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# NUMERICAL STUDIES ON EFFECT OF CHECK VALVE PERFORMANCE ON PRESSURE SURGES DURING PUMP TRIP IN PUMPING SYSTEMS WITH AIR ENTRAINMENT

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SUMMARY

Recent numerical investigations on pressure surges during pump trip in pumping installations showed that by including an air entrainment variable wave speed model, reasonable predictions of transient pressure surges with proper phasing and attenuation of pressure peaks can be obtained. These calculated results are consistent with similar field measurements made with the pumps operating at low pump cut-out levels, when air entrainment due to an attached surface vortex was observed. However, in the numerical calculation procedures it is assumed that the inertia of the moving elements of the check valve is small and that the check valve closes at zero reverse flow velocity. In practice, check valves seldom close precisely at zero reverse flow velocity. With the check valves not closing at zero reverse velocity, the present numerical computations show that the air content in a fluid system can adversely affect the check valve performance. With the fluid system operating within a critical range of air entrainment values, the present analysis showed that there is a possibility of 'check valve slamming' when the check valves are selected based only on the analysis of an air-free system. This phenomenon is confirmed through field observations

KEY WORDS: air entrainment; pressure surges; pumping system; check valve slamming

## 1. INTRODUCTION

A common flow system arrangement in water and sewage engineering consists of a lower reservoir, a group of pumps with a check valve in each branch and a pipeline discharging into an upper reservoir (water tower, gravity conduit, aeration well, etc.). Check valves are items of equipment that are fitted to pipelines in order to prevent the lines from draining backwards when the pumps stop and sometimes also to prevent the downstream reservoirs from emptying. In addition, they prevent reverse rotation of pumps, thereby avoiding damage to seals and to the brush gear of the driving flows have been very well documented by Whiteman and Pearsall,[1,2] Provoost,[3–6] Kubie,[7] Collier and Hoerner[8] and Thorley.[9,10] However, none of these researchers considered the effect of air entrainment in a pipeline system and its subsequent effect on check valve performance. Air entrainment in fluid systems frequently occurs in reality owing to falling jets of sewage from the comminutors into the sump near the operating pump bell-mouths, attached vortex formation arising from the operation of the pumps at low cut-out levels, air-entraining undulating turbulent flow towards the operating pumps due to flow baffling, etc. The flow in a pipeline will also contain free gas, although the volumetric proportion may be small. Trapped air pockets at the top of the pipe cross-section at high points along the pipe profile can also be present owing to the incomplete removal of air during commissioning and filling-up

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operations or the progressive upward migration of pockets of air. Air may also be admitted into the pipeline as a result of vortex action in an inadequately designed air vessel. Most sewage also contains dissolved gases in solution. Gas bubbles will be evolved from the liquid during the passage of low pressure transients. When the liquid is subject to high transient pressures, the free gas will be compressed and some may be dissolved. The resultant fluid transient characteristics of a fluid system and their effects on check valve performance can thus be expected to be complex.

A pipeline contour of a pumping station with possible air entrainment is shown in Figure 1. A detailed analysis of this pipeline contour with a simplified assumption on check valve closure with no reverse flow allowed was presented earlier by Lee.[11] The most severe case of all the pumps in the station failing simultaneously owing to power failure was analysed in terms of the maximum and minimum pressure variations along the pipeline. When the pumps tripped, the forward flow diminished rapidly to zero and the check valves were assumed closed at zero reverse flow. In the present study the check valves are not assumed closed during the period of forward flow deceleration to zero flow. It is assumed instead that the flow may reverse and the check valves will close during reverse flow, possibly resulting in unexpectedly high pressure variations and slamming of the valves. It is thus important to know in the present work the reverse flow characteristics of the fluid system and the check valve performances so that suitable types of check valves can be selected to satisfy the design specifications of a pumping system. For this purpose a numerical model was set up in the present work to investigate the check valve performance in a pumping system with air entrainment. Although in theory it is possible to include a model of the dynamic behaviour of the check valve[4] in the above transient analysis, in practice this is not often possible, because the data necessary (flow-induced torque, etc.) are very seldom known for check valves. The valves typically used in sewage pumping stations are of the swing check valve type. The valves are usually mounted horizontally at the immediate upstream end of the pumps. The axis of the valve flap is located at the top of the valve body. In order to speed up the closure of the valves, sometimes an extra weight is attached to the axis via a bar. When the valve is closed, the angle between the bar and the horizontal plane is small (typically about 2°–5°); when the valve is not fully open (e.g. one pump running), the angle is somewhat larger (in the range 20°–25°). Quick-acting recoil-type check valves are now increasingly being used for high-flow-rate and high-lift pumps. Such valves include spring-assisted nozzle-type check valves. These spring-assisted nozzle check valves are usually more expensive in terms of higher energy consumption due to added pressure

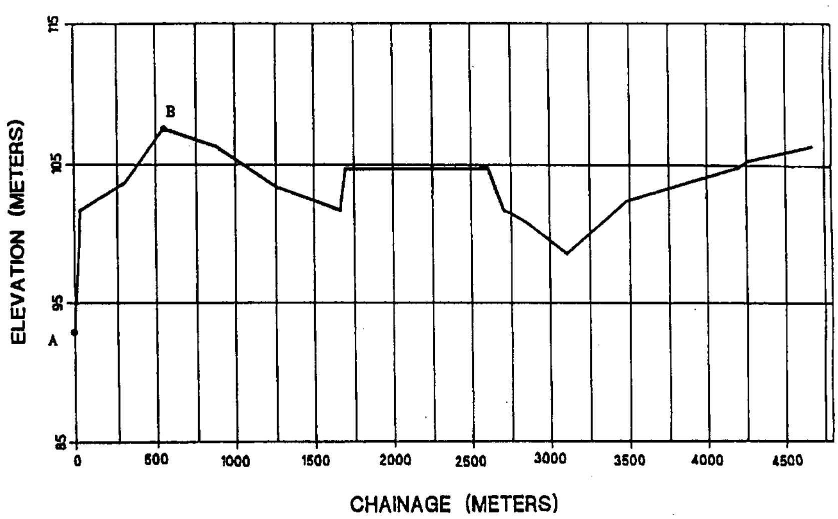


Figure 1. A typical pipeline contour of pumping station with entrainment

losses through the valve, higher initial capital cost and more maintenance being required in the long run. However, very often the installation of this type of valve is necessary, since it offers a quick closure time which minimizes the check valve slamming problem. These valves are especially suitable for pumping systems with multiple high-flow-rate and high-lift pumps in various modes of operation. To avoid slamming of the check valve, it is necessary that the moving element of the check valve reaches its closed position at the instant the forward flow becomes zero. Whether this will occur in the prototype pumping system can be analysed by calculating the time interval during which the flow in the relevant pipe system decreases to zero and the check valve closing characteristics. However, as mentioned earlier, in order to be able to predict the closure of the valve more accurately, the motion of the water during the valve closing stage must also be analysed. A more detailed analysis of the motion of the fluid just before valve closure will lead to a better prediction of whether slamming will occur or not. The analysis presented in this paper provides a mean of analysing the above phenomena. It is assumed here that for a given geometry of the valve the valve opening during steady state is determined by the steady state flow rate $V_0$. The higher the steady state flow rate, the larger is the valve opening and the longer it takes the moving elements of the check valve body to close completely when the flow reverses. The fluid velocity gradient $|dV/dt|$ is also assumed to play an important part in the dynamic phenomena during valve closure. If the fluid system flow rate decrease is gradual (e.g. using an appropriately sized flywheel) with a longer time period for the first flow reversal (i.e. flow reversal time $t_R$), valve slamming and high-pressure surges can generally be minimized. Making the above assumptions, in the present analysis we consider a fluid system with air entrainment $\varepsilon$ and study numerically the latter's effect on the flow reversal time and hence determine the appropriate valve selection for a given fluid system to minimize the check valve slamming problem. From the available literature on dynamic valve closure characteristics[3–6,9,10] it is observed that the rate of decrease for a given flow rate is approximately linear over the interval during which the flow rate starts to decrease and eventually reverses. This physical observation is implemented in the present numerical calculation procedures in order to better determine the suitability of a check valve type for a given pumping system.

## NUMERICAL METHOD FOR METHOD OF CHARACTERISTICS WITH VARIABLE WAVE SPEED

The method of characteristics applied to the above pressure transient problem with variable wave speed $a_i$ can be described by the following finite difference expressions—$C^+$ characteristics:

$$\frac{g}{a_R}\frac{H_i^{k+1}-H_R}{\Delta t^k}+\frac{V_i^{k+1}-V_R}{\Delta t^k}+\frac{g}{a_R}V_R\sin\alpha_i+\frac{f_{t_R}}{2D}V_R|V_R|=0, \tag{1a}$$

$$\frac{x_i-x_R}{\Delta t^k}=V_R+a_R; \tag{1b}$$

$C^-$ characteristics:

$$-\frac{g}{a_S}\frac{H_i^{k+1}-H_S}{\Delta t^k}+\frac{V_i^{k+1}-V_S}{\Delta t^k}-\frac{g}{a_S}V_S\sin\alpha_i+\frac{f_{t_S}}{2D}V_S|V_S|=0, \tag{2a}$$

$$\frac{x_i-x_S}{\Delta t^k}=V_S-a_S. \tag{2b}$$

With conditions known at points $i-1$, $i$ and $i+1$ at the $k$th time level, the conditions at R and S can be evaluated by a linear interpolation procedure. The conditions at R and S are then substituted into

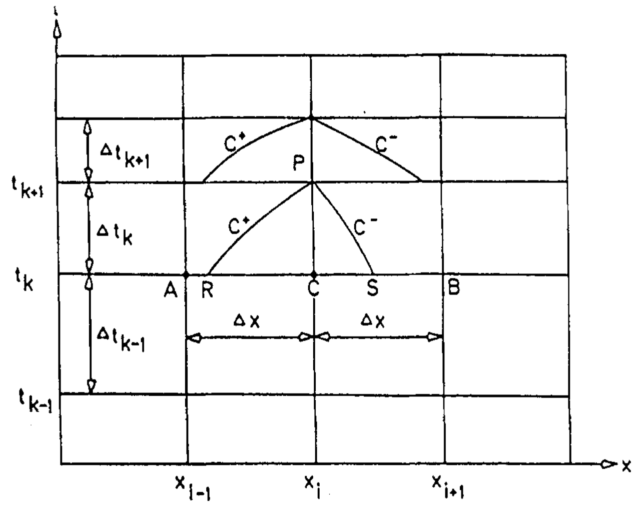


Figure 2. Computational grid for variable wave speed with variable time steps

equations (1) and (2) and the solutions at the $(k+1)$th time level at point $i$ are obtained for $i=0, 1, \ldots, N$. A mesh size of $N=1000$ was used for the solutions presented in this work. With reference to the irregular $t$- and regular $x$-grid notation used in Figure 2, $i$ denotes the regular $x$-mesh point values at location $x=(i\Delta x)$ and $k$ denotes the irregular time level corresponding to time at $t^k = \sum(\Delta t^k)$.

With air entrainment the transient computation of the fraction of air content $a_i^k$ along the pipeline depends on the local pressure and local air volume and is given in Reference 11. The loss factor $f_1$ used in conjunction with the method of characteristics[12] with air entrainment and gas release in a pipeline system is evaluated at the local point $i$ using the characteristics of the flow at that point. The steady state overall loss factor at the operating point of a system can be determined from the pump characteristic curve and the system curve.

## NUMERICAL COMPUTATION OF PUMP RUN-DOWN WITH DYNAMIC CHECK VALVE RESPONSES

Downstream of the above pipeline profile of the pumping station a constant head reservoir, i.e. $H_N^{k+1}=\text{constant}$, is assumed for all time levels and this is solved with the $C^+$ characteristic line for $V_N^{k+1}$ at each time level. The upstream pumping station is modelled by an equivalent pump characteristic with $n_p$ pumps running in parallel. During pump stoppage and pump run-down the homologous relationship for $n_p$ pumps in the form of an equivalent pump is modelled numerically.[11] When a reverse flow is encountered in the pump, the check valve responds according to the experimentally obtained dynamic characteristics of check valves given by Thorley.[10] The general characteristics of the check valve are numerically modelled here as

$$V_R/V_0 = D_1 + D_2(A^*) + D_3(A^*)^2 + D_4(A^*)^3, \tag{3}$$

where $A^* = |dV/dt|/[(V_0)^2/D]$ is the reverse flow deceleration parameter through the check valve, $V_0$ is the steady state flow velocity, $dV/dt$ is the reverse flow velocity gradient, $D$ is the check valve nominal diameter, $D_1$, $D_2$ and $D_3$ are the characteristic parameters of the type of check valve and $D_4$ is the characteristic parameter of the check valve due the effect of air entrainment. For no entrainment $D_4=0{\cdot}0$.

Check valves serve to prevent the reversal of flow in a pumping system. If, however, a reversal of flow occurs in a very short time, the valve may close after the flow has already been reversed. Depending on the type of check valve used, a sudden decrease in the reversal of flow will occur, possibly resulting in unallowable pressure variations and slamming of the valve. To theoretically predict whether slamming of a check valve is to be expected, data under dynamic conditions of the valve have to be known. In most cases these data are not known. The possibility of a check valve slamming, however, may be estimated through a numerically predicted 'flow reversal time' $t_R$. Prototype valve tests showed that the variation in the fluid velocity gradient $|dV/dV|$ with time is of decisive importance when considering the valve slamming problem. Obviously, the shorter the time of transient flow reversal from pump trip, the more likely it is that valve slamming will occur. This again depends on the type of check valve used. Nozzle-type of recoil check valves tend to have a better dynamic response than the conventional swing-type check valve. It is thus important to know the 'flow reversal time' during pump trip so that at the design stage of a hydraulic project it is possible to predict whether the chosen type of check valve will satisfy the design specifications of the pumping system. Specifically, the aim of this study is to determine the effect of air entrainment on the 'flow reversal time' and thus its effect on check valve selection and the consequences of pressure surges for a pumping system.

## RESULTS AND DISCUSSION

Initially the effects of air entrainment on pressure transients generated by the simultaneous pump trip of all pumps operating in a pumping station with the undulating pipeline contour as shown in Figure 1 were investigated for a selecting swing check valve configuration. The pumping station uses three parallel centrifugal pumps to supply $1{\cdot}08\ m^3\ s^{-1}$ of water to a tank 19·7 m above the sump level, through a 0·985 m diameter main of 4720 m legnth. Swing check valves were installed downstream of the pumping station. The pumpset moments of inertia (including the flywheel) were studied for equivalent pumpset moments of inertia $I_e$ of 0·1, 0·5, 1·0, 2·0, 5·0 and 10 times a reference pumpset inertia $I_r = 99{\cdot}9\ kg\ m^2$. The air void fractions $\varepsilon$ studied were in the range of 0·00–0·03.

Figures 3–5 show the effects of air entrainment on the pressure transients at a point A (immediately downstream of the check valves) and at a point B (at the peak) of the pipeline contour for various equivalent pumpset moments of inertia. Five distinct pressure transient characteristics were observed from the above numerical experiments. (i) The pressure peak varies with $\varepsilon$ and is above that predicted by the constant wave speed model ($\varepsilon = 0{\cdot}000$), with the transient time that occurs differeing. (ii) The damping of the surge pressure is noticeably larger with $\varepsilon > 0{\cdot}000$ when compared with the constant wave speed model ($\varepsilon = 0{\cdot}000$). (iii) With $\varepsilon > 0{\cdot}000$ the pressure surges are asymmetric with respect to the static head, while the pressure transient for the constant wave speed model was symmetric with respect to the static head. (iv) When air was entrained in the system, the pressure transient showed long periods of downsurge and short periods of upsurge when compared with the gas-free constant wave speed case. Previous surge measurements[12,13] indicate that the damping is faster in reality, suggesting that energy dissipation mechanism 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. An increased pump inertia (by attaching a flywheel to the drive shaft) generally produces a slower rate of deceleration of the flow after pump trip and a smaller amplification of the first pressure peak as compared with the constant wave speed model.

The above numerical model also provides useful information on the effects of air entrainment on the maximum and minimum wave speeds within a given pumping system. Figure 6 shows that for $0{\cdot}00 < \varepsilon 0{\cdot}02$ the maximum and minimum wave speeds depend predominantly on the amount of air within the pumping system. The pumpset inertia has very little influence, except when a very large

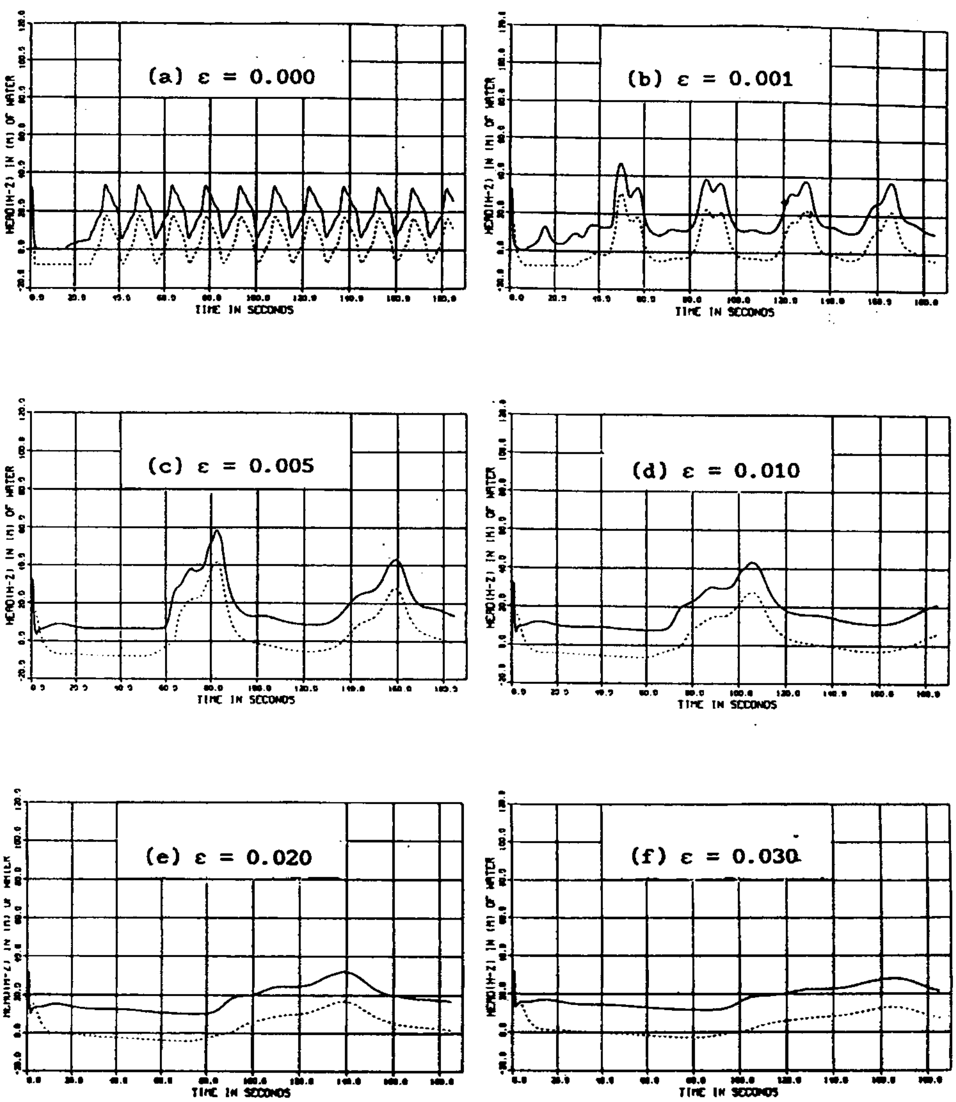


Figure 3. Air entrainment effects on transient pressure heads (pumpset inertia with flywheel, $I_e = 0{\cdot}1 I_r$) downstream of pumping station at a (———) and at peak of pipeline elevation at B (– – – –)

flywheel is attached to the shaft of the motor driving the impellers of the pumps. Similarly, Figure 7 shows that for $0{\cdot}00 < \varepsilon < 0{\cdot}02$ the maximum and minimum pressures along the pipeline are predominantly dependent on the air entrainment level. For a given pumpset inertia the maximum pressure peaks and valve slamming problem need not necessarily occur at the minimum or maximum air entrainment level. They can occur in an intermediate critical range of air entrainment values. This range of critical air entrainment values can only be obtained through numerical experimentation for a given pumping system. Figure 8 shows that entrained, entrapped or released gases significantly after the flow reversal time. From the available flow reversal times the reverse flow velocity gradient with respect to time is obtained. The maximum reverse flow gradient and the corresponding maximum reverse flow for a given pumping configuration at various air entrainment levels for a swing check

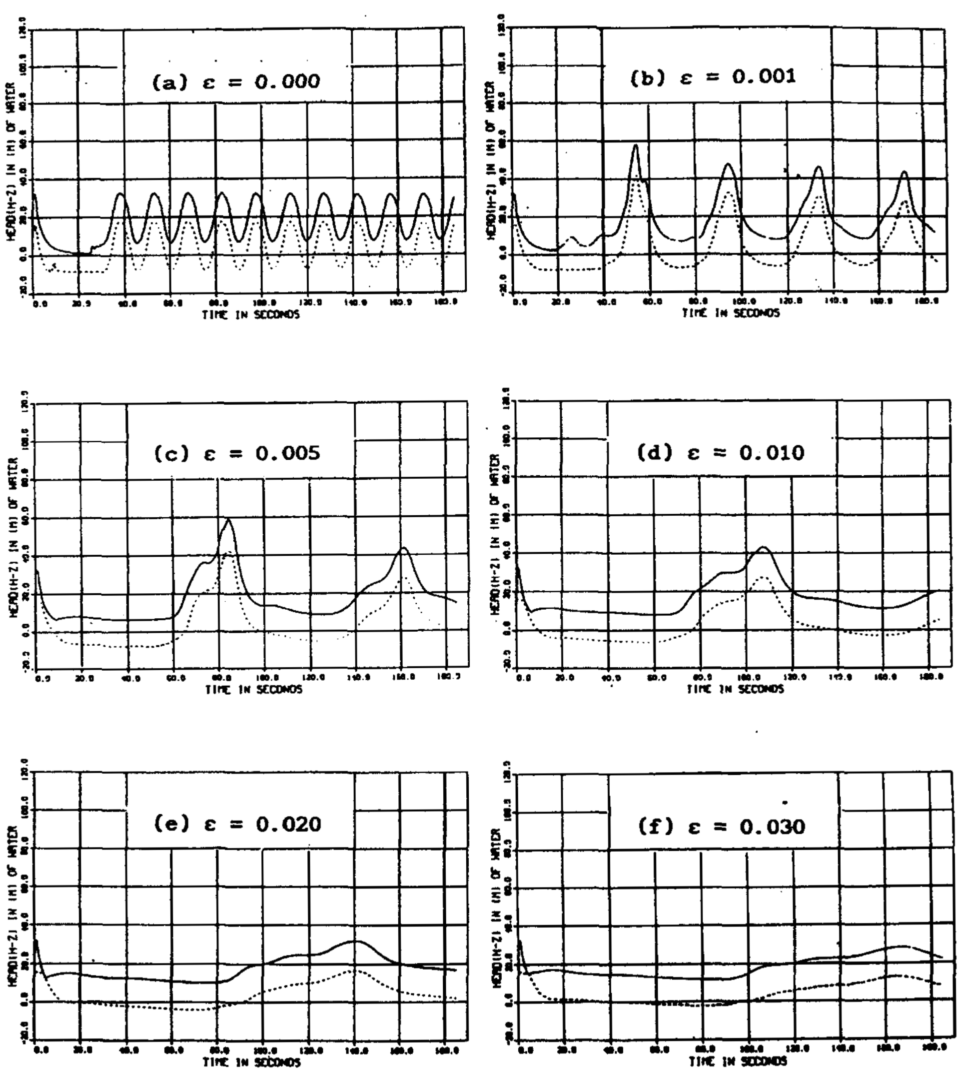


Figure 4. Air entrainment effects on transient pressure heads (pumpset inertia with flywheel, $I_e = 1{\cdot}0I_r$) downstream of pumping station at A (———) and at peak of pipeline elevation at B (– – – –)

valve are shown in Figure 9. The results show that although the selection of the swing valve is in general satisfactory for the above pumping system, there is still a possibility of the 'valve slamming' problem unless the pumpset inertia is significantly increased. The results presented in Figure 9 show that the effects of the check valves in transient flow can be predicted by means of numerical computation of the equivalent flow reversal time. When analysing these effects, both the steady flow rate and fluid velocity gradient with time have to be considered. For systems with large flow rates the fluid velocity gradient is of decisive importance. The time interval necessary for flow reversal and the mode of the flow rate versus time variation during the flow reversal period give an indication of the necessary characteristics for satisfactory flow and pressure predictions for check valve closure. The above observations through numerical experiments are consistent with field measurements and observations[12,13] of pressure surges in prototype pumping stations for various modes of normal pumps

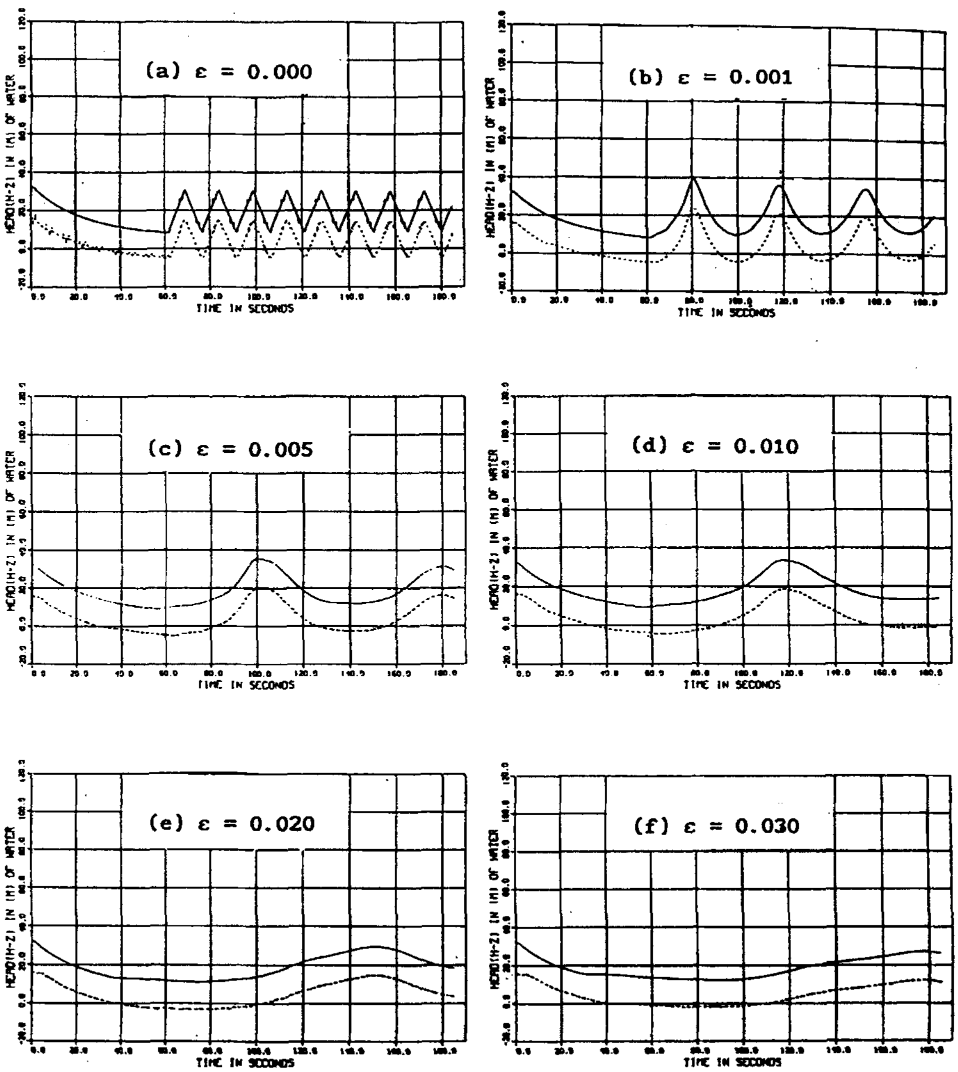


Figure 5. Air entrainment effects on transient pressure heads (pumpset inertia with flywheel, $I_e = 10I_r$) downstream of pumping station at A (———) and at peak of pipeline elevation at B (– – – –)

operations and pumps operating near low-water cut-out levels with air entrainment due to an attached surface vortex. Observations showed that the commonly used swing check valve closed with the flow reversed. At the instant of valve closure a large pressure variation was initiated.

## CONCLUSIONS

The present analysis shows that the effects of air entrainment on check valve performance in transient flow with various modes of pump operation may be predicted by means of numerical computation of the equivalent flow reversal time. Both the steady flow rate and fluid velocity gradient with time needs to be considered in the computation. The time interval required for flow reversal and the mode of the flow rate versus time variation during the flow reversal period give an indication of the required check valve characteristics necessary for a satisfactory pressure transient in a pumping system without the

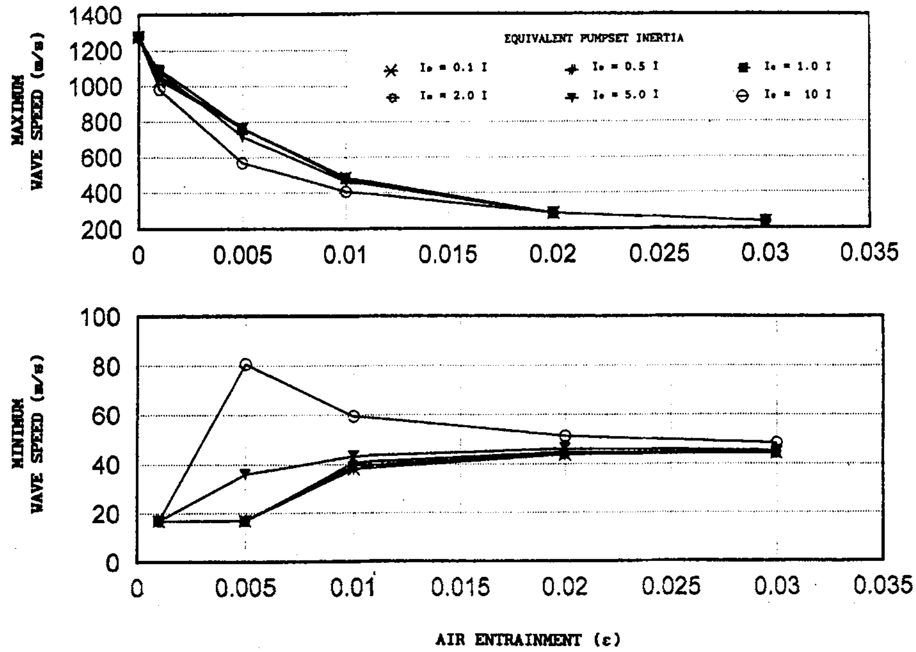


Figure 6. Effects of air content ε on pipeline system maximum transient and wave speed pipeline system minimum transient wave speeds

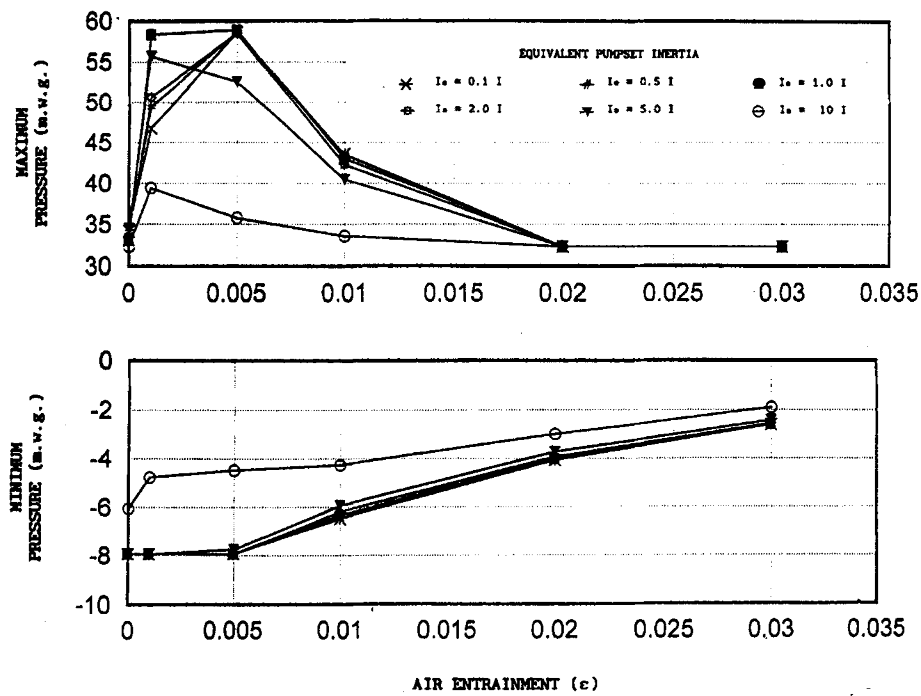


Figure 7. Effects of air content ε on pipeline system maximum transient and pressure head pipeline system minimum transient pressure heads

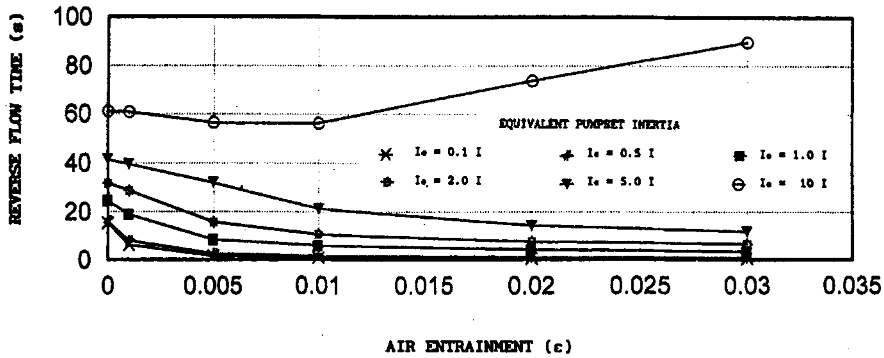


Figure 8. Effects of air content ε on flow reversal time

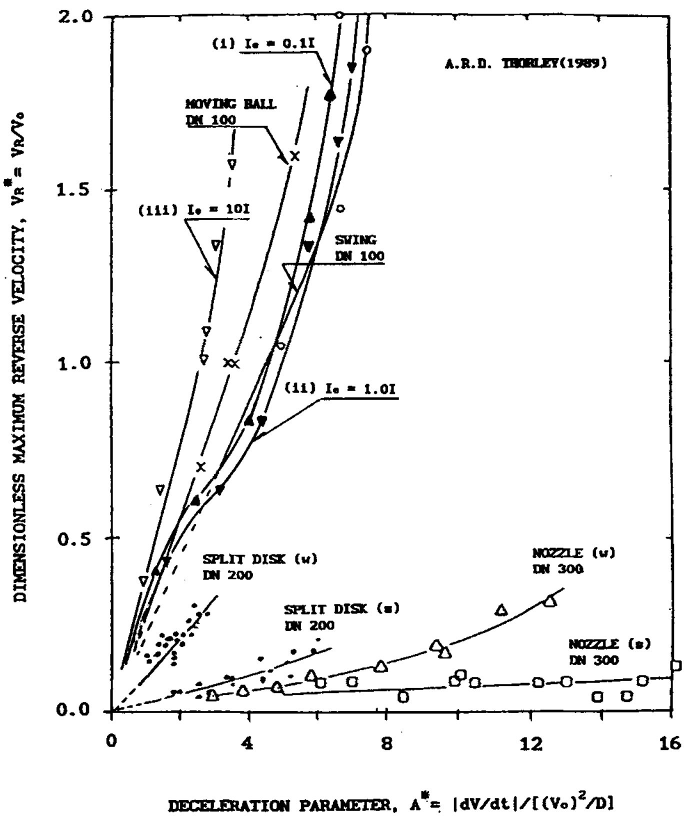


Figure 9. Effects of air content ε on selection of check valves (results of superimposed on check valve performances of Thorley[10])

valve slamming problem. From the above analysis it is seen that in order to find the most severe conditions for pressure surges and the valve slamming problem in a given fluid system, it is necessary to perform a complete air entrainment computer study. The reason is that for a given system the maximum pressure peaks and valve slamming phenomenon need not necessarily occur at the minimum or maximum air entrainment level. They can occur within a range of intermediate air entrainment values. This range of critical air entrainment values can only be obtained through numerical experimentation.

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## APPENDIX: LIST OF SYMBOLS

| | |
|---|---|
| $a$ | wave speed |
| $f$ | friction factor |
| $g$ | gravitational acceleration |
| $H$ | gauge piezometric pressure head |
| $H_g$ | gas release pressure head ($=2{\cdot}4$ m water absolute) |
| $i$ | node point at $x_i = (i-1)\Delta x$ |
| $I$ | pump set moment of inertia including flywheel |
| $k$ | time level at $t^k = \sum \Delta t^k$ |
| $L$ | length of pipe |
| $n_p$ | number of pumps in a pumping station |
| $N$ | total number of node points along pipeline |
| $P$ | pressure inside pipe |
| $Q$ | fluid flow rate |
| R | $C^+$-line intercept on $x$-axis at $k$th time level |
| $Re$ | Reynolds number |
| S | $C^-$-line intercept on $x$-axis at $k$th time level |
| $t$ | time |
| $V$ | flow velocity |
| $V_0$ | steady state flow velocity of fluid system |
| $V_R$ | reverse velocity during check valve closure |
| $x$ | distance along pipeline |
| $z$ | pipeline elevation w.r.t. pump intake level |

*Greek letters*

| | |
|---|---|
| $\alpha$ | pipeline inclination (positive downward) |
| $\Delta t^k$ | time step at $k$th time level |
| $\Delta x$ | node point distance along pipeline |
| $\varepsilon$ | fraction of air in liquid |
| $\nu_s$ | sewage kinematic viscosity |
| $\nu_g$ | gas kinematic viscosity |
| $\rho$ | density of fluid |

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