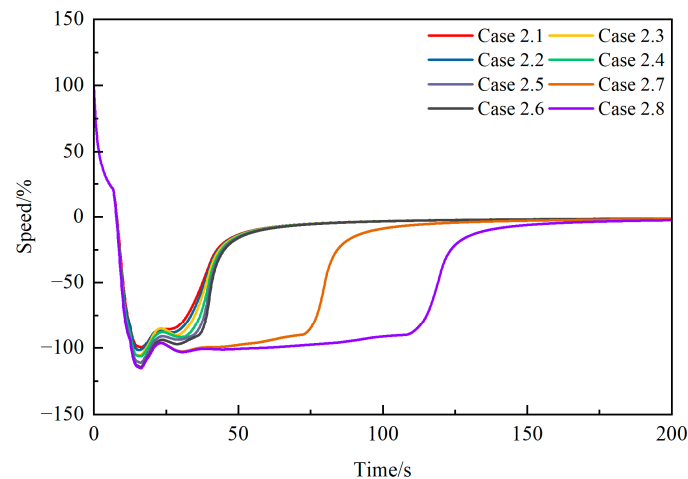


Figure 14. Variation in faulty pump speed under different valve closure rules.

As shown in Table 5 and Figure 12, the two-stage valve closure groups all adopt combined fast closure + slow closure, and different closure processes are formed by adjusting the fast closure time and angle. When the latter remains unchanged at 80%, as the fast closure time extends from 5 s to 15 s, the maximum system pressure gradually increases from 239.42 m to 257.52 m, the absolute value of the maximum reverse speed increases from 99.35% to 105.51%, and the minimum pipeline pressure rises slightly from −20.92 m to −19.44 m. When the time remains unchanged by 5 s, as the angle decreases from 80% to 40%, the maximum system pressure increases significantly from 239.42 m to 303.98 m, the absolute value of the maximum reverse speed increases from 99.35% to 111.10%, and the minimum pipeline pressure rises from −20.92 m to −19.52 m. Considering the pressure distribution law along the line, the pressure variation trends of all two-stage valve closure conditions are consistent: the maximum pressure is concentrated at the start of the pipeline, and the minimum pressure occurs in the middle. Considering the speed variation, a higher proportion and rate of fast closure lead to a faster drop in the pump speed and a lower reverse rotation peak, indicating a stronger inhibitory effect on pump reverse rotation. On the whole, the fast closure stage of two-stage valve closure can quickly cut off most of the flow area and inhibit water backflow. Therefore, within a certain range, the shorter the fast closure time and the larger the fast closure angle, the more significant the inhibition effect on pump reverse rotation and the lower the maximum system pressure; however, this will aggravate the pipeline negative pressure to a certain extent. The reason for this phenomenon is that rapid fast-stage closure sharply interrupts water replenishment from the pump side to the downstream pipeline. Driven by water column inertia in the long-distance pipeline, downstream flow continues towards the downstream high-level outlet sump at the initial velocity, creating a flow deficit in the middle and rear sections and triggering a rapid local pressure drop, which further exacerbates negative pressure.

The three groups involving linear valve closure adopt a uniform closure mode, with total durations of 40 s, 80 s and 120 s respectively. Increasing the total valve closure duration decreases the maximum system pressure gradually from 304.38 m to 260.37 m, and the absolute value of the maximum reverse speed first drops rapidly from 144.35% to 115.13%, and then remains stable with further extension of the valve closure time. The minimum pipeline pressure is only slightly affected by the closure duration, maintained at about −19.3 m in all groups, similar to the negative pressure level under two-stage valve closure. Among these, the maximum reverse speed of the 40 s linear closure condition far exceeds the control limit of 120% rated speed, failing equipment safety requirements. After the duration is extended to more than 80 s, the reverse speed meets the standard, but

the valve closure process takes significantly longer, meaning it cannot quickly block water backflow in the early stages of an accident, making its timeliness weaker than that of two-stage valve closure.

An overall comparison of the valve closure strategies shows the superior comprehensive protection effect of two-stage valve closure. With similar total valve closure durations, the maximum pressure and reverse speed of two-stage valve closure are significantly lower than those of linear valve closure. Moreover, the fast closure stage can cut off most of the valve's opening in a short amount of time, quickly inhibiting the continuous reverse rotation of the pump and water backflow in the pipeline, and has stronger emergency response capability. Overall, Case 2.1 is the best-performing among the investigated valve-closure cases: the absolute value of the maximum reverse speed is 99.35%, the lowest among the eight groups, which can effectively prevent the pump from experiencing reverse rotation for a long time. However, compared with the unprotected condition, after adopting this two-stage valve closure scheme, the maximum system pressure increases from 225.13 m to 239.42 m, and the minimum pipeline pressure decreases from −18.66 m to −20.92 m. This indicates that although optimizing pump outlet valve closure alone solves the problem of long-term pump reverse rotation, it will slightly increase the amplitude of rising pressure under water hammer in the pipeline and aggravate the pipeline's negative pressure; it thus still cannot meet the control requirement of a minimum pressure no lower than −4 m. Therefore, it is necessary to further combine other water hammer protection measures to jointly suppress pipeline negative pressure and reduce the level at which water hammer causes pressure to rise in order to meet the control standards of the water conveyance system.

Regarding the engineering feasibility of synchronizing the three pump outlet valves, the synchronization error is constrained by the centralized PLC system and electro-hydraulic actuators to within tens of milliseconds (e.g., <50 ms), accounting for less than 1% of the 5 s fast-closure stage. During this stage, the transient response is dominated by the substantial rotational inertia of the pump–motor units, making the influence of such minor timing deviations negligible, while the subsequent 35 s slow-closure stage is sufficiently gentle for the long-distance pipeline to act as a natural low-pass filter, attenuating high-frequency disturbances and preventing water hammer superposition. Crucially, the numerical model assumes perfectly synchronized closure ($t_{\text{delay}} = 0$), which represents the most conservative bounding scenario by maximizing wave superposition potential. In actual engineering, minor synchronization discrepancies inherently introduce a staggered effect that disperses flow cut-off and marginally reduces extreme pressure peaks. Consequently, the current perfectly synchronized model provides a safe and conservative upper-bound envelope for engineering design.

4.4. Analysis of Hydraulic Transient Process Under Accidental Pump Shutdown with Combined Protection of Valve Closure Strategy and Air Vessel

To develop an effective water hammer protection strategy, the specific protective effects of the valve, air vessel, and air valve within the combined system were first quantitatively analyzed. The pump outlet valve adopts a two-stage closure rule involving a 5 s fast closure time, an 80% fast closure angle, a 35 s slow closure time, and a 40 s total duration. The air vessel is installed at pipeline stake 1 + 470 m (elevation 1805.00 m), and the terminal air valve is arranged at stake 3 + 705 m (elevation 1951.2 m, air intake diameter 300 mm). Based on this setup, a quantitative attribution analysis was conducted by comparing the hydraulic transient characteristics of four combinations: valve-only, valve + air vessel, valve + air valve, and the complete combined system. The comparison results are presented in Table 6.

Table 6. Comparative analysis of hydraulic transient characteristics under different protection combinations.

Protection Scheme	Maximum Pressure (m)	Minimum Pressure (m)	Location of Min Pressure	Maximum Reverse Speed (%)
Valve	239.42	−20.92	1 + 460	−99.35
Valve + Air Vessel	244.56	−4.78	3 + 695	−93.67
Valve + Air Valve	243.30	−20.92	1 + 460	−99.73
Valve + Air Vessel + Air Valve	245.924	−2.48	0 + 015	−93.86

As shown in Table 7, when valve-only control is applied, although the maximum reverse speed is effectively suppressed to −99.35%, an extreme negative pressure of −20.92 m occurs at the mid-pipeline (stake 1 + 460), indicating a significant potential cavitation risk. With the valve + air vessel scheme, the mid-section negative pressure is significantly mitigated (minimum pressure rises to −4.78 m). However, due to the lack of terminal protection, the negative pressure risk remains. When the valve + air valve scheme is adopted, its mid-section negative pressure (−20.92 m) is identical to that of the valve-only case, and the suppression of reverse speed is the weakest (−99.73%), demonstrating the highly localized nature of the air valve, which fails to simultaneously address mid-section negative pressure and rotational speed control.

In contrast, the complete combined system (valve + air vessel + air valve) exhibits significant synergistic advantages. This scheme not only completely eliminates potential cavitation risks across the entire pipeline (minimum pressure of −2.48 m) but also controls the maximum reverse speed within a safe range (−93.86%). The above attribution analysis indicates that single or partial combinations possess inherent limitations in simultaneously ensuring full-line safety, thereby validating the effectiveness and necessity of this multi-measure synergistic protection system.

Given the demonstrated superiority of this complete combined strategy among the investigated schemes, further hydraulic transient calculations were performed to optimize the air vessel parameters. Seven comparison conditions are established by adjusting the air vessel's total volume, connecting pipe diameter, and water–air ratio (the ratio of initial water volume to initial air volume), with the initial gas pressure set to standard atmospheric pressure [32,33]. The calculation results are shown in Table 7 and Figures 15–18.

Table 7. Working conditions and calculation results for different air vessel parameters.

Condition	Total Volume (m ³)	Connecting Pipe Diameter (m)	Water–Air Ratio	Maximum Pressure (m)	Minimum Pressure (m)	Maximum Reverse Speed (%)
Case 3.1	15	0.5	1:1	244.98	−28.16	−97.57
Case 3.2	15	0.3	1:1	245.92	−2.48	−93.86
Case 3.3	15	0.1	1:1	234.48	−5.41	−94.92
Case 3.4	10	0.3	1:1	269.13	−28.25	−97.61
Case 3.5	20	0.3	1:1	244.98	−2.48	−88.41
Case 3.6	15	0.3	2:3	235.66	−72.77	−94.43
Case 3.7	15	0.3	3:2	251.39	−2.48	−98.80

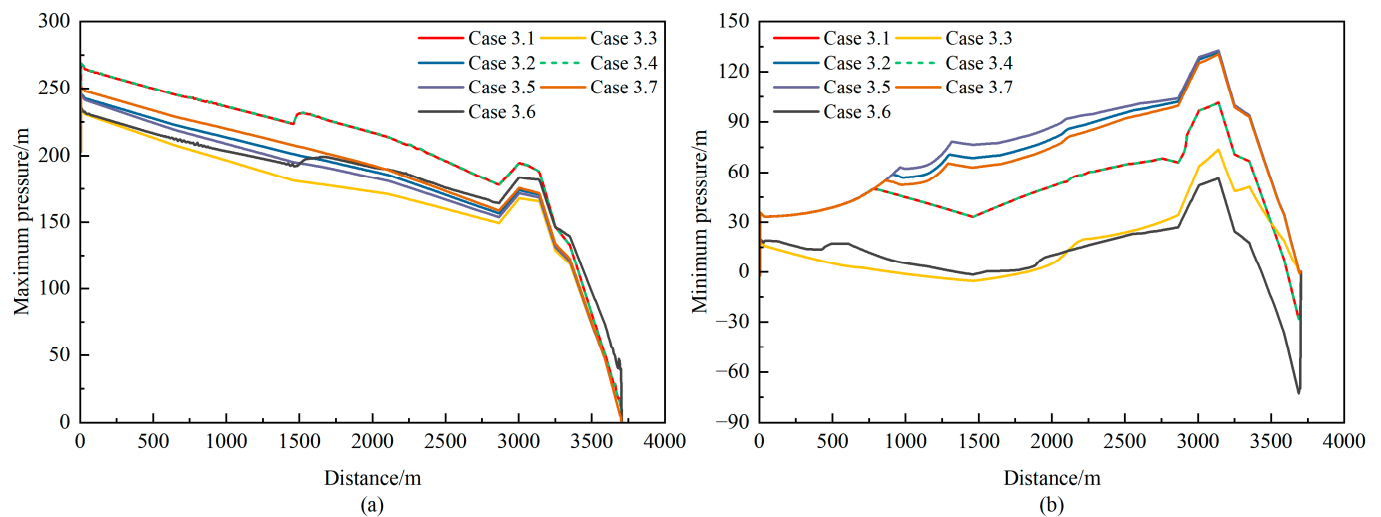


Figure 15. Variation in water pressure in pipeline under different air vessel parameters: (a) maximum pressure; (b) minimum pressure.

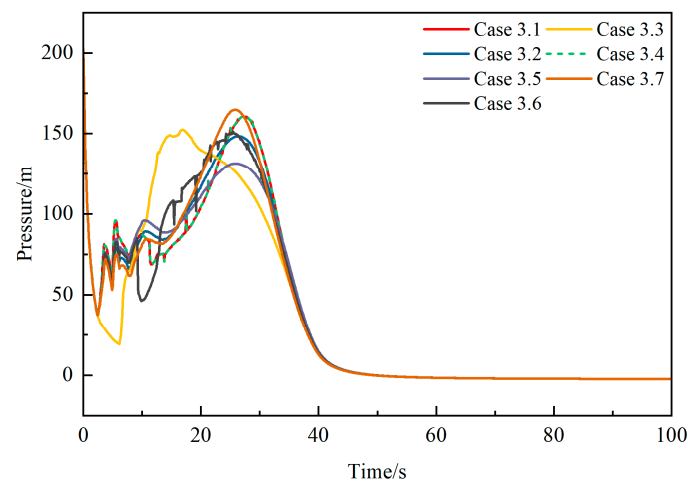


Figure 16. Variation in faulty pump outlet pressure under different air vessel parameters.

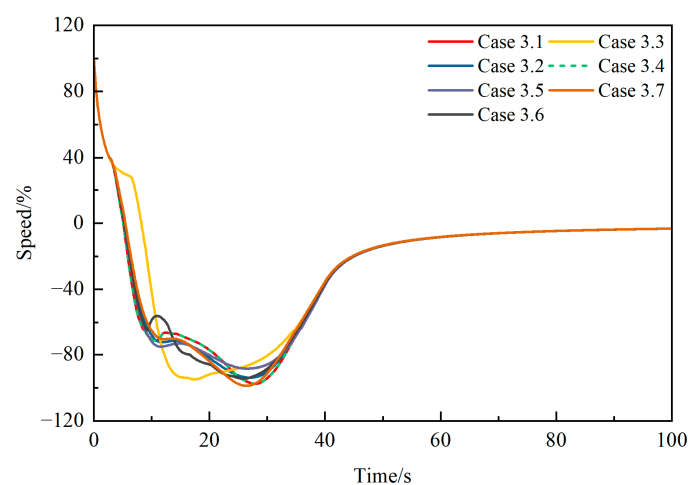


Figure 17. Variation in faulty pump speed under different air vessel parameters.

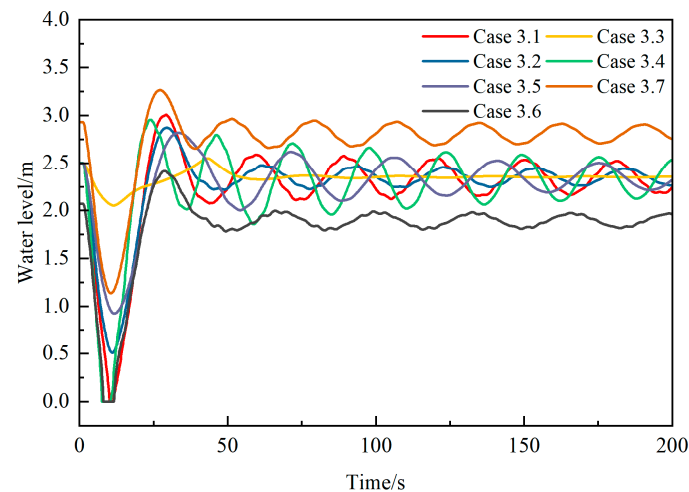


Figure 18. Variation in water level in the air vessel under different air vessel parameters.

As shown in Table 7 and Figure 15, air vessel parameters have significant effects on both maximum and minimum pressures, and the regulation effects of different parameters are notably different. With a tank volume of 15 m³ and a water–air ratio of 1:1, as the connecting pipe diameter decreases from 0.5 m to 0.3 m, the maximum pressure of the water conveyance system increases slightly from 244.98 m to 245.92 m, while the minimum pipeline pressure rises significantly from −28.16 m to −2.48 m, with a significantly improved effect of suppressing negative pressure. When the connecting pipe diameter further decreases to 0.1 m, the maximum pressure of the water conveyance system drops to 234.48 m, but the minimum pressure falls back to −5.41 m, with a weakened effect of negative pressure control. The above phenomenon is determined by the flow resistance and water replenishment capacity of the connecting pipe: an excessively large pipe diameter results in low flow resistance, causing the stored water in the air vessel to discharge rapidly at the initial stage of pump shutdown and even leading to tank emptying, which makes the vessel lose continuous pressure regulation capacity; an excessively small pipe diameter brings large throttling resistance, limiting the water replenishment rate to the pipeline and weakening the negative pressure suppression effect. Among the three analyzed diameters, 0.3 m achieves the best comprehensive control effect. With a connecting pipe diameter of 0.3 m and a water–air ratio of 1:1, as the tank volume increases from 10 m³ to 20 m³, the maximum pressure of the water conveyance system gradually decreases from 269.13 m to 244.98 m, and the minimum pressure rises from −28.25 m to −2.48 m. This indicates that within the analyzed volume range, increasing the tank volume helps improve the suppression effect on both positive and negative pressure heads, but the improvement marginal benefit gradually decreases as the volume increases. With a tank volume of 15 m³ and a connecting pipe diameter of 0.3 m, when the water–air ratio is adjusted from 1:1 to 2:3, the proportion of air in the tank increases, the maximum pressure of the water conveyance system drops to 235.66 m, and the minimum pressure reaches −72.77 m. This indicates severe indicative negative-pressure severity and potential cavitation risk. When the water–air ratio is adjusted to 3:2, the maximum pressure of the water conveyance system rises to 251.39 m, and the minimum pressure remains at −2.48 m, with the negative pressure control effect consistent with that of the 1:1 water–air ratio. It can be seen from the pressure envelope along the line that the air vessel has the most prominent suppression effect on pressure fluctuations near its installation position, and the terminal air valve effectively improves the negative pressure level in the rear section of the pipeline. Their cooperation realizes pressure regulation of the whole pipeline.

The combined protection scheme has a significant inhibitory effect on pump reverse rotation, and the water level change in the air vessel can intuitively reflect its working process of water replenishment and pressure regulation. The maximum reverse speeds under all conditions are lower than 100% of the rated speed, far better than the control limit of 120%. Among these conditions, increasing the tank volume can effectively reduce the peak reverse speed; the maximum reverse speed of Case 3.5 is only 88.41%, the lowest among all conditions. Deviation of the connecting pipe diameter from the reasonable range and increasing the water–air ratio will both lead to a slight increase in reverse speed and weaken the inhibitory effect on pump reverse rotation. It can be seen from the time–history curve of the air vessel’s water level that the pressure in the pipeline drops rapidly in the initial stage of accidental pump shutdown, and the water in the air vessel flows into the main pipeline quickly to replenish flow, causing the water level in the vessel to drop rapidly in a short time. Among the conditions studied here, the water levels in the vessels of Case 3.1, Case 3.4 and Case 3.6 dropped directly to 0 m in the initial stage of the accident, showing tank emptying. The main reasons for this phenomenon are as follows: for Case 3.1, the connecting pipe diameter is 0.5 m, the resistance of water inflow and outflow is too small, and a large amount of water in the vessel flows out in a short time; for Case 3.4, the tank volume is only 10 m³, the effective water storage is insufficient, and it is quickly exhausted after rapid water replenishment; for Case 3.6, the water–air ratio is 2:3, the initial water proportion in the vessel is low, and the available water for replenishment is limited. After the tank is emptied, the air vessel loses its capacity for continuous water replenishment and pressure regulation, and the pipeline negative pressure of the corresponding conditions deteriorates significantly, with a greatly reduced protection effect. The air vessels under other conditions do not show emptying. After the water level drops to the bottom, it presents periodic oscillation attenuation under the reciprocating action of positive and negative water hammer waves. The smaller the connecting pipe diameter, the greater the resistance of water flowing into and out of the vessel, and the lower the frequency and amplitude of water level oscillation. The larger the tank volume, the more sufficient the water storage in the vessel, the gentler the water level fluctuation process, and the stronger the continuous water replenishment capacity. In practical engineering operation, the air vessel should not be completely emptied; otherwise, air may enter the main pipeline and affect the normal operation of the water conveyance system. Therefore, Case 3.1, Case 3.4 and Case 3.6 are not feasible in engineering practice and are excluded from the comparison of recommended schemes.

Case 3.2 is selected as the best-performing among the seven investigated options, on the premise that a larger tank volume is economically unjustified. Admittedly, Case 3.5 (20 m³ volume) provides superior hydraulic parameters across the three main criteria: it yields a maximum pressure of 244.98 m, a minimum pressure of −2.48 m, and a maximum reverse speed of 88.41%, compared to 245.92 m, −2.48 m, and 93.86% for Case 3.2, respectively. However, the 15 m³ tank in Case 3.2 still ensures that all three core control indicators fully meet their respective safety standards. Its maximum system pressure remains well below the positive control limit of 263.66 m, its minimum pressure satisfies the negative pressure requirement, and its reverse speed complies with safe equipment operation limits. Therefore, it is selected as the best-performing among the investigated cases to balance hydraulic safety without incurring the presumed, albeit uncalculated, additional costs of a larger tank. Furthermore, the 15 m³ volume and 0.3 m connecting pipe diameter fall within a reasonable engineering range, ensuring controllable water level oscillations and long-term operational stability. Ultimately, this scheme forms a robust multi-level water hammer protection system—using a valve closure strategy to suppress pump reverse rotation, an air vessel to mitigate middle-section pressure fluctuations, and an air

valve to alleviate terminal negative pressure—providing reliable safety guarantees for actual engineering operations.

5. Conclusions

In this paper, we aimed to address the prominent water hammer risk of accidental pump shutdown in water conveyance projects involving floating pump stations experiencing large water level fluctuations, as well as the difficulty of establishing a single protection measure, taking into account multi-dimensional management and control. Toward this aim, we established a numerical model of hydraulic transient processes based on the method of characteristics and systematically studied the mechanisms of water hammer regulation, as well as performed parameter optimization, to create a combined protection strategy consisting of valve closure rules and air vessels. The main conclusions are as follows:

- (1) Among the investigated cases, the simultaneous shutdown of three pumps at the minimum operating water level is the most severe condition. Without protection, the minimum pressure in the middle of the pipeline is as low as -18.66 m, far below the safety limit. This indicates pronounced negative-pressure severity and a potential cavitation/water-column-separation risk. The maximum reverse speed of the pump reaches 115.20% of the rated speed. Although this meets the control requirements, long-term continuous reverse rotation will aggravate equipment wear.
- (2) Optimizing valve closure rules can effectively suppress pump reverse rotation and positive pressure rise, and the comprehensive protective effect of two-stage valve closure is significantly better than that of linear valve closure. With the scheme involving a 5 s fast closure time + 80% fast closure angle + 35 s slow closure time, the maximum reverse speed can be reduced to 99.35% of the rated speed, but it will simultaneously aggravate pipeline negative pressure. Single valve regulation cannot meet the negative pressure safety requirements.
- (3) The combined protection provided by valve closure rules, air vessels, and air valves has a significant synergistic effect. Two-stage valve closure effectively suppresses the risks of water backflow and pump reverse rotation, and air vessels improve positive and negative pressure fluctuations through bidirectional regulation. Comparative analysis of multiple parameters reveals that while a 20 m^3 air vessel yields superior hydraulic performance, the scheme utilizing an air vessel with a volume of 15 m^3 , a connecting pipe diameter of 0.3 m and a water–air ratio of 1:1, matched with the above two-stage valve closure and a terminal air valve, is the best-performing among the investigated cases, assuming that a larger volume is economically unjustified. In this case, the maximum pipeline pressure is 245.92 m, the minimum pressure is -2.48 m, and the maximum reverse speed is 93.86% of the rated speed. All three indicators meet the specification requirements.

This best-performing scheme among the investigated cases relies on the specific conditions of this project (a 21 m water level fluctuation, simultaneous shutdown of three pumps, and a specific pipeline layout); therefore, its parameters should be verified before it is applied to other projects. In similar engineering projects, the connecting pipe diameter of the air vessel should not be excessively large; otherwise, the rapid discharge of water during the initial stage of pump shutdown may empty the tank, eliminating its pressure-regulation capacity and severely worsening negative pressure in the pipeline. Additionally, the air vessel volume and water–air ratio must be matched to the pipe diameter to balance hydraulic safety and economy. Furthermore, meeting numerical criteria does not by itself guarantee the operational safety of the actual water conveyance system. The current model simplifies practical complexities, omitting pipeline strength, real valve and air

tank characteristics, electromechanical pump dynamics, and parameter uncertainties. Therefore, while these research findings can provide valuable technical support for similar floating water intake systems, future engineering implementations must thoroughly evaluate these potential real-world uncertainties so that system safety is assessed as comprehensively as practicable. Regarding cavitation, the related conclusions are limited by the single-phase model. The calculated negative pressures indicate negative-pressure severity and potential cavitation risk only; they do not quantify vapor-cavity growth, collapse, entrained air, or column rejoining. Future studies should therefore implement a suitable cavitation or gas-cavity model to verify these qualitative indications.

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References

1. Ouyang, C.; Zhao, Z.; Liu, C.; Zhao, Z.; Yin, X.; Yang, J. Does the upstream gate control scheme threaten the safety of extra-long pressurized water diversion tunnel: Gas–liquid evolution characteristics of the filling process. *Phys. Fluids* **2024**, *36*, 094130.
2. Lohrasbi, A.R.; Attarnejad, R. Water hammer analysis by characteristic method. *Am. J. Eng. Appl. Sci.* **2008**, *1*, 287–294.
3. Xue, Y.; Wang, C.; Zhang, Y.; Du, S. Analysis on Water Hammer Resistant Characteristics of Pipeline Valves. *Curr. Sci.* **2025**, *5*, 5152–5164.
4. Lescovich, J.E. The control of water hammer by automatic valves. *J.-Am. Water Works Assoc.* **1967**, *59*, 632–644.
5. Wang, M.; Liu, Y. Research on two-way flow passage self-irrigation and self-draining of gate open control based on CFX. *China Rural Water Hydropower* **2017**, *7*, 140–143.
6. Romuald, B.; Jian, Z.; Dong, Y.X.; Claire, D. Assessment and performance evaluation of water hammer in hydroelectric plants with hydropneumatic tank and pressure regulating valve. *J. Press. Vessel Technol.* **2021**, *143*, 041401.
7. Liu, W.X.; Tan, P.; Yang, J.G. Study on two-stage closure of valve stroking for water hammer protection of unexpected pump-stop. *Fluid Mach* **2020**, *48*, 53–57.
8. Lu, M.; Liu, X.; Xu, G.; Tian, Y. Optimal pump–valve coupling operation strategy of complex long-distance water-conveyance systems based on MOC. *Ain Shams Eng. J.* **2024**, *15*, 102318.
9. Wan, W.; Li, F. Sensitivity analysis of operational time differences for a pump–valve system on a water hammer response. *J. Press. Vessel Technol.* **2016**, *138*, 011303.
10. Han, Y.; Shi, W.; Xu, H.; Wang, J.; Zhou, L. Effects of closing times and laws on water hammer in a ball valve pipeline. *Water* **2022**, *14*, 1497.
11. Toumi, A.; Sekiou, F. Optimal valve closing law for improved water hammer control: A case from a water supply pipeline in Guelma, Algeria. *AQUA—Water Infrastruct. Ecosyst. Soc.* **2024**, *73*, 200–216.

12. Stephenson, D. Simple guide for design of air vessels for water hammer protection of pumping lines. *J. Hydraul. Eng.* **2002**, *128*, 792–797.
13. Ramalingam, D.; Lingireddy, S. Neural network–derived heuristic framework for sizing surge vessels. *J. Water Resour. Plan. Manag.* **2014**, *140*, 678–692.
14. Bergant, A.; Kruisbrink, A.; Arregui, F. Dynamic Behaviour of Air Valves in a Large-Scale Pipeline Apparatus. *J. Mech. Eng./Stroj. Vestn.* **2012**, *58*, 225.
15. Mahmoudi-Rad, M.; Najafzadeh, M. Effects of surge tank geometry on the water hammer phenomenon: Numerical investigation. *Sustainability* **2023**, *15*, 2312.
16. Bettaieb, N.; Taieb, E.H. Assessment of failure modes caused by water hammer and investigation of convenient control measures. *J. Pipeline Syst. Eng. Pract.* **2020**, *11*, 04020006.
17. Ali, N.A.; Abozeid, G.; Darweesh, M.S. Studying of water hammer in water supply pipes network with different abnormal operation conditions (case study-Assiut city network). *J. Eng. Sci.* **2013**, *41*, 2036–2054.
18. Khokhlov, A.V.; Khokhlov, V.A.; Titova, Z.O. Technical Means of Protecting Pumping Station Equipment from a Water Hammer. *Power Technol. Eng.* **2019**, *53*, 56–61.
19. Himr, D. Computation of water hammer protection of modernized pumping station. *EPJ Web Conf.* **2014**, *67*, 02035.
20. Lyu, J.; Zhang, J.; Wang, X.; Xu, T. A combined water hammer protective method for optimizing the volume of the air vessel in water supply systems. *AQUA—Water Infrastruct. Ecosyst. Soc.* **2021**, *70*, 1217–1230.
21. Chen, X.; Zhang, J.; Yu, X.; Chen, S.; Shi, L. Study on joint protection of air tank and air valve in long-distance water supply system. *J. Press. Vessel Technol.* **2022**, *144*, 061701.
22. Hamed, A.M.; Abdulaziz, A.M. Experimental and numerical optimization of pressurized air vessel performance for water hammer mitigation. *Sci. Rep.* **2025**, *15*, 39575.
23. Ilić, J.; Božić, I.; Petković, A.; Karadžić, U. A Novel Approach to the Improvement of the Hydropower Plants Protective Measures—Modelling and Numerical Analyses of the Semi-Pneumatic Surge Tank. *Appl. Sci.* **2022**, *12*, 2353.
24. Niu, F.; Xu, R.; Wang, L.; Ma, Y. *Mobile Intake Pump Station Engineering*; Yellow River Water Conservancy Press: Zhengzhou, China, 2001; Volume 10. (In Chinese)
25. Wylie, E.B.; Streeter, V.L.; Suo, L. *Fluid Transients in Systems*; Prentice Hall: Englewood Cliffs, NJ, USA, 1993.
26. Akpan, P.U.; Jones, S.; Eke, M.N.; Yeung, H. Modelling and transient simulation of water flow in pipelines using WANDA Transient software. *Ain Shams Eng. J.* **2017**, *8*, 457–466.
27. Chen, X.; Zhang, J.; Yu, X. Thermodynamic behaviour and heat transfer characteristics of air vessels in water delivery systems. *Appl. Therm. Eng.* **2024**, *236*, 121428.
28. Tasca, E.; Besharat, M.; Ramos, H.M.; Luvizotto, E., Jr.; Karney, B. Exploring the sensitivity of the transient response following power failure to air valve and pipeline characteristics. *Water* **2023**, *15*, 3476.
29. Zhai, X. Research on Hydraulic Performance of a New Type Air Vessel in Long-Distance Pressure Water Transfer System. Master's Thesis, North China University of Water Resources and Electric Power: Zhengzhou, China, 2020. (In Chinese)
30. GB 50265-2022; Standard for Pumping Station Design. China Planning Press: Beijing, China, 2022.
31. GB 50013-2018; Standard for Design of Outdoor Water Supply Engineering. China Planning Press: Beijing, China, 2018.
32. Besharat, M.; Rabbani, A.; Yang, X.; Martinez, J.S.; Moran, V.; Evans, B.; Ramos, H. Data-driven predictive analysis and visualisation of air–water dynamics in an air vessel. *J. Hydroinformatics* **2025**, *27*, 787–804.
33. Shi, X.; Mu, Z.; Hu, D.; Song, Y.; Zhou, Z. Air vessel–one-way surge tower hybrid protection strategy with dynamic valve control for water hammer mitigation in long-distance pumped flow systems. *AQUA—Water Infrastruct. Ecosyst. Soc.* **2025**, *74*, 626–646.

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