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UNDERSTANDING AIR RELEASE THROUGH AIR VALVES

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ABSTRACT

Filling and emptying pipeline systems involve the movement of large volumes of water and air. Water transients occurring during these operations can originate large pressure peaks that can severely damage distribution networks. Entrapped air, depending on the circumstances, can have a damping or amplifying effect on these undesirable pressure peaks. Unfortunately, too often, the complexity of the phenomenon makes it difficult to obtain a fully reliable prediction on when air pockets will mitigate or accentuate water transients. For such prediction, equations governing the movement of the water column and the behaviour of the air pockets must be combined and, for practical reasons, simplified. These simplifications conventionally lead to questionable hypotheses (e.g. normal cross-section of water-air interface) required to solve the numerical model with a reasonable calculation time.

Furthermore, the value of some of the parameters used in the numerical model cannot be calculated nor measured and need to be determined through a calibration process. Inevitably, this circumstance will lead to an even higher uncertainty of the results.

With the aim of overcoming most of the aforementioned uncertainties, this paper summarises a complete set of tests conducted at WL | Delft Hydraulics. These tests were satisfactorily simulated by means of a tailored numerical model that includes a set of parameters whose values (the main obstacle to be overcome) were determined by means of a calibration process. The experimental setup (a large-

scale facility) consisted of a single steep pipeline with an air valve installed at its top end. Air release through different air valves were tested under different conditions while the main variables (flow rate and pressure) were registered continuously at several points along the pipe. Lastly, and in order to be able to provide air valve manufacturers with helpful guidelines, the relevance of the parameters was evaluated by means of a sensitivity analysis.

Keywords: Air valves, entrapped air, transient analysis.

INTRODUCTION

Trapped air, if not properly removed when filling operations take place, can cause considerable damage to pipeline systems and other installed elements. This is a well-known problem that may give rise to pipe ruptures with important economic consequences (Chaiko and Brinckman, 2002). Additionally, air pockets may have other negative effects like reducing pipeline flow capacity, diminishing pump efficiency, hindering filter operation, generating pipe vibrations, damaging mechanical meters, and increasing errors in flow measurements, among others (Lauchlan et al., 2005). Some of these negative effects have been widely reported in technical literature. For instance, air pressure was responsible for lifting up manhole covers and blowing up sewerage pipes in Hamilton, Ontario, Canada (Hamman and McCorquodale, 1982) – events that were widely analysed two decades later (Zhou, 2002).

In contrast, if adequately managed, entrapped air can have a cushioning effect, damping overpressures when filling operations are being conducted. This positive behaviour is only achieved when pipelines are filled slowly (Izquierdo et al., 1999) and the air valves used to control the air flow, both in and out of the system, are carefully sized and located (Jönson, 1985; Thorley, 2004; Yongliang and Vairavamoorthy, 2006).

Theoretical and experimental analyses of hydraulic transients with entrapped air have been widely described in the technical literature. Some of them do not consider air release through orifices or through valves (Abreu et al., 1992), while others take into account both scenarios (Martin, 1976; Zhou, 2002; DeMartino et al., 2008). In this line, Lingireddy et al. (2004)

proposed a simplified equation, which did not consider the compressibility of the air pocket, to estimate the pressure surges in pipeline systems resulting from air release. Despite all the work carried out to date, the influence of the dynamic behaviour of air valves on water transients is not fully understood and works dealing with this topic are still scarce. The availability of adequate facilities to conduct the experiments on a full-scale level, the economic cost of the tests, the complexity of the physical phenomenon to be modelled and the best-known and simplest strategy to avoid problems in the field (cautiousness) explain why attention is rarely paid to this topic.

This work aims to fill this lack of data by providing experimental results on a set of tests carried out in a full-scale facility at WL | Delft Hydraulics. Some initial results from these tests have already been published (Arregui et al., 2003; Kruinsbrick et al., 2004). The setup was built to simulate the expulsion of air through commercial air valves located at the top end of a pipe. The experimental results describe the time history of the main variables recorded during the tests: water column velocity, system pressure at different points of the facility and air valve float displacement.

As two fluids are involved, a numerical model that describes the phenomenon must take into account both, the water column equations and the air pocket behaviour. Additionally, the numerical solution to these transients should also consider a moving boundary (water-air interface) so as to be able to handle the variation in the length of the liquid column (Martin, 1976). In this particular study, the method of characteristics with a mobile boundary condition (Abreu et al., 1992) was used. However, other alternatives like the one proposed by Malekpour and Karney (2008) could be used to simulate the movement of the water column.

To accurately reproduce the physical phenomena, the model tailored for this particular case includes several parameters. Some of these parameters characterise the fluid-pipe behaviour, such as the air compression coefficient, k , the pressure wave celerity, a , and the pipe friction factor, f . The remaining parameters are directly linked to the behaviour of the air valve, i.e. the air valve discharge coefficient, C_d , the volume of trapped air when the air valve begins to close, V_{air} , and the air valve closing time, t_c . These parameters, which are related to the

behaviour of the air valve, cannot be measured in the laboratory in a straightforward way. Some of them have to be estimated indirectly, clearly increasing the uncertainty associated to the estimation of their real value.

Although in practical applications the main target is the calculation of the maximum pressure peak (Epstein, 2008), this work aims to contribute to the understanding of the physical phenomena. Subsequently, two are the main objectives of this paper. First, to identify the set of values of the six parameters used in the mathematical model that best describe the air expulsion events generated in the laboratory. And second, identify by means of a sensitivity analysis which of them have the greatest influence on the final results of the tests.

Last, it is important to underline that the mathematical model presented does not consider the dynamic characterization of the air valves. However, Kruinsbrink et al. (2004) made an initial attempt on this subject. The approach used was similar to the one successfully followed for the dynamic characterization of check valves. Unfortunately, air valve characterization is a much more complex matter, and it should include the interaction of two fluids having different phases. Furthermore, most of the times, these devices are installed in parallel (although not in the case of the tests described in this work), increasing the complexity of the interaction between the two fluids and the system components – air valves can be connected to pipes with many different setups. Just as a matter of fact, a considerable number of the parameters that can be involved in the dynamic characterization are exclusively linked to gaseous phase. Finally, even though in this particular case the dynamic characterization has not been considered, theoretical and practical results match very well.

EXPERIMENTAL SETUP

The laboratory tests that reproduced the filling of an empty pipe were conducted at the WL | Delft Hydraulics facilities using the setup shown in Fig. 1. A DN100 single body air/vacuum

release valve, having a stainless steel spherical float (457 gr.), was installed at the top end of a galvanized steel pipe with an internal diameter of 489 mm and 9.5 mm of thickness.

The accuracy and frequency response of the instrumentation used to measure physical variables were carefully selected and verified before starting the tests. All instruments, including pressure transducers and the electromagnetic flowmeter (EMF), were calibrated in the laboratory. The dynamic pressure transducers had a measuring frequency response as high as 50 kHz. However, the greatest uncertainty was related to the real measuring frequency response of the EMF available in the laboratory. As it is well known, there is not an easy procedure for the dynamic calibration of this type of device. However, this specially designed meter was successfully used in previous studies that were conducted to characterise the dynamic behaviour of check valves (Kruinsbrick A.C.H., 1996). The measurements results confirmed the quick reaction time of the instrument, with a frequency response slightly higher than 40Hz.

The objective of the experiments was to simulate the filling of a pipe that is partially full of water with an air pocket at one end. Three different water column lengths, having initial water levels of 1238, 1702 and 1988 mm with respect the axis of the horizontal pipe, were used. At this initial stage, the air valve was open and the air inside the pipe and the tank was under atmospheric pressure conditions. When the pressure in the tank with water increased (from atmospheric pressure to a controlled pressure P_{ini}), the water column moved to the top end of the pipe. In its movement, the water column pushed the air pocket located at its end. Because of the fast response of the air admission valve (butterfly valve 2) installed between the high capacity air vessel and the tank with water, pressurisation was almost instantaneous. In order to attain several water column accelerations, various differential pressure values between the two tanks, from 0.15 to 0.30 bar, were used in the experiments. The large size of the air vessel (70 m³) made it possible to keep the pressure in the upstream tank almost constant throughout the experiments. When the moving water column reached the top end of the pipe, the air valve float moved up, and its closing produced a fast change in water velocity with an important rise in pressure. At this final stage, the pipe was almost full of water with small air pockets.

The movement of water and air generated the transient shown by the time history of the pressure (registered by the transducer located next to the air valve), the water column velocity and the displacement of the air valve float measured by a displacement transducer attached to it (Fig. 2). In this figure, P_m is the maximum pressure reached in the system while t_m is the instant when it occurs; t_z is the time when the water column velocity is zero. In practice, t_m and t_z are almost equal. Still related with the pressure wave, t_p is the moment when the pressure begins to rise after applying the initial pressure, P_{inh} , from the high capacity air vessel. Lastly, t_{p0} is the time when the compression of the air pocket ends. Two additional times linked to the movement of the air valve float, t_o and t_c , are presented in Fig. 2. These times indicate the instants when the float starts its movement and when the air valve is completely closed. The volume of entrapped air, when the air valve starts closing, can be measured indirectly, with an acceptable degree of uncertainty, by integrating the EMF measurements.

NUMERICAL MODEL

The system of equations that describes the phenomena models the behaviour of both fluids inside the pipe. On the one hand, it provides the mass and momentum conservation equations of the water column and, on the other hand, it also includes the air pocket compression-expansion equation. Boundary and initial conditions are also required to solve the system of equations.

Basic hypothesis

The numerical model assumes that:

- The water column pushes the air pocket in its upward movement. The water-air interface occupies the whole section. It is flat and normal to the axis of the pipe.
- A perfect gas evolution is assumed.
- The wall friction factor and wave celerity are constant, both in time and space.

- Fluid–structure interaction effects are negligible.
- The initial pressure applied to the water column (P_{ini}), is constant.
- Predicting the closing time of the air valve by means of a simplified model like the one used in the present work is unfeasible. For this reason, the air valve closing time is an input parameter of the model and not a consequence of the forces exerted on the float by the two-phase fluid.
- The air valve starts closing when the entrapped air becomes smaller than a predefined volume. For the test facility, this critical volume was typically smaller than 0.08 m^3 . Its value was estimated during the tests using the EMF data. The final volume of entrapped air depends on this critical volume and the air valve closing time.
- A linear evolution of the discharged coefficient with time is assumed (once the air valve starts closing).
- The system of equations is solved using the Method of Characteristics, with a mobile boundary condition (Abreu et al., 1992).

The mathematical model presented provides satisfactory results for positive fluid flow velocities. However, once the water column acquires a negative velocity and returns to the water tank, the entrapped air abruptly begins to expand. From this moment, the hypothesis of a water-air interface which is flat and normal to the pipe axis is not met anymore. Water and air are mixed unpredictably and several of the preceding hypotheses cannot be used to accurately reproduce the system behaviour.

Governing equations

The behaviour of a one-dimensional water column can be described by the set of equations proposed by Chaudhry (1987) for a simplified elastic model.

$$\frac{g}{a^2} \frac{dH}{dt} + \frac{\partial V}{\partial x} + \frac{g}{a} V \sin \alpha = 0 \quad (1)$$

$$\frac{dV}{dt} + f \cdot \frac{V \cdot |V|}{2D} + g \cdot \frac{\partial H}{\partial x} = 0 \quad (2)$$

H being the piezometric head, V the water column velocity, a the fluid celerity, g the gravity acceleration, f the friction factor, and D is the diameter of the pipe.

The air pocket evolution (Martin, 1976) is characterised by:

$$\frac{dH^*}{dt} = -k \cdot \frac{H^*}{\nabla_a} \cdot \frac{d\nabla_a}{dt} - k \cdot \frac{H^*}{\nabla_a} \cdot Q_a \quad (3)$$

where H^* is the absolute piezometric head of the air pocket, Q_a is the volumetric air flow discharge through the air valve orifice, ∇_a is the air volume and k represents the polytropic coefficient. Changes in the volume of the air pocket are calculated according to the water column velocity at each instant.

For subsonic conditions ($H^*/H_0^* < 1.89$), the volumetric flow discharge through the air valve is calculated as:

$$Q_a = C_d \cdot A_0 \cdot Y \cdot \sqrt{2 \cdot g \cdot \frac{\rho}{\rho_a} \cdot (H^* - H_0^*)} \quad (4)$$

where A_0 is the orifice area of the air valve, C_d is the discharge coefficient, Y is the expansion factor of the air flow through the orifice, ρ is the water density, ρ_a is the air density at the pipe conditions and H_0^* is the downstream air pressure, in this case, atmospheric pressure. The discharge coefficient of the air valve can either be determined experimentally or estimated using tables and equations from the technical literature (AWWA, 2001) or from standards such as ISO 5167-2:2003.

According to Martin (1976), the expansion factor can be expressed as:

$$Y = \sqrt{\frac{n}{n-1} \cdot \left(\frac{H_0^*}{H^*}\right)^{2/n} \cdot \frac{1 - \left(\frac{H_0^*}{H^*}\right)^{(n-1)/n}}{1 - \frac{H_0^*}{H^*}}} \quad (5)$$

The parameter n , adopts a value of 1.4 for dry air, and represents the isentropic coefficient for compressible flow through an orifice. This coefficient is different from the polytropic coefficient

k, used in Eq. 3, which describes the behaviour of the air pocket expansion/compression process. For sonic conditions ($H^*/H_0^* > 1.89$ for dry air) the orifice is choked and the volumetric flow discharge remains constant, regardless of the internal pressure head of the air inside the pipe.

$$Q_a = C_d \cdot A_0 \cdot \sqrt{\left(g \cdot \frac{\rho}{\rho_a} \cdot H^*\right) \cdot \left[n \cdot \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]} \quad (6)$$

Initial conditions

The instant before the transient starts, the water velocity is zero, the air valve is open and the pressure of the air entrapped in the pipe between the air valve and the water column is under atmospheric conditions.

Boundary conditions

The piezometric head in the water tank is assumed to be constant. Although the volume of the upstream vessel is considerable, this condition was not strictly met and small variations in the piezometric head at the inlet of the pipe, due to oscillations in the water column, were measured. The pressure and the initial location of the water-air interface along the pipe is also assumed to be constant. The air pressure is defined by Eq. 3, while the position of the interface is recalculated at every iteration, taking into account the water velocity at the previous instant.

MODEL CALIBRATION

Empirical determination of some of the parameters used in the model, like the average friction factor or the polytropic coefficient, is unfeasible in the laboratory and therefore the uncertainty

about which values to use becomes too large. The model needs to be calibrated to obtain the set of numerical values of the parameters that best reproduce the laboratory tests. This was an arduous and sometimes complicated process that was conducted for all the tests – more than thirty for each air valve. It was carried out by assigning five values to each parameter of the model. Combining the different values of the parameters meant that each test was simulated more than fifteen thousand times. An error function was defined to establish the values that best fitted the experimental data. The mean squared error (MSE) between simulated and measured values (for flow velocity and air pressure at the high point of the facility) within the time interval defined by t_p and t_{p0} (see Fig. 2) was calculated for each simulation. The set of parameters providing the minimum MSE was selected as a reference for the test. Additional details of this analysis can be found in Carlos (2007).

Fig. 3 presents the set of parameters that provide the best adjustment for one of the tests analysed. The differential pressure between the air vessel and the tank with water for the test analysed is 0.23 bar. The initial water level in the tank with respect the axis of the horizontal pipe is 1238 mm. Pressure at the air valve and velocity are plotted from t_i to t_{p0} .

SENSITIVITY ANALYSIS

A sensitivity analysis was also conducted to evaluate the contribution of each parameter in the transient results. The aim of the analysis is to show how pressure and flow velocity during the transient are affected by a controlled variation of a single parameter around the value that provided the best fit in the numerical model. For the sake of clarity, only one test (the one presented in Fig. 2) is used. Nevertheless, the same conclusions can be reached from the remaining tests.

The results are presented from less to greater influence. This means that the first are those linked to the characteristics of the pipe–fluids and then the parameters that characterise the dynamic behaviour of the air valve.

Fluids–pipe parameters

Polytropic coefficient

The polytropic coefficient k determines the type of air evolution during the compression and expansion process. The polytropic coefficient used in the optimal fit (Fig. 3) was $k = 1.2$. The explored values correspond to the isothermal ($k = 1$) and adiabatic ($k = 1.4$) evolutions (Fig. 4). In the first case the transient is slightly delayed, whereas the opposite applies for the adiabatic case. Taking into account the time scale (tenths of a second), the time shift is negligible. The maximum pressure differences are also insignificant, although for the isothermal case the maximum pressure calculated is somewhat below the measured value, while the duration of the air compression-expansion process (ΔT_{c-e}) is slightly longer. Since the transient under study is very short and the pressures are relatively low, the value of this coefficient was not found to be relevant. For longer transients (i.e. the air chamber case), this coefficient plays a more important role (Graze, 1972).

Wave celerity

Pressure wave propagation depends on the characteristics of the fluid and the pipe. As the amount of air dissolved in the water increases and the modulus of elasticity of the pipe material decreases, the celerity becomes smaller. However, the numerical model presented in this work considers that the water-air interface is well defined and fluids are not mixed during the transient. For this reason the model assumes that wave celerity is constant in time and space.

As expected, and because of the high compressibility of the air pocket, wave celerity only becomes significant at the final stage, when pressure rises to a significant level. Nonetheless, even at this stage, the differences corresponding to the extreme values of wave celerity (Fig. 5) are almost negligible. This is consistent with other authors' formulations which use a rigid model to solve the first stage of a water transient with entrapped air (Zhou et al., 2002).

Friction factor

Due to the complexity of the dissipative effects in water transients, the friction factor, f , and its variation in time and space in hydraulic transients have received a great deal of attention over the last four decades (Zielke, 1968; Brunone et al., 1991; Abreu and Betamio de Almeida, 2009). In the case studied here (a fast transient involving two accelerating fluids) the complexity is even greater. Unfortunately, this parameter can have a major impact when its effects are accumulated in time. Since the main objective of this paper is to analyse the influence of an air valve on the water transient, attention is only paid to identifying the effect of an equivalent friction factor. The best fit for the test in Fig. 3 was obtained for $f = 0.03$. This value is larger than the one derived from the Colebrook-White equation for the average velocity during the transient, which confirms the fact that dissipative effects in water transients are higher than those corresponding to steady flow. Nonetheless, it is important to underline the great influence of this parameter. With other values ($f = 0.02$ and $f = 0.05$), results are quite different (Fig. 6). For the lower value, the model predicts that the water column reaches the air valve earlier than in the experiments, while the peak pressure is higher. And, indeed, the opposite applies when $f = 0.05$ is considered.

Air valve behaviour parameters

Discharge coefficient

This is a key parameter for evaluating the air release flow [Eqs. (7) and (9)]. For the air valve used in the experiments there is little uncertainty about its value because it was experimentally determined under steady flow conditions, obtaining a mean value of 0.71 (Arregui F.J. et al., 2003). The good agreement between the experimental and the theoretical results (Fig. 7) confirms the accuracy of the measurement. As seen in Fig. 7, the results obtained with extreme C_d values (0.61 and 0.81) confirm a well-known fact, i.e. that any action slowing down the water column (for example a small C_d value) reduces the impact velocity and hence the pressure peak amplitude. And again the opposite applies for a bigger C_d value. Fig. 7 shows these differences, which are particularly significant from the initial instant, t_i , until t_p , which is the time when the pressure starts to increase. In conclusion, the control of the water column

velocity is crucial to avoid high overpressures, and this objective can be achieved by selecting valves with adequate C_d values.

Entrapped air volume

The amount of entrapped air that remains in the system depends on several parameters. Some of them are related to the design of the air valve (the float size, weight and shape), while others are related to the layout of the pipeline and magnitude of the water transient. The first affect the closing behaviour of the valve once the water column has arrived. Consequently, for the same installation site and transient conditions, different valves may produce dissimilar results, as the volumes of air that remain inside the pipe can vary. While some air valves close shortly after the first drops of water reach the valve, others expel significant volumes of water before the closure takes place (Fig. 8).

The transient magnitude can also affect the amount of air in the system before the air valve starts closing. An estimation of this air volume can be obtained for the laboratory tests using the measures from the EMF. Fig. 9 summarises the results for three types of tests (for the same air valve, but with different initial water levels) and several initial pressures applied to the water column (P_{ini}). For the facility used, there is an inverse correlation between the velocity at which the water column impacts the air valve and the volume of air entrapped in the system when this happens. This may be an indication that the mixing of water and air also depend on the magnitude of the transient.

From the sensitivity analysis it can be shown that the amount of air that remains inside the system plays a critical role in the maximum pressure generated during the transient. Any modification in this parameter changes the results dramatically. With the smaller value of the sensitivity analysis, 0.028 m^3 , the air pocket compression–expansion is significantly shorter than the one measured in the test, while the maximum pressure and the water column deceleration are much higher (Fig. 10). And, as expected, the opposite applies if a larger amount of trapped air (0.048 m^3) is considered.

Air valve closing time

The numerical model used here assumes that the air valve starts closing when the volume of air inside the system falls below a predefined value that can be estimated from the test measurements. From that point on, the discharge coefficient of the air valve shows a linear reduction with time until it is fully closed. Therefore, during the closing event, additional air volumes are expelled from the pipe. As a result, longer closing times will lead to smaller volumes of entrapped air.

Two additional air valve closing times (0.02 s and 0.06 s) are simulated. The faster closing time leads to a larger amount of trapped air within the system after the air valve has completely closed. This is reflected in a longer duration of the compression–expansion process. Furthermore, the greater cushioning effect of the air pocket reduces the maximum pressure (Fig. 11). The results obtained for a longer closing time ($t_c = 0.06$ s) show just the opposite behaviour with regard to the compression-expansion process and the maximum pressure value. Furthermore, a greater water column deceleration rate is observed due to the smaller amount of air remaining inside the pipe.

DISCUSSION OF THE RESULTS

The work presented shows how a simplified model can accurately describe a water transient with entrapped air if appropriate values are correctly identified for the main parameters involved. Nonetheless, in practice, the numerical values of several of the parameters used in the model cannot be measured or even estimated with the required degree of confidence.

Consequently, calibration of the model using experimental data was required. This calibration process provided, for each test, the combination of values that produced the best fit between the model and the laboratory data. Afterwards, a sensitivity analysis in which every parameter was changed one at a time was conducted for one of the tests. This methodology allowed for the assessment of the individual relevance of each parameter.

In order to simplify the analysis, the parameters of the model were classified into two groups. The first one (k , a and f) was connected to both fluids (water and air) and to the pipe. The second group was associated with the air valve behaviour (C_d , V_{air} and t_c). Two of the parameters in the first group (k and a) were found to exert little influence (see Fig. 4 and 5), while the last one, f , was a key factor in the process since the water column velocity, just before impacting the air valve, very much depended on its value (Fig. 6).

The arrival velocity of the water column was also affected by C_d (Fig. 7). A smaller value of this parameter is equivalent to a larger friction factor or a lower driving action represented by P_{ini} . Both effects reduce the water column velocity and, subsequently, the pressure peak magnitude. This is the reason why air discharge operations require the selection of air valves with the lowest possible air discharge capacity. Oversized air valves will very likely lead to large water column velocities yielding dangerous pressure peaks.

The amount of air remaining in the system was also found to exert a critical effect on the overpressure magnitude. As a general rule, the greater the volume of entrapped air the larger the cushioning effect. However, the volume of residual air in the system is difficult to control and depends not only on the dynamic behaviour of the air valve but also on the pipeline layout and the magnitude of the water transient.

As a result, the closing time of the air valve only plays a significant role in short systems like the one used for the tests, where a closing time of 20 or 40 ms appreciably changes the relative amount of air that remains entrapped in the pipe after the air valve has closed completely. For longer pipeline systems, a faster or slower closing time (within certain limits) will not change the air volume in relative terms. In those cases, the amount of air entrapped in the system will instead depend on the pipeline layout or the water–air mixing process during the water column movement and not on the closing dynamics of the air valve. Therefore, in most real systems, the closing time of the air valves will not be a critical factor for the pressure peak magnitude.

All the preceding comments are summarized in Fig. 12, in which the relative change of the parameters is plotted against the relative change of the pressure peak magnitude. The larger

the slope of the curve the greater the influence of the parameter on the maximum pressure generated in the transient. For the system studied and the variation range of the parameters considered all the curves are monotonic. However, as discussed in Arregui et al. (2010) this may not always be the case.

All in all, these results confirm that in order to minimise the negative effects that entrapped air may have in the system when restarting it, it is essential to proceed with extreme care. If this is the case, water velocity in the pipes can be kept to low values and entrapped air volumes will be large enough to provide an adequate cushioning effect. Moreover, selecting air valves with lower discharge coefficients and smaller orifice areas will help to mitigate the maximum pressure peak that may appear in the system. By following this sizing criterion, entrapped air volumes will always be larger and the velocity of the approaching water column smaller, due to a higher counter-pressure.

Lastly, assuming that air valves work as expected, the most dangerous situations will be those in which small uncontrolled volumes of air remain inside the pipe because they cannot be removed properly through an air valve. This will very likely lead to pipe bursts. In contrast, the presence of large volumes of air at specific locations cannot be considered undesirable as long as they can be released slowly in a controlled manner. Large volumes of entrapped air will significantly reduce the magnitude of the water transients. Thus, a correct design of the pipeline layout should help by guiding the air pockets to specific locations where they can be released afterwards.

CONCLUSIONS

Hydraulic transient analysis with entrapped air in pressurized pipes on real field conditions is a very complex exercise. In first place, the initial conditions of the system - like the initial volume of entrapped air in the pipes, the number and length of isolated water columns, etc. - are not usually known. Secondly, the physical behaviour of the system, involving two fluids in different phases and hydraulic devices with moving elements, is extremely difficult to be described by a

resoluble numerical model. Finally, some of the parameters needed to solve the model cannot be measured in the field and in case this is possible, the accuracy of the measurement would not reach the required level.

The main achievement of the work presented has been to accurately reproduce by means of a relatively simple numerical model the measurements taken in a large scale laboratory setup. Unfortunately, it must be underlined that in practice such configuration (pipe having a significant slope, air valve installed at the dead end of the duct) it is not very common. Even worse, there are numerous air valve designs that can be installed in many different ways that will modify the system behaviour. Consequently, the results obtained for this particular case, which are correct from a quantitative and qualitatively perspectives, can only be qualitatively extrapolated for other layouts. Nevertheless, this may not be a minor finding, mainly because some of the conclusions are counter intuitive.

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List of figures

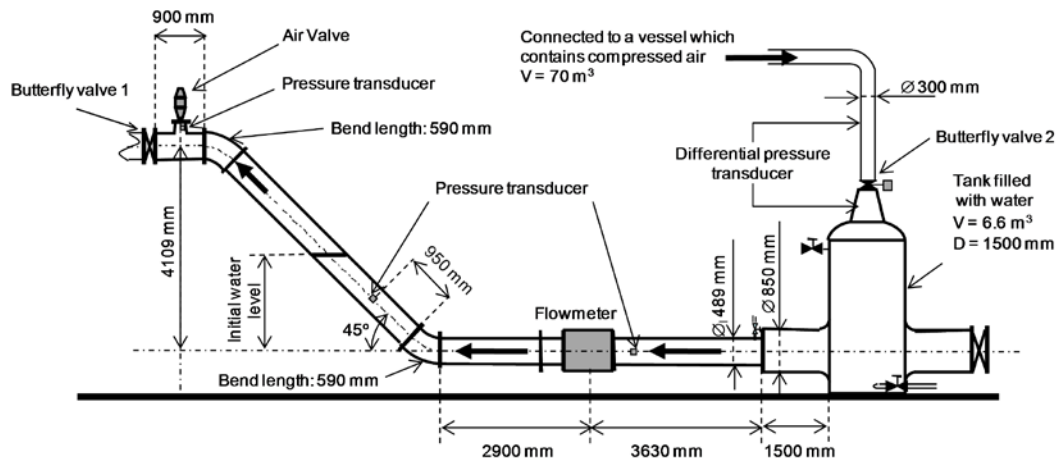


Figure 1. Delft Hydraulics facility scheme

Figure 2. Pressure and velocity evolution in an air expulsion test

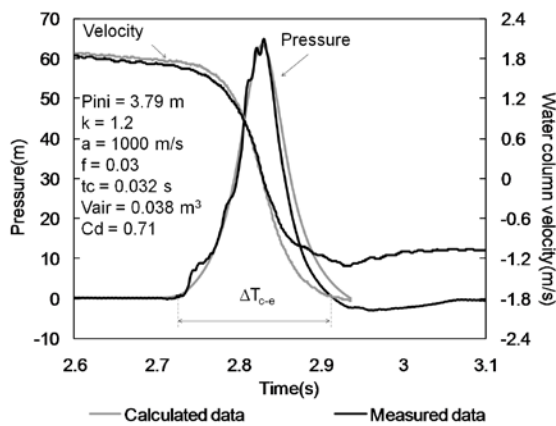


Figure 3. Experimental vs. simulation results after calibration of the model

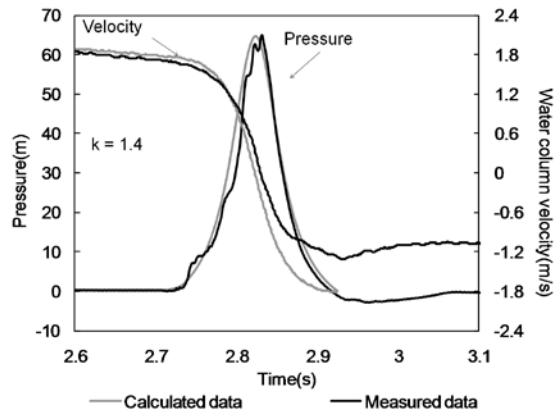
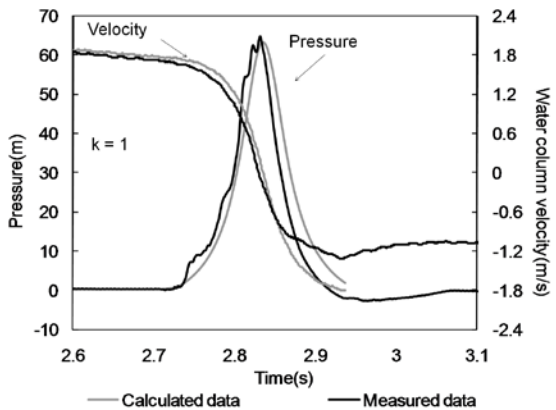


Figure 4. Effect of the polytropic coefficient on the transient simulation

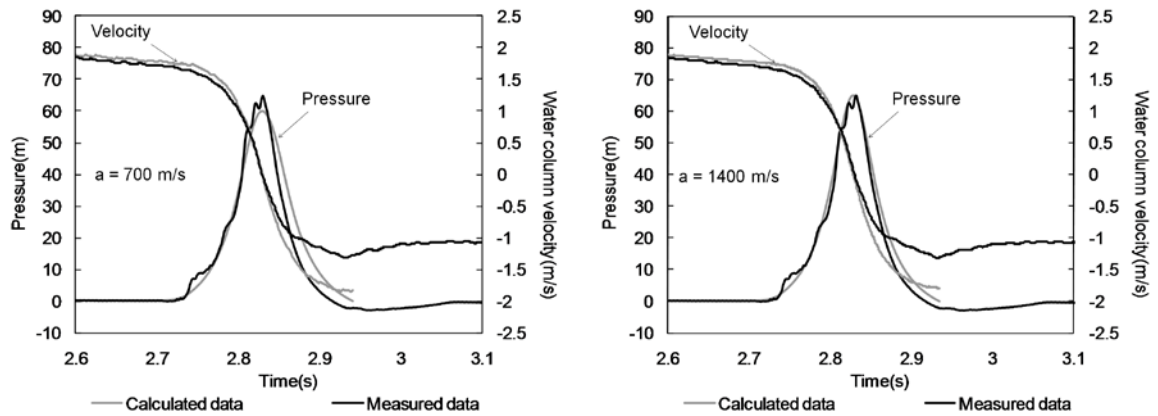


Figure 5. Effect of wave celerity on the transient simulation

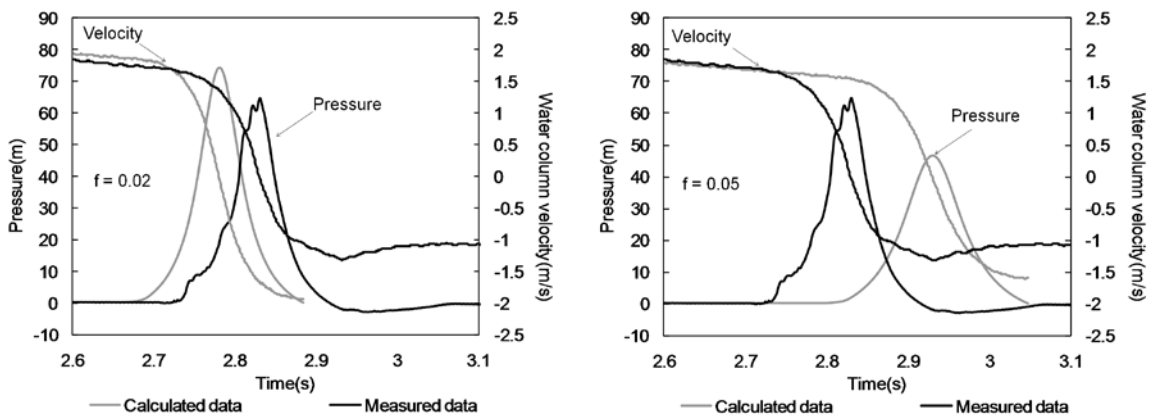


Figure 6. Effect of the friction factor on the transient simulation

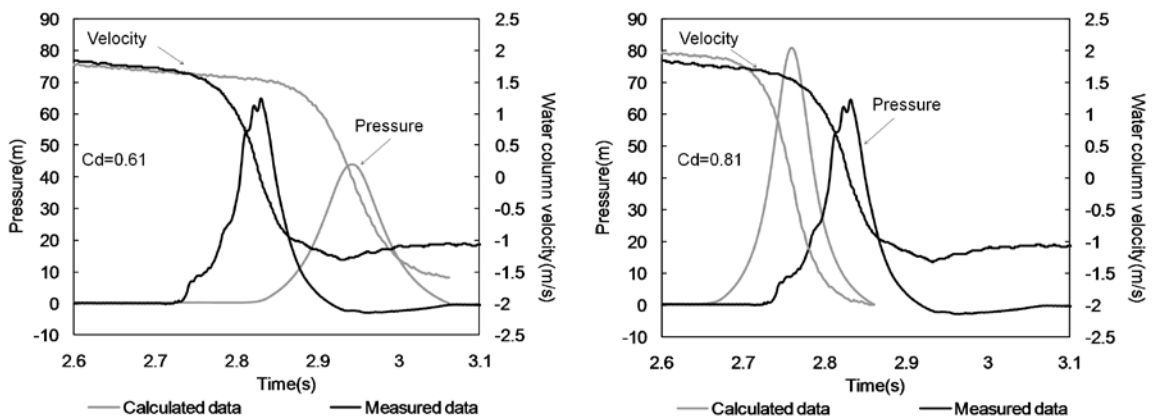


Figure 7. Effect of the discharge coefficient on the transient simulation

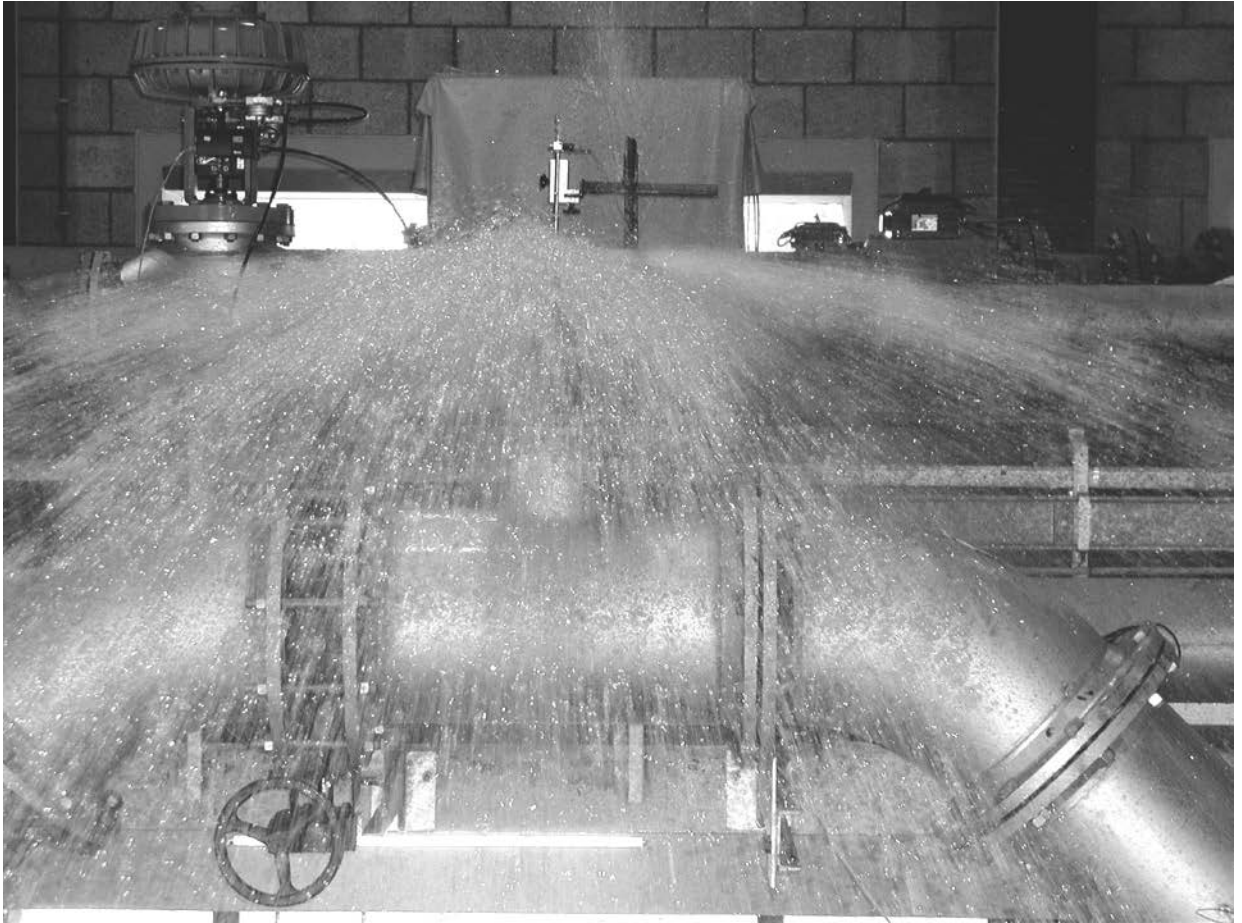


Figure 8. Water discharge through an air valve

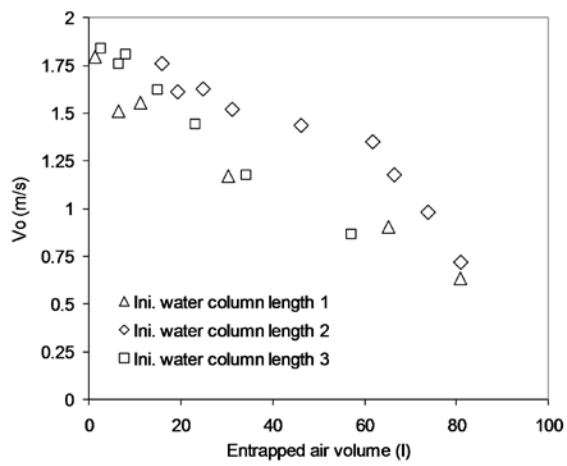


Figure 9. Measured entrapped air volume vs. impacting water column velocity

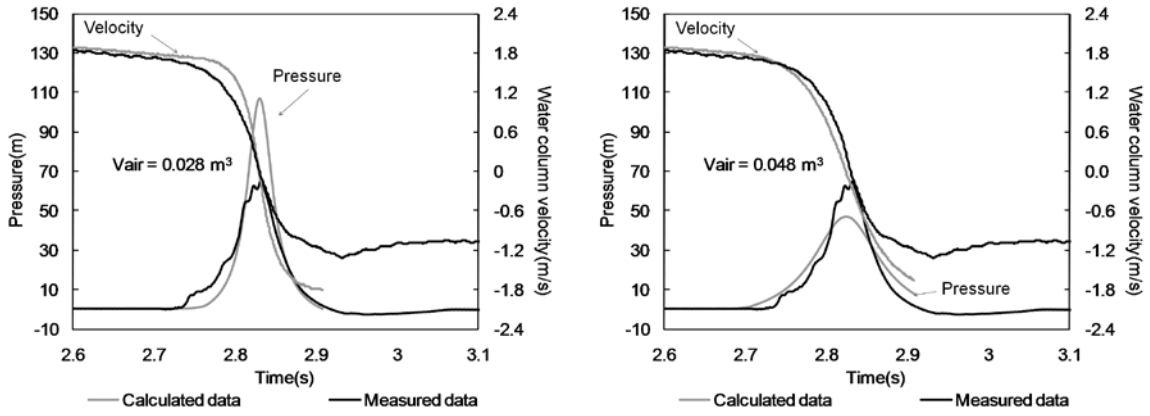


Figure 10. Effect of the entrapped volume of air on the transient simulation

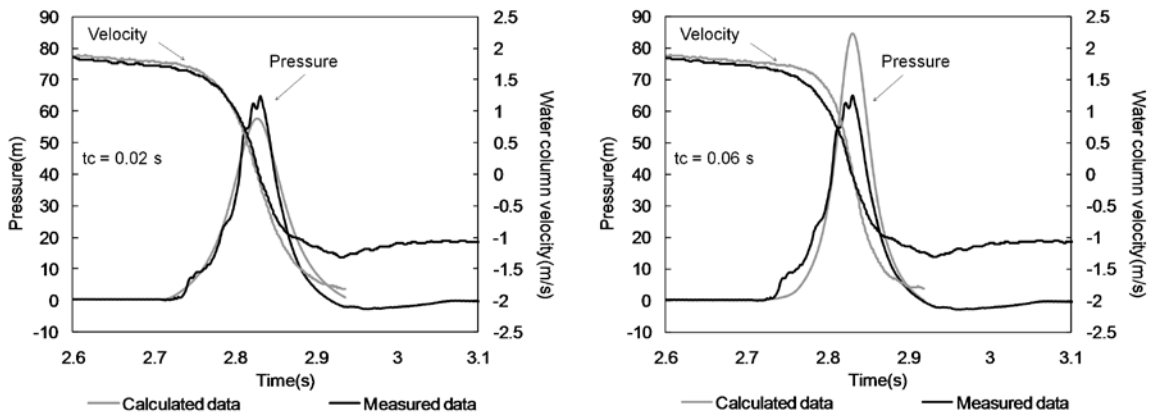


Figure 11. Effect of the closing time of the air valve on the transient simulation

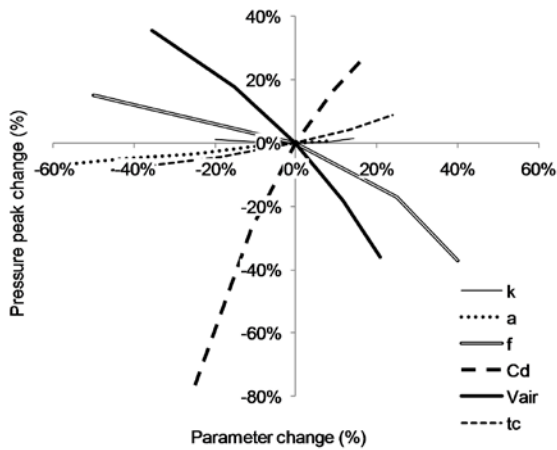


Figure 12. Sensitivity analysis of the parameters involved in the water transient