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NUMERICAL STUDY ON THE EFFECTS OF AIR VALVE CHARACTERISTICS ON PRESSURE SURGES DURING PUMP TRIP IN PUMPING SYSTEMS WITH AIR ENTRAINMENT

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SUMMARY

An improved numerical model and computational method are proposed here for the study of the effects of air valve characteristics on pressure surges during pump trip in fluid systems with air entrainment. Numerical experiments have shown that air valves with higher inflow characteristics installed at peak locations of a fluid system may reduce the magnitude of the extreme negative pressure surges. However, for near zero air entrainment levels in the fluid system, air valves with higher outflow characteristics tend to result in higher positive pressure surges. The results of the numerical simulations seem to be in qualitative agreement with available field observations [T.S. Lee and H.F. Cheong, *Teck Hock Pumping Station: Flow Rate and Pressure Transient Measurements*, Johnson Pacific, Singapore, 1994]. Copyright © 1999 John Wiley & Sons, Ltd.

KEY WORDS: air valve; pressure surges; air entrainment; pump trip

1. INTRODUCTION

Air valves are usually installed at the high points in a fluid system to prevent buckling of the pipeline due to the extreme reduction in the magnitude of negative pressure surges during a pump trip. In the typical day-to-day operations of a pumping system, pipelines at high points are also frequently subjected to subatmospheric pressure conditions that may cause the pipelines to buckle if unprotected. To protect pipelines from this problem, air valves may be installed at high points along the pipeline system. When the line pressure at the air valve drops below atmospheric pressure, the air valve opens to admit air. This prevents a further drop in the pressure in the pipeline. During a subsequent rise in line pressure to above the atmospheric pressure, the admitted air is allowed to escape in a controlled manner. However, the rapid expulsion of air from an air valve may lead to the complete exhaustion of air within a short duration. This will in turn result in dangerous pressure transients in the fluid system due to the severe flow deceleration that occurs when the water strikes the air valve while moving at the same rate as the exhausting air.

In a series of studies undertaken at Colorado State University on air release valve designed to remove air from pipelines, Kolp and Andrews (see Tullis [1]) revealed in the studies that the

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use of very small air release valves should aid in the reduction of any pressure rise due to the exhaustion of air under pressure. The study also showed that the use of large automatic air release valves may result in severe undesirable occurrences. One of these possibilities—for certain pipeline configurations and flow rates—may be the rapid closure of air valves as the final amount of air is exhausted, causing serious pressure rise due to rapid deceleration of the flow in the pipeline. Field measurements by Lee and Cheong [2] also showed similar phenomena.

In another study by Campbell [3] on the Wasis Scheme for an additional supply of water to Riyadh, it was concluded that non-return air valves are capable of cushioning slams that result from negative surges, and thereby allow the omission or substantial reduction of surge vessels or other surge protection devices. The study considered non-return air valves as an important and cheap method of surge protection, which can save costs at the same time as increasing the protection of the line. Campbell proposed that the use of non-return air valves should be as common as the use of car seat-belts and for the same reason, they both protect against the unexpected.

2. FLUID TRANSIENT MODEL

In this present study, a numerical method is proposed and numerical experiments were conducted to investigate the effects of air valve characteristics on the pressure transients during a pumping trip in a pumping station with air entrainment. The present study improves the previously used numerical models and computational procedures developed by Pearsall [4], Provoost [5] and Lee [6–8]. The present study also includes an improved computational procedure for the absorption of free gases and the evolution of dissolved gases in the transported fluid [9] and cavitation at vapour pressure under transient conditions [10,11]. Detailed description of the air entrainment model and the method of characteristics for the transient flow with air entrainment are given in [12] and will not be repeated here.

Figure 1 shows a common flow arrangement in fluid engineering, such as water, sewerage, etc., consisting of (i) a lower reservoir, (ii) a group of pumps with a check valve in each branch, and (iii) a pipeline discharging into an upper reservoir (water tower, gravity conduit, aeration well, etc.). In order to safeguard the pipeline and its hydraulic components from over- and/or underpressurisation, it is important to determine extreme pressure loads under transient conditions. Pump stoppage is an operational case that has to be investigated and which often gives rise to maximum and minimum pressures. The most severe case occurs when all the pumps in a station fail simultaneously owing to a power failure. In this case, the flow in the pipeline rapidly diminishes to zero and then reverses. The pump also rapidly loses its forward rotation and reverses shortly after the reversal of the flow. As the pump speed increases in the reverse direction, it causes great resistance to the back flow, which produces high pressure in the discharge line near the pump. To prevent reverse flow through the pump, a check valve is usually fitted immediately after each pump. When the flow reverses, the check valve is activated and closed. A large pressure transient occurs in the pipeline when the flow reverses and the check valves of the pumps close rapidly. The equivalent pump characteristics in the pumping station during pump stoppage and pump run-down can be described by the homologous relationships for n_p, number of pumps [12]. In the case where the check valve closure time is known, the flywheel or pumpset inertia can be sized such that the pump continues delivery for a period longer than the check valve closure time. This will ensure non-reversal of flow before the check valve is able to close. Downstream of each of the above profiles, a constant head reservoir is assumed.

3. NUMERICAL COMPUTATION WITH AIR VALVES

The improved numerical model for the air valve used in this study was incorporated as a subroutine in a main program that simulates pressure transients caused by a trip in a pumping station with air entrainment. The original air valve model was given in [2,13]. The new assumptions made for the present improved air valve model are as follows:

1. air enters and leaves the pipe through the valve under isentropic flow conditions;
2. the air mass within the pipe follows a polytropic law (polytropic index $= n$);
3. the air admitted to the pipe stays near the valve where it can be expelled;
4. the elevation of the liquid surface remains substantially constant, and the volume of air is small compared with the liquid volume of a pipeline reach.

The flow rate of air through the air valve depends on the absolute atmospheric pressure p_o and temperature T_o outside the pipe, as well as the absolute temperature T and pressure p within the pipe. As a result of the above assumptions made, the coefficients and indices of the modelled equations are significantly different from those given in the earlier air valve models of [2,13]. For the present study, four cases of flow conditions are considered:

1. Subsonic air flow in ($p_o > p > 0.528p_o$):

$$\dot{m} = C_{in}A_{in}\sqrt{2\rho_o p_o\left[\left(\frac{p}{p_o}\right)^{2/n} - \left(\frac{p}{p_o}\right)^{(1/n)+1}\right]}. \tag{1}$$

2. Critical air flow in ($p < 0.528p_o$):

$$\dot{m} = C_{in}A_{in}\frac{0.528\sqrt{n}}{\sqrt{RT_o}}p_o. \tag{2}$$

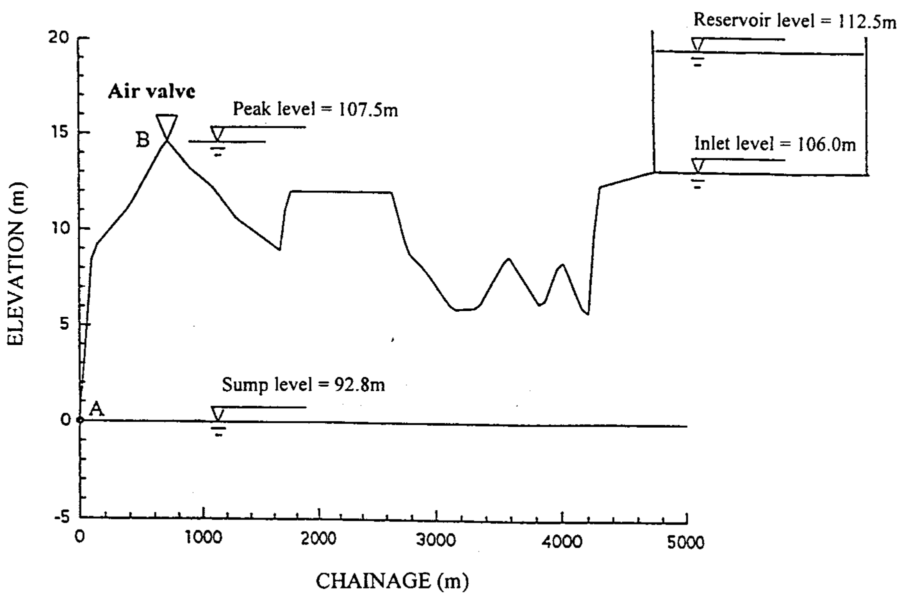

Figure 1. Pumping station pipeline profile.

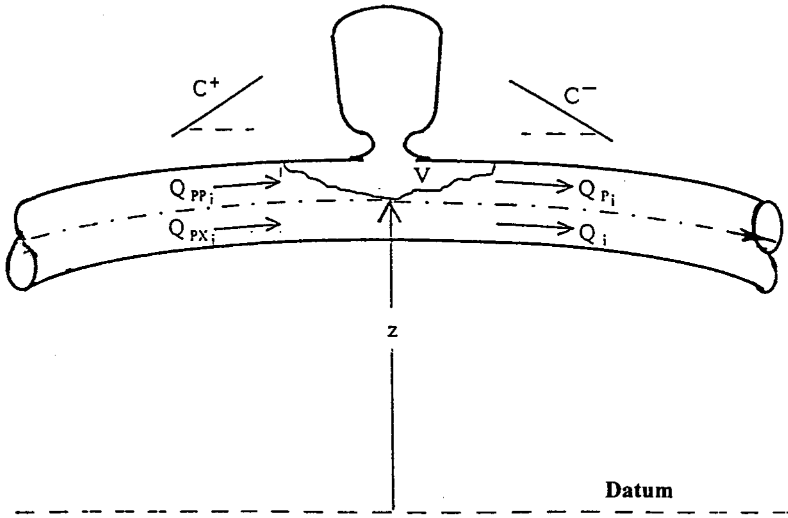

Figure 2. Air valve at location B.

3. Subsonic air flow out ($p_o/0.528 > p > p_o$):

$$\dot{m} = -C_{out}A_{out}p\sqrt{\frac{2}{RT}\left[\left(\frac{p_o}{p}\right)^{2/n} - \left(\frac{p_o}{p}\right)^{(1/n)+1}\right]}. \tag{3}$$

4. Critical air flow out ($p > p_o/0.528$):

$$\dot{m} = -C_{out}A_{out}\frac{0.528\sqrt{n}}{\sqrt{RT}}p. \tag{4}$$

The isentropic subsonic inflow and outflow formulae for predetermined intervals of $p' = p/p_o$ are then replaced with parabolas of the form:

$$\dot{m} = A_{2_i}p'^2 + A_{1_i}p' + A_{0_i} \qquad \text{Subsonic inflow for } p' \text{ between 0.528 and 1.0,} \tag{5}$$

$$\dot{m} = D_{2_i}p'^2 + D_{1_i}p' + D_{0_i} \qquad \text{Subsonic outflow for } p' \text{ between 1.0 and 1.894.} \tag{6}$$

The coefficients for A are computed for the parabola representing subsonic inflow between $p' = 0.528$ and 1.0 with NR intervals. Similarly, the coefficients for D are computed for the parabola representing subsonic outflow between $p' = 1.0$ and 1.894 with NS intervals.

When the pressure head is greater than atmospheric and no air is present in the pipe, the boundary condition at a junction of two is the usual internal section solution of H_{P_i} and Q_{P_i}. When the pressure head drops below pipe elevation, the air valve opens for air to enter, and the general gas law, $pV = mRT$, is to be satisfied at the end of each time increment of the computation until the admitted air is expelled.

With reference to air valve flow shown in Figure 2,

$$p\cdot[V_i - 0.5\Delta t((Q_{PX_i} - Q_i) + (Q_{PP_i} - Q_{P_i}))] = [m_0 + 0.5\Delta t(\dot{m}_0 + \dot{m})]\cdot RT, \tag{7}$$

where V_i is the volume of air cavity at beginning of time increment Δt; Q_{PX_i}, the initial fluid inflow into cavity (at start of Δt); Q_i, the initial fluid outflow from cavity; Q_{PP_i}, the final fluid inflow into cavity (at end of Δt); Q_{P_i}, the final fluid flow outflow from cavity; m_0, initial mass of air in cavity; $\dot{m}_0$, the initial rate of air mass flow into or out of cavity; $\dot{m}$, the final rate of air mass flow into or out of cavity.

The C^+ and C^- characteristics equations for the transient flow when applied to the air valve junction can be expressed in the short form as:

$$C^+ : H_{P_i} = CP = B \cdot Q_{PP_i}, \tag{8a}$$

$$C : H_{P_i} CM + B \cdot Q_{P_i}, \tag{8b}$$

where CP, CM, B are known constants at any time level.

The relationship between H_P and p is given by

$$\gamma(H_P - z + \bar{H}) = p, \tag{9}$$

where $\bar{H}$ is the barometric pressure head; $\gamma = \rho g$, the specific weight of liquid; z, the elevation of air valve above datum for H_P.

From Equations (8) and (9), Q_{PP_i} and Q_{P_i} can be expressed as a function of p'. Substituting Q_{PP_i} and Q_{P_i} as a function of p' into Equation (7), the resulting equation is to be solved for the end of each time increment when a cavity is present (i.e. $V' > 0$).

The solutions for H_P, Q_P are initially obtained from the C^+ and C^- characteristic lines for the pressure transient of the general pipeline system. When the local pressure H_P falls below atmospheric pressure, the air valve installed at the low pressure location is activated and p' is determined. The air inflow rate and air volume are then calculated from Equations (1)–(4) and Equation (7) respectively. During the subsequent pressure transient, the condition for inflow or outflow of air from the air valve is determined by the transient p' values and the volume of air residue in the pipeline near the air valve location. Equations (5) and (6) and Equations (7)–(9) are sufficient for the transient solution of the H_P and Q_P for the pipeline system with the air valves.

4. RESULTS AND DISCUSSION

The typical pipeline profile of a pumping station, together with the locations of a check valve at A and an air valve at B, are shown in Figure 1. The pumping system considered here uses three centrifugal pumps to supply 1.08 $m^3 s^{-1}$ of water to a tank 19.7 m above the sump level, through a 0.985 m diameter main, 4720 m in length. Swing check valves were installed downstream of each of the operating pumps. The pumpset moments of inertia (including the flywheel) were studied for a combined equivalent pumpset moment of inertia of $I_r = 99.9$ kg m^2. The air void fraction ε studied were in the range of 0.000–0.010. The effects of air entrainment on the pressure transients generated by the simultaneous pump trip of all pumps with the undulating pipeline contour shown in Figure 2, were investigated.

The non-return check valve is assumed closed when flow reverses. The air valve installed along the pipeline is assumed operative when the local pressure falls below atmospheric pressure or when there is a reverse positive pressure with a non-zero volume of the trapped air near the air valve area of influence. Various characteristics of the air valves are studied here. From the characteristic curves of the air valves used by Lee and Cheong [2] in the field studies, the flow coefficients for each air valve under the inflow and outflow conditions can be calculated from Equations (1)–(4) (model modified from [13]). It is noted in this study that high surge pressures can be caused by the sudden closure of the air valve following the admission of air into the pipeline. In practice, the air valve can be prevented from opening ($C_{in} = 0.00$) by fitting an inflow check valve. This modification allows the valve to vent ($C_{out} > 0.00$) any air trapped within the pipeline, but prevents air from entering, following an internal subatmospheric pressure condition. Under all operating conditions, the air valve should retain all liquid in the pipeline without loss of liquid to the atmosphere.

With no air valve installed, Figure 3 shows the effects of air entrainment on the pressure transient at a point A (immediately downstream of the check valves) and at a point B (at the peak where the air valve is to be installed). Immediately after the pumps shut down, there is a rapid decay in flow through each pump, and a loss in pumping pressure immediately

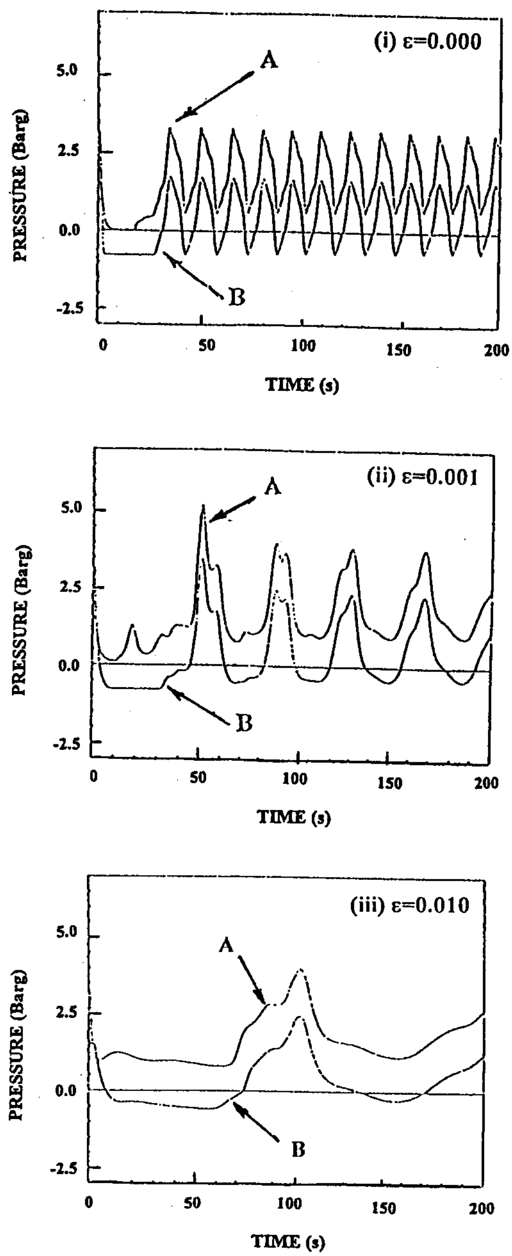

Figure 3. Pressure transients at locations A and B with no air valve.

downstream of the station. The continued forward momentum of the remainder of the flow in the pipeline causes the pressure to fall to subatmospheric levels, and this subatmospheric pressure propagates downstream. Typically, during a pump trip when the hydraulic gradeline of the system falls below the pipeline elevation at the peak location B, the line pressure at this location falls below atmospheric pressure. If the down surge is severe, the line pressure may even reach the vapour pressure of the pumping fluid. This will result in the release of dissolved gases in the fluid system. The subatmospheric horizontal transient portion of the curve in Figure 3 indicates that the line pressure has reached its vapour pressure during the initial transient period. Several distinct pressure transient characteristics are also observed from the above numerical experiment: (i) the pressure peak varies with ε and can be higher than that predicted by the constant wave speed model ($\varepsilon = 0.000$), with the transient time that occurs differing; (ii) the damping of the surge pressure is noticeably larger with $\varepsilon > 0.000$ when compared with the constant wave speed model ($\varepsilon = 0.000$); (iii) with $\varepsilon > 0.000$, the pressure surges are asymmetric with respect to the static head, while the pressure transient for the constant wave speed is symmetric with respect to the static head; (iv) when air is entrained into the system, the pressure transient shows long periods of downsurge and short periods of upsurge when compared with the gas-free constant wave speed case. Previous surge measurements [2,3,9,10] indicated that damping is faster in reality, suggesting that energy dissipation mechanisms other than ordinary friction are also operating; (v) the degree of amplification of the first pressure peak is dependent upon the rate of deceleration of the flow after pump trip. The above numerical experiments also provide useful information on the effects of air entrainment on the maximum and minimum wave speeds within a given pumping system. In general, the magnitude of the maximum wave speed decreases as the air entrainment value is increased.

Figure 3 shows that for $0.00 < \varepsilon < 0.010$, the maximum and minimum wave speed varies significantly depending predominantly on the amount of air within the pumping system. For $\varepsilon > 0.010$, the effects of air entrainment level on the transient wave speed is small as the main bulk of the fluid system would have low wave speed. The above observations through numerical experiments are consistent with available field measurements and observations [2,3,5,9,10] of pressure surges in prototype pumping stations for various modes of normal pump operations and pumps operating near low-water cut-out levels with air entrainment due to falling inflow jets and attached surface vortex. Observations also showed that the commonly used swing check valve closed when the flow reversed. At the instant of check valve closure, a large pressure variation was initiated. With air valve installed at location B, Figures 4 and 5 show the effects of air valve characteristics on the pressure transients of the fluid system considered under various air entrainment conditions. The results are presented here with respect to the variation of transient pressure, and the volumetric rate of air cavity formation/dissipation in the vicinity of the air valve in Figures 4 and 5.

Figures 4(a) and 5(a) generally show a reduction in the magnitude of the negative line pressure when an air valve is installed at the peak location B. Higher reductions in the magnitude of the negative line pressure can be obtained with the proper choice of the air valves (as shown in Figures 3 and 4(a) in which the C_{in} values of the installed air valves are not small. With higher C_{in} value, the rate of air inflow is sufficient to prevent the line pressure from falling below the vapour pressure. The effects of the different C_{in} values on the air flow rates can be seen in the amount of air cavity volume formed in Figures 4(b) and 5(b). Figures 4(a) and 5(a) generally show that, while installing an air valve at the peak location helps to reduce the magnitude of the negative line pressure, it may on the other hand also result in aggravated positive pressure surges, especially at a low air entrainment level (i.e. higher average system

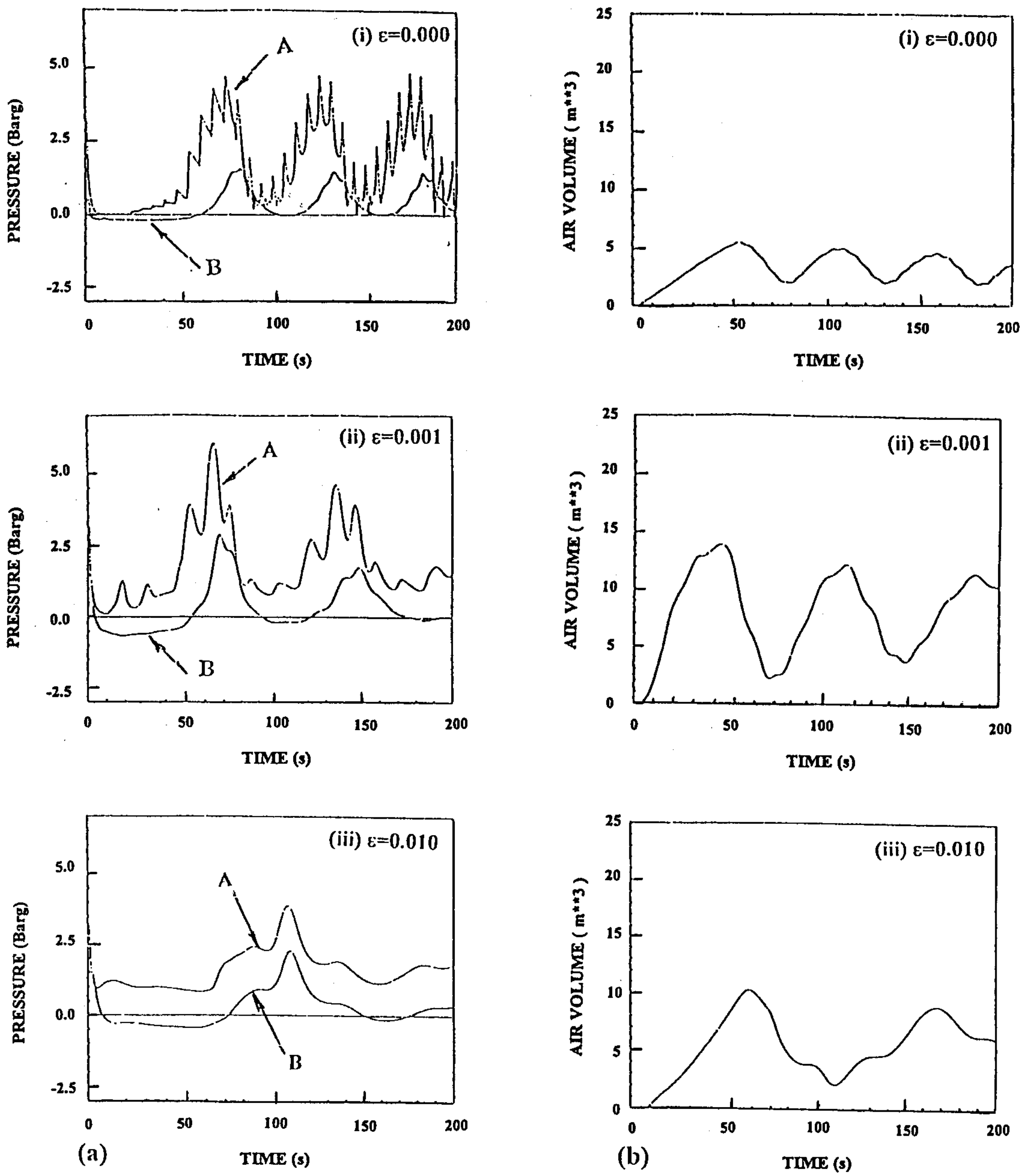

Figure 4. Fluid transient with air valve: $C_{in} = 0.237$; $C_{out} = 0.009$; (a) pressure transients at locations A and B; (b) transient air cavity volume at location B.

wave speed). The aggravated positive pressure surges worsen with higher values of C_{out} as shown in Figure 5(a). After substantial amount of air is admitted into the pipeline to restore the negative line pressure to atmospheric pressure, the admitted air will be expelled to the atmosphere when the positive surge pressure returns. With higher values of C_{out}, the admitted air may be expelled rapidly and completely within a short time interval. This results in a sudden rise in the line pressure when the water at the air–water interface comes into contact with the closing air valve seat. Figure 5(a)(i) illustrates the sudden rise in the line pressure when the final amount of admitted air is being expelled for a higher C_{out} value of 0.744 at zero air entrainment value. The high pressure resulting from rapid air valve closure can be alleviated by fitting an outflow restriction device to the air valve. This results in the air being vented out

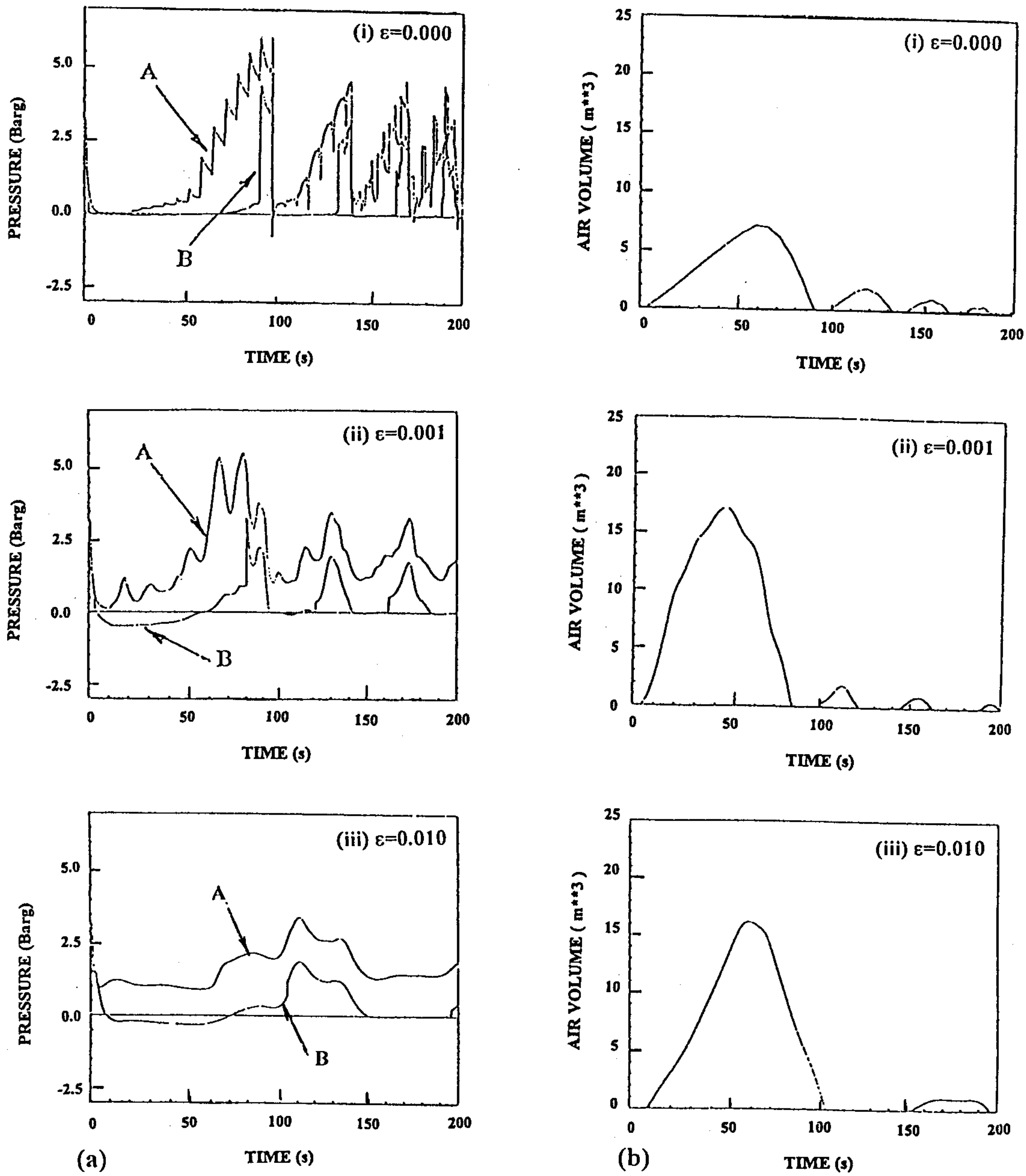

Figure 5. Fluid transient with air valve: $C_{in} = 0.504$; $C_{out} = 0.744$; (a) pressure transients at locations A and B; (b) transient air cavity volume at location B.

more slowly (lower C_{out}), reducing the reverse flow rate upon air valve closure and thereby alleviating subsequent high surge pressures. Figure 4(a)(i) illustrates a more benign pressure transient for a low C_{out} value of 0.009 at zero air entrainment value. The positive air cavity volume serves as a cushion between the water and the air valve so that a sudden rise in the surge pressure will not occur in the fluid system. In order to reduce the subatmospheric pressure produced upon pump shutdown, it was conceived that the number of air valves might be increased so that air could be admitted into the pipeline system more rapidly, thus achieving a greater equalisation in pressure between atmospheric and subatmospheric pressures within the pipeline. However, care should be taken here, since doubling or quadrupling the number of air valves used, has the same effect as increasing the C_{in} as well as the C_{out} values of an equivalent air valve. Thus, additionally, the maximum transient pressures may increase with the number of air valves used as this causes air to be vented more rapidly, causing a greater deceleration in flow when the air valves close.

From the above studies, it can also be concluded that in order to address the problem of increased air outflow (large C_{out}) occurring when using multiple air valves (which leads to rapid air valve closure), a vented non-return valve (reducing C_{out}) can be fitted to the air valve, which gives a restriction in outflow (low C_{out}) while maintaining the standard (high C_{in}) rate of inflow.

5. CONCLUSIONS

A numerical computational model was proposed here for the study of the effects of air valves on a transient fluid system. Numerical studies were also carried out to investigate the effectiveness of air valves with different inflow and outflow characteristics in reducing the magnitude of pressure surges in a typical pumping station. Numerical experiments showed that air valves with a high inflow coefficient C_{in} generally reduce the magnitude of negative line pressure at high points in the pumping station. Air valves with a low outflow coefficient C_{out} would ensure that large positive pressure surges due to air valve slamming would not occur. Air valves with high outflow characteristics would very often result in rapid and complete exhaustion of the air admitted when the line pressure falls below atmospheric pressure, resulting in large positive pressure surges in the pipeline.

APPENDIX A. NOMENCLATURE

General symbols

g	gravitational acceleration
H	gauge piezometric pressure head
I	node point at $x_i = (i-1) \cdot \Delta x$
k	time level at $t^k = \Delta\Sigma t^k$
n_p	total number of pumps
NR	domain intervals used in Equation (5)
NS	domain intervals used in Equation (6)
P	pressure inside the pipe
Q	fluid flow rate
R	universal gas constant
t	time
V	flow velocity

V_o steady state flow velocity of fluid system
x distance along pipeline
z elevation with respect to pump intake level

Symbols for air valve
A inflow or outflow nozzle area of the air valve
C inflow or outflow coefficients of the air valve
$\dot{m}$ rate of air mass flow into or out of air cavity at the location of the air valve
MC critical air mass flow rate through the air valve nozzle at choke flow conditions
n polytropic index
p absolute pressure inside the pipeline at the location of the air valve
p_o absolute atmospheric pressure outside the pipeline at the location of the air valve
p' pressure ratio $p' = p/p_o$
Q liquid flow rate within the pipeline into or away from the air valve region
T temperature in K
V_i volume of air cavity at beginning of time increment Δt

Greek letters
Δt^k time step at kth time level
Δx node point distance along pipeline
ε fraction of air in liquid
ρ density of the fluid

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