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Numerical modelling of pipelines with air pockets and air valves

VICENTE S. FUERTES-MIQUEL, Associate Professor, Departamento de Ingeniería
Hidráulica y Medio Ambiente, Universitat Politècnica de València, Camino de Vera s/n,
46022 Valencia, Spain

Email: vfuentes@upv.es (Corresponding Author)

P. AMPARO LÓPEZ-JIMÉNEZ, Associate Professor, Departamento de Ingeniería Hidráulica
y Medio Ambiente, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia,
Spain

Email: palopez@upv.es

F. JAVIER MARTÍNEZ-SOLANO, Associate Professor, Departamento de Ingeniería
Hidráulica y Medio Ambiente, Universitat Politècnica de València, Camino de Vera s/n,
46022 Valencia, Spain

Email: jmsolano@upv.es

GONZALO LÓPEZ-PATIÑO, Associate Professor, Departamento de Ingeniería Hidráulica y
Medio Ambiente, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia,
Spain

Email: glpatin@upv.es

Numerical modelling of pipelines with air pockets and air valves

Abstract

This work considers the behaviour of air inside pipes when the air is expelled through air valves. Generally, the air shows isothermal behaviour. Nevertheless, when the transient is very fast, it shows adiabatic behaviour. In a real installation, an intermediate evolution between these two extreme conditions occurs. Thus, it is verified that the results vary significantly depending on the hypothesis adopted. To determine the pressure of the air pocket, the most unfavourable hypothesis (isothermal behaviour) is typically adopted. Nevertheless, from the perspective of the water hammer that takes place when the water column arrives at the air valve and abruptly closes, the most unfavourable hypothesis is the opposite (adiabatic behaviour). In this case, the residual velocity with which the water arrives at the air valve is higher, and, consequently, the water hammer generated is greater.

Key words: air pockets, air valves, flows in pipes and nozzles, mathematical models, pressurised flow, transient flow, water pipelines.

Introduction

There are many causes that lead to the presence of air inside pipelines; they include filling and emptying the system, transient interruptions in the supply, vortex creation in the pump groups, depressurisation occurring in the air valves due to transient operations, deliverance of dissolved air, etc. Whatever the origin of air in the pipes, its presence creates unwanted problems in the majority of cases. One of the most important problems is overpressure generation (Martin 1976; Hamam and McCorquodale 1982; Fuertes 2001; Zhou *et al.* 2002; Ramezani *et al.* 2015). Other problems include section reduction inside the pipe, which may cause collapse; additional generation of head losses, which cause an increase in the electricity consumption; problems with noise and vibrations; interior corrosion due to the oxygen transported by the air; failures in the measurement instrumentation; etc.

In water distribution networks under pressure, the installation of air valves is necessary. Nevertheless, air valves are not always a guarantee of good system performance. They can cause some problematic circumstances. An inappropriate valve selection, an inaccurate performance measurement, or inadequate maintenance may cause important system problems (Fuertes 2001).

Air valves allow air to enter when the pipe becomes depressurised (thus preventing collapse), and they facilitate the expulsion of air (preventing the appearance of high overpressures). Of the two situations, the phase of air expulsion is the most critical. When the air valve expels air, it must allow a volume of outflow that is sufficient for the pressure of the air inside the pipe not to exceed a certain value. The exit must be controlled (the outflow does not have to be excessive) because in very fast expulsions, the water column advances at a high velocity, and when it arrives at the position of the air valve and the abrupt closing of the valve takes place, the water hammer that is generated can be very problematic (Campbell 1983; Funk *et al.* 1992; Stephenson 1997; Leow and Lee 1998; Zhou *et al.* 2002; Fuertes *et al.* 2009; Pozos *et al.* 2010a).

The classic texts (Chaudhry 1987; Wylie and Streeter 1993) assume isothermal behaviour for the air inside the pipe. Nevertheless, to calculate the amount of air that leaves through the air valve, they consider the expressions of the isentropic flow, characteristic of a fast transient without heat exchange (adiabatic behaviour).

When air exits through the air valve, the required time for the air to cross the air valve is too short to allow significant heat transfer. Thus, the hypothesis (adiabatic evolution) seems quite suitable. Nevertheless, to analyse the behaviour of the air inside the pipe, the validity of the hypothesis that is normally used (isothermal evolution) is not so evident. The transient velocity can determine the behaviour of the air inside the pipe; thus, its evolution can be adiabatic (quickly transient), isothermal (slower transient), or any intermediate situation (this option is the closest to the reality). In fact, in real installations it is very difficult to find a perfectly isothermal or adiabatic process. In real installations it occurs temperature change and heat transfer. The modelling and representation of the compression of air in a pipeline system is the main goal of this work. The results presented show important differences that depend on the hypothesis considered.

The analysis of the evolution of the air inside pipes usually supposes isothermal behaviour, whereas the evolution of the air in other elements, for example, air vessels, has always been widely discussed. For air vessels, the analysis of the behaviour of the air has interested numerous researchers, who have considered all the possible options, including adiabatic behaviour, isothermal behaviour, and an intermediate situation, which is called polytropic evolution.

The behaviour of the air pockets in pipes has not been studied in depth. In this case, as already mentioned, the behaviour is usually considered to be isothermal. Nevertheless, some studies (Martin 1976; Izquierdo *et al.* 1999; Fuertes *et al.* 1999; Zhou *et al.* 2013a; Zhou *et al.* 2013b; Bouso and Fuamba 2014) have analysed the evolution of the air for different values of n . On the other hand, it has been demonstrated that major pressure peaks are reached after isothermal evolutions (Abreu *et al.* 1991; Fuertes 2001; Pozos *et al.* 2010a; Bouso and Fuamba 2014), although there are no criteria established for how fast the

hydraulic transient must be to neglect the heat exchange and consider adiabatic behaviour.

The experimental data available in the literature (Lee and Martin 1999; Fuertes *et al.* 2000; Fuertes 2001), given the reduced dimensions of the facilities where the measurements were made, correspond to very fast transients. In these cases, the exponent that best fits the experimental results is $n = 1.4$ (adiabatic behaviour).

Thus, some authors propose an isothermal evolution (Almeida and Koelle 1992; De Martino *et al.* 2008; Trindade and Vasconcelos 2013; Bousso and Fuamba, 2014), while other authors propose an adiabatic evolution (Lee and Martin 1999; Zhou *et al.* 2002; Zhou *et al.* 2004; Martins *et al.* 2009; Vasconcelos and Wright 2011; Liu *et al.* 2011; Zhou *et al.* 2011; Zhou *et al.* 2013a; Zhou *et al.* 2013b) or intermediate values such as $n = 1.2$ (Martin 1976; Jönsson 1985; Li and McCorquodale 1999; Pozos *et al.* 2010b; Carlos *et al.* 2011; Vasconcelos and Leite 2012) or $n = 1.3$ (Thorley and Spurrett 1989). In fact, the polytropic coefficient value depends on the thermal characteristics of the system and how fast the transient occurs and most likely varies during the transient.

Different models can be used to study transients with air pockets. The numerical model presented in this paper is the rigid column approach. This model was selected because it provides good estimates for the maximum and minimum pressures involving air compression. The good estimates are explained by the dominant effect of air pockets in determining the unsteady flow pressures with respect to other problem features, such as the pipe wall elasticity and water compressibility.

Liou and Hunt (1996) proposed a rigid column model of flow start-up in empty pipelines with undulating elevation profiles assuming a vertical interface between the air and water phases. They proposed a velocity-based criterion to justify the application of the rigid-column approach. Using the rigid water column theory and a compressible bubble concept, Li and McCorquodale (1999) developed a mathematical model to simulate pressure transients during the transition from free-surface to pressurised flow in a single sewer. Zhou *et al.* (2002, 2004) presented an experimental and numerical investigation on the description of the rapid filling of an empty horizontal pipe with limited ventilation. Similar to the approach

proposed by Martin (1976), the numerical model was constructed using a lumped inertia approach and assumed a vertical interface separating the advancing waterfront and the air that initially filled the pipe. Vasconcelos and Leite (2012) presented results of a combined experimental and numerical investigation of rapid filling flows in closed conduits in which sudden air pocket entrapment occurs. A two-phase lumped inertia flow model was proposed, and its results were compared with the experimental data.

Other authors use different model approaches, such as a model based on the Saint-Venant modified equations (Vasconcelos *et al.* 2006; Vasconcelos and Wright 2009; Trindade and Vasconcelos 2013), using finite volume models (León *et al.* 2008; León *et al.* 2010), or applying the method of characteristics (Martins *et al.* 2009; Pozos *et al.* 2010b; Carlos *et al.* 2011; Zhou *et al.* 2011; Zhou *et al.* 2013a; Zhou *et al.* 2013b). Each model has its strengths and limitations (Bousoo *et al.* 2013).

Martins *et al.* (2009) use the elastic model based on the method of characteristics, considering a moving liquid boundary, and the results are compared with those achieved with the rigid column model. Liu *et al.* (2011) develop a rigid plug elastic water model by applying an elastic water hammer to the majority of the water columns while using a rigid water column analysis to a small portion near the air-water interface. Zhou *et al.* (2011, 2013a, 2013b) propose an elastic water model with a reasonable simplification (the liquid inertia and energy loss of a tiny water column adjacent to the air-water interface can be ignored) to capture the air-water interface and avoid interpolation complexity. They compare the elastic and rigid models and conclude that the rigid model can get consistent results (except in some cases, such as small air pocket and column separation. Bousoo and Fuamba (2014) developed a two-phase numerical model, combining the method of characteristics for solving the free-surface flow conditions, the rigid column for the pressurized flow conditions, and the ideal gas law.

The goal of the present work was to evaluate and compare the greatest peak pressure generated in the process of air expulsion through an air valve for extreme behaviours (isothermal and adiabatic) of the air inside the pipe. The flow through the air valve was

considered adiabatic. By comparing the two options, it was verified that the results differed significantly depending on the hypothesis.

Furthermore, it is important to have the most realistic knowledge of air valve behaviour. The behaviour of an air valve is determined by its characteristic curve, that is to say, the relation between the mass flow expelled or admitted by the air valve and the existing pressure inside the pipe. To determine the characteristic curve of the air valve, diverse models can be used.

Mathematical model

The results derived from the extreme behaviours of the air inside the pipe were compared. An installation with an air pocket was considered (Fig. 1), and the air pocket must be evacuated through an air valve located at the edge of the pipe. For this analysis, the installation was simplified by assuming that there was a closed edge after the air valve.

Fig. 1. Analysed system.

The system was fed by a pump with a well-known characteristic curve (H_p-Q). To simulate slower or faster transients, different cases were analysed by varying the size of the air valve, the initial length of the water column and the opening time of the valve.

Generally, hydraulic transients with air pockets can be studied with the rigid column model. Because of the easy compressibility and low inertia of air, the water and pipe elastic effects can be neglected, and good results can be obtained. Thus, unless the size of air pockets is very small, the rigid column model can be used for this type of transient.

Because the movement of the liquid column can be studied without considering the elastic effects (Liu and Hunt 1996; Li and McCorquodale 1999; Fuertes 2001; Zhou *et al.* 2002; Vasconcelos and Leite 2012; Malekpour *et al.* 2016), the equations that model the behaviour of the system are as follows:

(a) Filling water column dynamics. Mass oscillation equation:

$$(1) \quad \frac{dv}{dt} = \frac{p_0^* - p_a^*}{\rho L} - g \frac{\Delta z}{L} - \frac{f v |v|}{2D}$$

where v = water column velocity, t = time, p_0^* = upstream pressure, p_a^* = air pocket pressure, ρ = water density, L = water column length, g = gravity factor, Δz = geometric head between the extremes of the water column, f = Darcy-Weisbach factor and D = pipe diameter (* indicates absolute pressure).

(b) Filling water column dynamics. Interface position:

$$(2) \quad L = L_0 + \int_0^t v dt \quad \left(\frac{dL}{dt} = v \right)$$

where L_0 = initial water column length.

(c) Mass balance of the air pocket:

$$(3) \quad m_a(t) = m_a(t_0) + \int_0^t \frac{dm_a}{dt} dt$$

where m_a = mass of the air inside the pipe and dm_a/dt = air mass flow rate.

(d) Air pocket behaviour (Martin 1976):

Case d1: Isothermal (no temperature variation)

$$(4) \quad p_a^* \forall = p_a^* (L_T - L) A = m_a R T_a$$

Case d2: Adiabatic (no heat transfer)

$$(5) \quad \frac{dp_a^*}{dt} = -k \frac{p_a^*}{\forall} \frac{d\forall}{dt} + \frac{p_a^*}{\forall} \frac{k}{\rho_a} \frac{dm_a}{dt}$$

where \forall = air volume, L_T = total length, A = pipe cross-sectional area, R = universal gas constant, T_a = air temperature, ρ_a = air density and k = specific heat ratio ($k = 1.4$ for air).

In an adiabatic process, when air is compressed, the pressure increases, temperature increases and vapour pressure increases (depends on the temperature). If at any moment of the process the pressure is lower than the vapour pressure, water vapour will be formed, and the model will no longer be valid. However, the results obtained up to that time will be valid.

- (e) Upstream boundary condition. A pump is the energy source (see Fig. 1), and thus, p_0^* is given by

$$(6) \quad H_R + \frac{p_{atm}^*}{g\rho} + H_P = \frac{p_0^*}{g\rho} + \frac{v^2}{2g} + \zeta \frac{v^2}{2g}$$

where H_R = reservoir elevation, p_{atm}^* = atmospheric pressure, H_P = pump head and ζ = valve loss coefficient.

- (f) Downstream boundary condition (Fuentes 2001):

Case f1: Subsonic air velocity through the valve ($p_{atm}^* < p_a^* < 1.893 \cdot p_{atm}^*$)

$$(7) \quad \frac{dm_a}{dt} = C_d A_v p_a^* \sqrt{\frac{7}{RT_a} \left[\left(\frac{p_{atm}^*}{p_a^*} \right)^{1.4286} - \left(\frac{p_{atm}^*}{p_a^*} \right)^{1.7143} \right]}$$

Case f2: Sonic air velocity through the valve ($p_a^* > 1.893 \cdot p_{atm}^*$)

$$(8) \quad \frac{dm_a}{dt} = C_d A_v \frac{0.6847}{\sqrt{RT_a}} p_a^*$$

where C_d = air valve discharge coefficient and A_v = air valve orifice area.

This set of equations determines six unknown variables, the motor and resistant pressures (p_0^* and p_a^*), the two characteristic variables of the liquid column (L and v), and the two variables that define the air pocket (ρ_a and m_a). Of the eight equations, four are common ((1), (2), (3) and (6)), while the other four ((4)-(5) and (7)-(8)) use an alternative form based on the hypothesis considered. Therefore, for any of the possible cases, there is a system with six independent equations and six unknown variables.

This mathematical model has been validated by measurements in an experimental setup similar to a rising main, with a pipe (diameter 18.8 mm and length 6.9 m) of irregular profile and a centrifugal pump that acts as the energy source that raises water from the suction tank up to the upstream tank (Fuertes *et al.* 2000; Fuertes 2001). On the other hand, this rigid water model has been compared with the elastic water model and experimental results (Zhou *et al.* 2011; Zhou *et al.* 2013a). Both models show fairly good overall agreement with experiments.

Air valve modelling: comparison between compressible and incompressible flow

For the analysis of systems with air pockets, isothermal behaviour of the air inside the pipe and adiabatic behaviour of the air flowing through the air valve are generally assumed. For the modelling of the air valve behaviour, it is common to make an analogy between the air flow through an air valve and isentropic flow in nozzles (Chaudhry 1987; Wylie and Streeter 1993; Fuertes 2001).

Fig. 2. Convergent nozzle.

The flow of any gas or steam in a nozzle is practically adiabatic because the required time for each element of fluid to pass through the nozzle is too short to allow much heat transfer. If the flow also does not have friction, the expansion of the fluid is isentropic. Thus, it is possible to determine the characteristic equations that model the operation of the air valves analytically.

Assuming the behaviour of the air is that of a perfect gas, considering isentropic evolution and neglecting the difference in elevation, the energy equation simplifies to the following form:

$$(9) \quad \frac{kR}{k-1} \cdot dT + v \cdot dv = 0$$

where k = specific heat ratio ($k=1.4$ for air), R = gas constant, T = temperature and v = velocity.

By integrating, this equation can be obtained:

$$(10) \quad v_2 = \sqrt{\frac{2k}{k-1} \frac{p_1^*}{\rho_1} \left[1 - \left(\frac{p_2^*}{p_1^*} \right)^{\frac{k-1}{k}} \right]} = \sqrt{\frac{2k}{k-1} RT_1 \left[1 - \left(\frac{p_2^*}{p_1^*} \right)^{\frac{k-1}{k}} \right]}$$

with v_2 = exit air velocity, p_1^* = entrance air absolute pressure, ρ_1 = entrance air density, p_2^* = exit air absolute pressure and T_1 = entrance air temperature.

Substituting in the continuity equation leads to

$$(11) \quad \frac{dm_a}{dt} = p_1^* A_2 \sqrt{\frac{2k}{k-1} \frac{1}{RT_1} \left[\left(\frac{p_2^*}{p_1^*} \right)^{\frac{2}{k}} - \left(\frac{p_2^*}{p_1^*} \right)^{\frac{k+1}{k}} \right]}$$

in which dm_a/dt = air mass flow rate and A_2 = exit section.

This expression provides values corresponding to an “upper bound” because it assumes that the isentropic evolution of the air is completely reversible. In fact, because of friction and turbulence, the process is not reversible, and the real mass flow is lower:

$$(12) \quad C < 1 \quad \rightarrow \quad \frac{dm_a}{dt} = C p_1^* A_2 \sqrt{\frac{2k}{k-1} \frac{1}{RT_1} \left[\left(\frac{p_2^*}{p_1^*} \right)^{\frac{2}{k}} - \left(\frac{p_2^*}{p_1^*} \right)^{\frac{k+1}{k}} \right]}$$

If C_d is the relation between the “real” mass flow and the “theoretical” mass flow expelled by the air valve (a value that must be obtained by means of laboratory testing) during expulsion, then

$$(13) \quad \left(\frac{dm_a}{dt} \right)_{\text{exp}} = C_d p_a^* A_v \sqrt{\frac{7}{RT_a} \left[\left(\frac{p_{am}^*}{p_a^*} \right)^{1.428571} - \left(\frac{p_{am}^*}{p_a^*} \right)^{1.714286} \right]}$$

with p_a^* = absolute pressure of the air inside the pipe, A_v = orifice section of the air valve and p_{am}^* = atmospheric absolute pressure.

The previous expression is valid only when the flow through the air valve is subsonic, i.e., when the absolute pressure inside the pipe is lower than 1.92 bar (gauge pressure 0.91 bar). When this pressure is exceeded, sonic conditions are reached in the outflow orifice, and the velocity of the air remains constant (the phenomenon known as a “sonic blockade”) because supersonic flow cannot exist in a convergent nozzle.

In this case, the volumetric flow remains constant (if the air temperature inside the pipe remains constant, which is usually adopted as a hypothesis), but the mass flow does not. The mass flow increases with increasing pressure in the pipeline, which also increases the air density. Thus, when the gauge pressure inside the pipe is greater than 0.91 bar, the expelled flow is given by the following expression:

$$(14) \quad \left(\frac{dm_a}{dt} \right)_{\text{exp}} = C_d A_v \frac{0.6847}{\sqrt{RT_a}} p_a^*$$

Consider now the assumption of incompressible flow. This hypothesis is valid for low pressures (Vasconcelos *et al.* 2009; Trindade and Vasconcelos 2013). In this case, neglecting the difference in elevation, the energy equation is simplified as follows:

$$(15) \quad \frac{dp^*}{\rho} + v \cdot dv = 0$$

By assuming incompressible flow to model the air behaviour in the nozzle ($\rho_{ref} =$ constant density) and neglecting the kinetic term in the inflow section,

$$(16) \quad v_2 = \sqrt{2 \frac{p_1^* - p_2^*}{\rho_{ref}}}$$

Substituting this equation into the continuity equation gives

$$(17) \quad \frac{dm_a}{dt} = \rho_{ref} A_2 v_2 = \rho_{ref} A_2 \sqrt{2 \frac{p_1^* - p_2^*}{\rho_{ref}}} = A_2 \sqrt{2(p_1^* - p_2^*) \rho_{ref}}$$

where $Q_a =$ air volumetric flow rate and $dm_a/dt =$ air mass flow rate.

As above, this expression provides values for an "upper bound" because it assumes that the isentropic evolution of air is fully reversible. In fact, the real mass flow rate is lower:

$$(18) \quad C < 1 \quad \rightarrow \quad \frac{dm_a}{dt} = CA_2 \sqrt{2(p_1^* - p_2^*)\rho_{ref}}$$

Considering the coefficient C_d for an air valve in the phase of expulsion, the expression for the expelled mass flow is

$$(19) \quad \left(\frac{dm_a}{dt} \right)_{\text{exp}} = C_d A_v \sqrt{2(p_a^* - p_{atm}^*)\rho_{ref}}$$

The density is assumed constant. The reference density can be the density of the inflow section (ρ_a = air density inside the pipe), the density of the outflow section (ρ_{atm} = air density in atmospheric conditions) or an average density ($\rho_{ref} = \frac{\rho_a + \rho_{atm}}{2}$).

In the graph in Fig. 3, Expression (13), which corresponds to a model of compressible flow in which the velocity in the entrance section is not considered, is compared with Expression (19), which corresponds to a model of incompressible flow in which the velocity in the inflow section is not considered, and the reference density is the density of the inflow section [1], the density of the outflow section [2] and the average density between the inflow and outflow sections [3].

Fig. 3. Comparison between the compressible and incompressible flows ($D_v = 50$ mm and coefficient $C_d = 0.7$).

The differences between the models are small. Given that the air valves used for the filling and emptying of conductions are usually designed for a relative pressure of 0.3 bar and that they work even with lower pressures, the results obtained with the models of compressible and incompressible flow can be considered equally valid.

In addition, all curves in the previous graph are shown with the same C_d coefficient. By definition, this coefficient is the relation between the “real” mass flow and the “theoretical” mass flow, and thus, it is different for each expression. Next, the laboratory results were analysed by testing an air valve, and the C_d coefficients for the different models were compared. In fact, an air valve with a nominal diameter of 50 mm was tested under steady flow conditions in the laboratory by the authors. Fig. 4 shows the experimental results and the best fit (air flow versus relative pressure).

Fig. 4. Laboratory test results of air valve DN50.

Using the data collected during the test, the C_d coefficients for each model were calculated (Table 1), and the corresponding curves are shown in Fig. 5.

Fig. 5. Comparison between the experimental results and the results of the different models (air valve DN50).

Each of the models presented agrees with the experimental results obtained in the laboratory rather well. Therefore, all of them are perfectly valid.

Case study and results

The installation shown in Fig. 1 was considered, where the most important characteristics of the system are included. Once the pump reaches its steady regime velocity, the transient begins with the opening of the valve (different manipulation times of the valve are considered). The variables of interest are the maximum pressure of the air pocket and the residual velocity of the water when the air is totally expelled. The value of this residual velocity is important because it determines the water hammer that is generated when the water column arrives at the edge of the pipe and the air valve is closed abruptly.

By varying some parameters of the system, different cases were studied. In particular, twelve cases are presented in this work:

- (a) Size of the air valve: diameters of 25 mm and 50 mm.
- (b) Initial length of the water column: 1 m and 750 m.
- (c) Opening time: instantaneous opening ($t_{open} = 0$) and linear opening within 5 and 20 seconds (linear opening refers to a linear increase of velocity).

For each of the twelve systems, both extreme behaviours of the air inside the pipe were considered (isothermal and adiabatic). Fig. 6 shows the evolution of the most significant variables (pressure of the air and velocity of the water) for the case of an air valve of 25 mm in diameter, an initial length of the water column of 750 m ($L_0 = 750$ m) and a linear opening of the valve within 5 seconds ($t_{open} = 5$ seconds).

Fig. 6. Evolution of the pressure and velocity corresponding to an air valve DN25, $L_0 = 750$ m and $t_{open} = 5$ seconds.

The results shown in Fig. 6 show significant differences between the two hypotheses (isothermal or adiabatic behaviour). The peak of the pressure under isothermal conditions (113.1 water column meter) is remarkably greater than the maximum pressure under adiabatic conditions (92.5 wcm). On the other hand, in isothermal conditions, the opposition of the air pocket to the advance of the water column is greater, and as a result, the residual velocity of the water and the water hammer when the air valve closes are smaller.

The hypothesis that is generally adopted (isothermal behaviour) is often assumed to be more conservative, but in fact, it is not. From the perspective of the maximum pressure of the air pocket during the phase of expulsion, the hypothesis of isothermal evolution gives rise to a higher upper pressure (113.1 wcm against 92.5 wcm). Nevertheless, for the water hammer that is generated by the residual velocity when the water column arrives at the air

valve and the air valve closes abruptly, the results are the opposite. In fact, supposing a velocity of the pressure wave of $a = 1000$ m/s, the calculated residual velocities (0.84 m/s in the isothermal case and 1.69 m/s in the adiabatic case) generate significant pressure peaks as follows: residual pressure 76.9 wcm + water hammer 85.6 wcm = maximum pressure 162.5 wcm (isothermal behaviour) and 79.5 wcm + 172.3 wcm = 251.8 wcm (adiabatic behaviour). Therefore, adiabatic behaviour is the most unfavourable (the peak of pressure in the air pocket is 92.5 wcm, but the maximum pressure is 251.8 wcm when the air valve closes).

Fig. 7. Evolution of the pressure and velocity corresponding to an air valve DN50, $L_0 = 1$ m and instantaneous opening.

Fig. 7 shows air pressure and water velocity for the case of an air valve DN50, an initial length of the water column of 1 m ($L_0 = 1$ m) and an instantaneous opening of the valve. Tables 2 to 5 show the most significant results for the cases analysed. The main conclusions derived from the analysis are as follows:

- The maximum pressure of the air pocket inside the pipe is always greater when isothermal behaviour is assumed, whereas the residual velocity of the liquid column is the opposite. Therefore, if only the pressure of the air pocket is considered, the isothermal case is the most unfavourable. On the other hand, if the water hammer when the air valve is closed is considered, adiabatic behaviour is more critical (which is not usually assumed).
- The pressure peaks of the air inside the pipe depend on the parameters analysed in this order: the air valve size, the hypothesis (isothermal or adiabatic behaviour), the air pocket size, and the time of valve manipulation.
- The residual velocity of the water column when it is at the position of the air valve and the valve is closed (which is the most critical factor) shows a similar tendency. In this case, the most influential factor is the air valve size.

- A different tendency is shown in the transient duration. Logically, the most influential variable is the initial length of the water column, followed by the air valve size and the time of valve manipulation. It is necessary to emphasise that the transient duration is practically unaffected by the behaviour (isothermal or adiabatic) assumed to model the evolution of the air inside the pipe.

The air valve size is very important. Sizing is very problematic when selecting air valves. If the air valve is too small, it is not able to expel enough air, which can cause excessive compression of the air pocket left inside the pipe. Thus, high-pressure peaks can be reached. If the air valve is too large, it leads to a high residual velocity, creating a large water hammer.

When the size of the air valve is reduced, diverse peaks of the pressure can appear (in Fig. 6, two peaks are observed) because the air valve is too small and is not able to expel a sufficient amount of air during the filling of the pipe. Consequently, inside the pipe, an entrapped air pocket remains, which is compressed and expands repeatedly while the air slowly exits to the outside.

For an air valve 25 mm in diameter and a velocity of the pressure wave of approximately 1000 m/s, the order of magnitude of the maximum pressure of the air pocket is similar to the water hammer induced by the residual velocity when the air valve closes. Nevertheless, when the air valve is greater (50 mm, in this example), the pressure generated by the water hammer is greater than that reached during the evolution of the air pocket inside the pipe. Therefore, for an air valve of a certain size, the most unfavourable hypothesis is to assume adiabatic behaviour for the air pocket. However, if the air valve is very small (it is not able to expel enough air, it is not suitable), the most unfavourable hypothesis is isothermal behaviour.

Thus, the behaviour, isothermal or adiabatic, of the air inside the pipe is not a trivial question. The evolution of the air (isothermal or adiabatic) determines the value of the

maximum pressure that is generated in the process of air expulsion during the filling of the pipe.

Given the transient duration (several minutes in this case study), it seems that both hypotheses are, a priori, possible. In any case, it must be clear that they are the values for extreme conditions. In reality, the behaviour of the air will follow an intermediate evolution, essentially depending on the transient velocity.

Conclusions

The presence of air in pipes is, in many cases, inevitable. Air is introduced into pipes for many different reasons, and it can cause various problems. A strong preventative solution is the installation of air valves.

However, the sizing and selection of air valves must be carefully considered. Oversized air valves in the expulsion phase may lead to undesirable pressure surges and high overpressures, which can damage the installation. Undersized air valves are also unacceptable because they cannot admit or expel the amount of air required. Thus, if an air valve is too small or too large, it can cause major problems. The difficulty of estimating the accuracy of the flow of air to admit or evacuate an air valve leads to enormous complexity, which must be considered for the correct selection of an air valve for a particular installation.

The flow through an air valve is usually assumed to be adiabatic because the required time for the fluid to cross the air valve is too short for large heat transfer to occur. In addition, generally, a model of compressible flow is used to study air valve behaviour. Nevertheless, the simple results that a model of incompressible flow provides are also perfectly valid when the work pressures are small (i.e., the air valve is operating in subsonic conditions). In fact, the work pressures of the air valves of large orifices used for the filling and emptying of conductions are very small; they typically operate with pressures in the pipe below 0.3 bar relative.

This work demonstrated the remarkable influence that the behaviour of the air (isothermal or adiabatic) has on the maximum pressure that a pipe installation must support.

The process is easy to understand. In adiabatic conditions, the greater heating of the air assumes a greater thermal energy than elastic energy. Consequently, the pressure of the air pocket is smaller, the opposition to the water column is smaller, the residual velocity is greater, and the water hammer induced by the abrupt closing of the air valve is greater. If the behaviour were isothermal, the process would be the opposite; it would increase the pressure of the air pocket, stop the water column, decrease the residual velocity, and reduce the water hammer.

Thus, in many installations the hypothesis that is usually adopted (isothermal behaviour) does not provide the maximum values of the pressure. The pressure of the air during its evolution inside the pipe is greater when isothermal behaviour is assumed. However, it is not the most unfavourable case. The critical situation causes the water hammer that takes place when the air valve is closed and, in this case, the adiabatic behaviour to provide greater peaks of the pressure.

It is good practice to fill the pipes very slowly. Under these conditions, it is a perfectly acceptable isothermal compression hypothesis. However, if the filling of pipes is faster, it is interesting to analyse the system behaviour with the hypothesis of adiabatic compression.

Finally, a suitable selection of the air valve size to reduce the maximum values of the pressure is important. If the air valve is too small, the air pocket will be compressed excessively, and the pressure will be very high. If the air valve is too large, the air pocket will be expelled without problems, but the velocity of the water column will be very high; when the air valve is closed, a very dangerous water hammer will be generated. The most adverse situations occur because of the shock of the water column against the air valve. However, real installations do not usually have a closed edge. The situation can be more or less critical depending on what is downstream of the air valve. If, for example, there is a water column of large dimensions downstream, the air pressure evolution until the closure of the air valve could be similar to the case of the system analysed here. However, the water hammer due to

air valve closure could be different. In any case, slow closing air valves can also be used, whose purpose is to minimise the water hammer when the air valve closes.

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List of symbols

- a = wave propagation velocity [m s^{-1}];
- A = pipe cross-sectional area [m^2];
- A_v = air valve orifice area [m^2];
- A_1 = entrance section [m^2];
- A_2 = exit section [m^2];
- C_d = air valve discharge coefficient [-];

- D = pipe diameter [m];
- D_v = air valve orifice diameter [m];
- f = Darcy-Weisbach factor [-];
- g = gravitational acceleration [m s^{-2}];
- H_p = pump head [m];
- H_R = reservoir elevation [m];
- k = specific heat ratio [-];
- L = water column length [m];
- L_T = total length [m];
- L_0 = initial water column length [m];
- m_a = mass of the air inside the pipe [kg];
- n = polytropic exponent, being $1 \leq n \leq k$ [-];
- p_a^* = air pocket absolute pressure [wcm];
- p_{am}^* = atmospheric absolute pressure [wcm];
- p_0^* = upstream absolute pressure [wcm];
- p_1^* = entrance air absolute pressure [wcm];
- p_2^* = exit air absolute pressure [wcm];
- Q = water volumetric flow rate [$\text{m}^3 \text{s}^{-1}$];
- Q_a = air volumetric flow rate [$\text{m}^3 \text{s}^{-1}$];
- R = gas constant [$\text{m}^4 \text{kg}^{-1} \text{°K}^{-1}$];
- t = time [s];
- t_{open} = valve opening time [s];
- T_a = air temperature [°K];
- T_1 = entrance air temperature [°K];
- T_2 = exit air temperature [°K];
- v = water column velocity [m s^{-1}];
- v_1 = entrance air velocity [m s^{-1}];
- v_2 = exit air velocity [m s^{-1}];

Δz = geometric head between the extremes of the water column [m];

ρ = water density [kg m^{-3}];

ρ_a = air density [kg m^{-3}];

ρ_{atm} = air density in atmospheric conditions [kg m^{-3}];

ρ_{ref} = reference density [kg m^{-3}];

ρ_1 = entrance air density [kg m^{-3}];

ρ_2 = exit air density [kg m^{-3}];

ζ = valve loss coefficient [-];

\forall = air volume [m^3].

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Table 1. Values of the coefficient C_d for different models.

| | |
|---|---------------|
| Compressible Flow | $C_d = 0.616$ |
| Incompressible Flow (atmospheric density) | $C_d = 0.610$ |
| Incompressible Flow (air density inside the pipe) | $C_d = 0.520$ |
| Incompressible Flow (average density) | $C_d = 0.562$ |

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Table 2. Main results for an air valve DN25 and an initial water length of 1 m.

| | | Opening time $t_{open} = 0$ | Opening time $t_{open} = 5$ s | Opening time $t_{open} = 20$ s |
|----------------------------|------------|--------------------------------|----------------------------------|-----------------------------------|
| Maximum pressure | Isothermal | 110.0 wcm | 109.4 wcm | 105.2 wcm |
| | Adiabatic | 106.1 wcm | 104.9 wcm | 98.5 wcm |
| Residual water velocity | Isothermal | 0.94 m/s | 0.94 m/s | 0.94 m/s |
| | Adiabatic | 1.68 m/s | 1.68 m/s | 1.67 m/s |
| Transient duration | Isothermal | 229.7 s | 275.2 s | 356.5 s |
| | Adiabatic | 214.7 s | 260.1 s | 341.4 s |
| Maximum air temperature | Isothermal | 15 °C | 15 °C | 15 °C |
| | Adiabatic | 287.3 °C | 285.5 °C | 275.6 °C |

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Table 3. Main results for an air valve DN25 and an initial water length of 750 m.

| | | Opening time $t_{open} = 0$ | Opening time $t_{open} = 5$ s | Opening time $t_{open} = 20$ s |
|----------------------------|------------|--------------------------------|----------------------------------|-----------------------------------|
| Maximum pressure | Isothermal | 116.0 wcm | 113.1 wcm | 101.0 wcm |
| | Adiabatic | 95.5 wcm | 92.5 wcm | 81.0 wcm |
| Residual water velocity | Isothermal | 0.83 m/s | 0.84 m/s | 0.95 m/s |
| | Adiabatic | 1.68 m/s | 1.69 m/s | 1.64 m/s |
| Transient duration | Isothermal | 109.9 s | 138.3 s | 195.7 s |
| | Adiabatic | 106.0 s | 134.3 s | 191.8 s |
| Maximum air temperature | Isothermal | 15 °C | 15 °C | 15 °C |
| | Adiabatic | 270.8 °C | 265.9 °C | 245.8 °C |

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Table 4. Main results for an air valve DN50 and an initial water length of 1 m.

| | | Opening time $t_{open} = 0$ | Opening time $t_{open} = 5$ s | Opening time $t_{open} = 20$ s |
|----------------------------|------------|--------------------------------|----------------------------------|-----------------------------------|
| Maximum pressure | Isothermal | 48.5 wcm | 47.2 wcm | 39.4 wcm |
| | Adiabatic | 37.9 wcm | 36.2 wcm | 30.1 wcm |
| Residual water velocity | Isothermal | 3.34 m/s | 3.35 m/s | 3.28 m/s |
| | Adiabatic | 3.93 m/s | 3.91 m/s | 3.66 m/s |
| Transient duration | Isothermal | 153.3 s | 199.8 s | 285.3 s |
| | Adiabatic | 152.4 s | 198.9 s | 284.4 s |
| Maximum air temperature | Isothermal | 15 °C | 15 °C | 15 °C |
| | Adiabatic | 144.6 °C | 139.0 °C | 118.0 °C |

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Table 5. Main results for an air valve DN50 and an initial water length of 750 m.

| | | Opening time $t_{open} = 0$ | Opening time $t_{open} = 5$ s | Opening time $t_{open} = 20$ s |
|----------------------------|------------|--------------------------------|----------------------------------|-----------------------------------|
| Maximum pressure | Isothermal | 49.7 wcm | 44.9 wcm | 29.4 wcm |
| | Adiabatic | 29.9 wcm | 27.5 wcm | 21.0 wcm |
| Residual water velocity | Isothermal | 3.56 m/s | 3.52 m/s | 3.13 m/s |
| | Adiabatic | 3.96 m/s | 3.85 m/s | 3.27 m/s |
| Transient duration | Isothermal | 82.0 s | 111.3 s | 171.6 s |
| | Adiabatic | 82.0 s | 111.3 s | 171.4 s |
| Maximum air temperature | Isothermal | 15 °C | 15 °C | 15 °C |
| | Adiabatic | 117.2 °C | 108.1 °C | 79.9 °C |

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Fig. 1. Analysed system.

Fig. 2. Convergent nozzle.

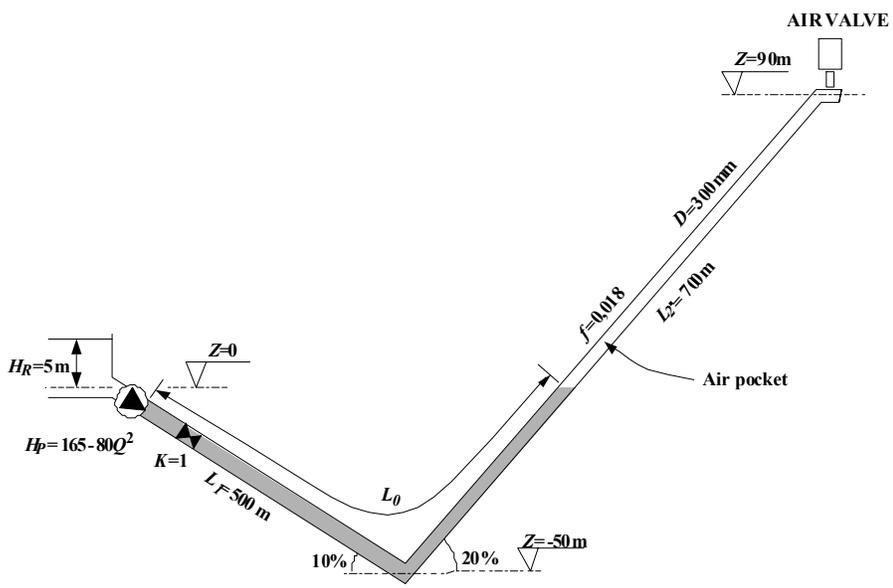
Fig. 3. Comparison between the compressible and incompressible flows ($D_v = 50$ mm and coefficient $C_d = 0.7$).

Fig. 4. Laboratory test results of air valve DN50.

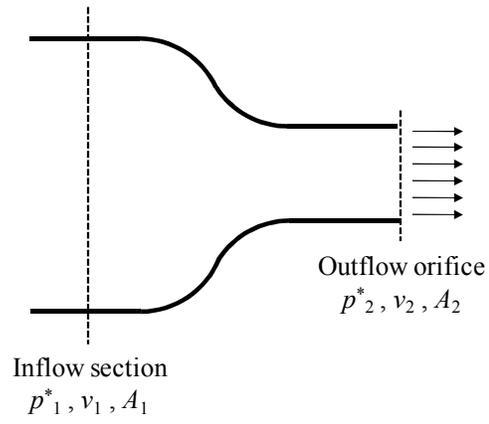
Fig. 5. Comparison between the experimental results and the results of the different models (air valve DN50).

Fig. 6. Evolution of the pressure and velocity corresponding to an air valve DN25, $L_0 = 750$ m and $t_{open} = 5$ seconds.

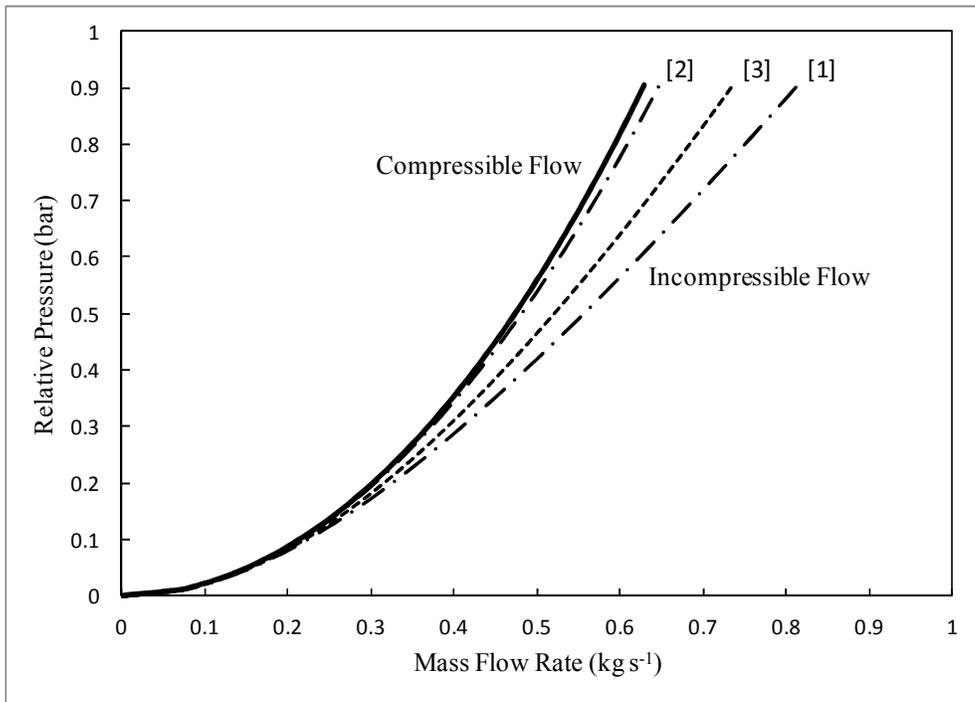
Fig. 7. Evolution of the pressure and velocity corresponding to an air valve DN50, $L_0 = 1$ m and instantaneous opening.



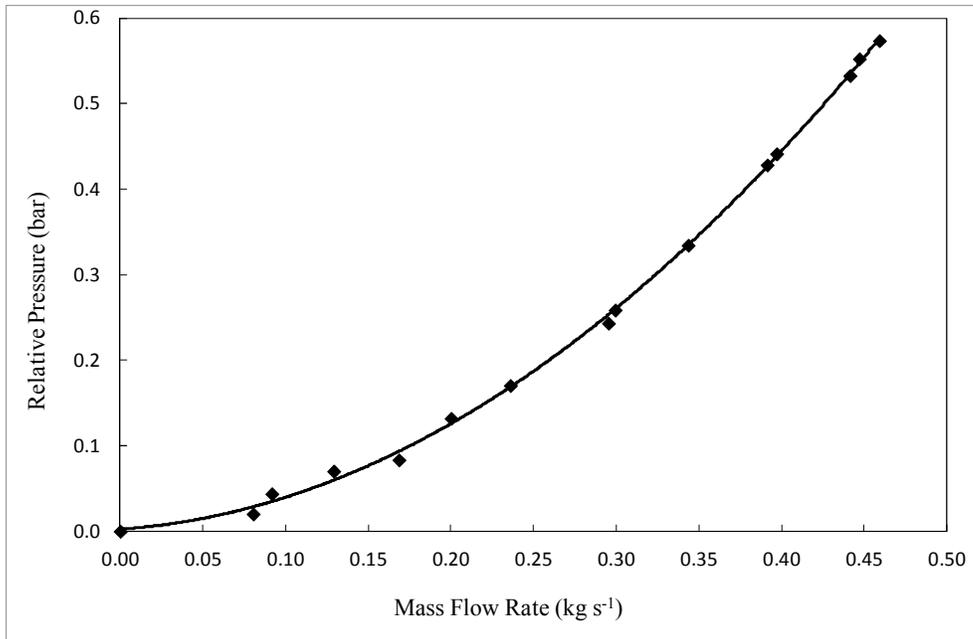
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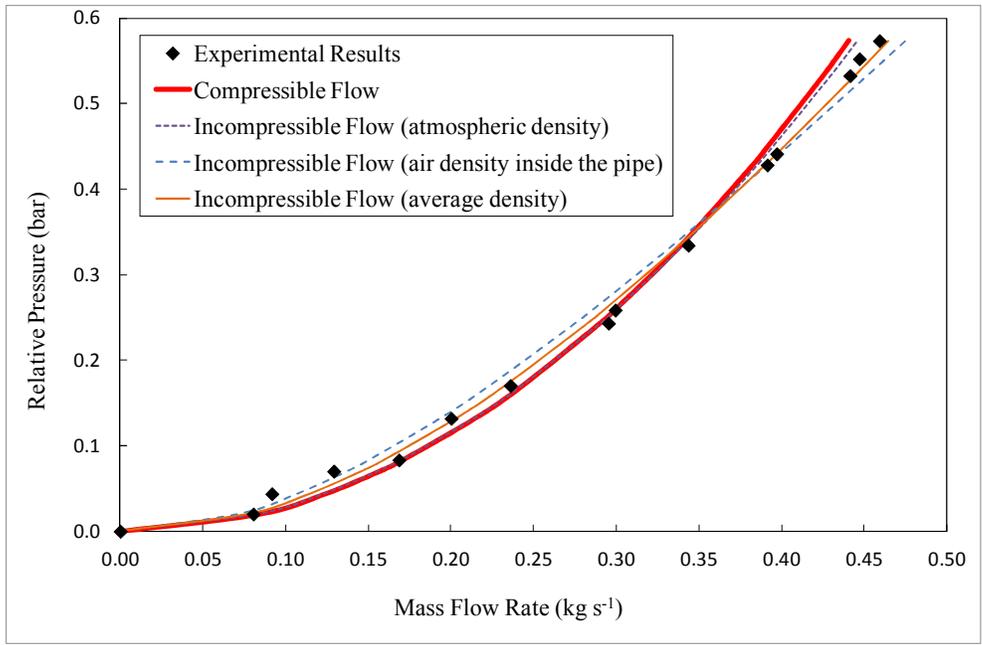
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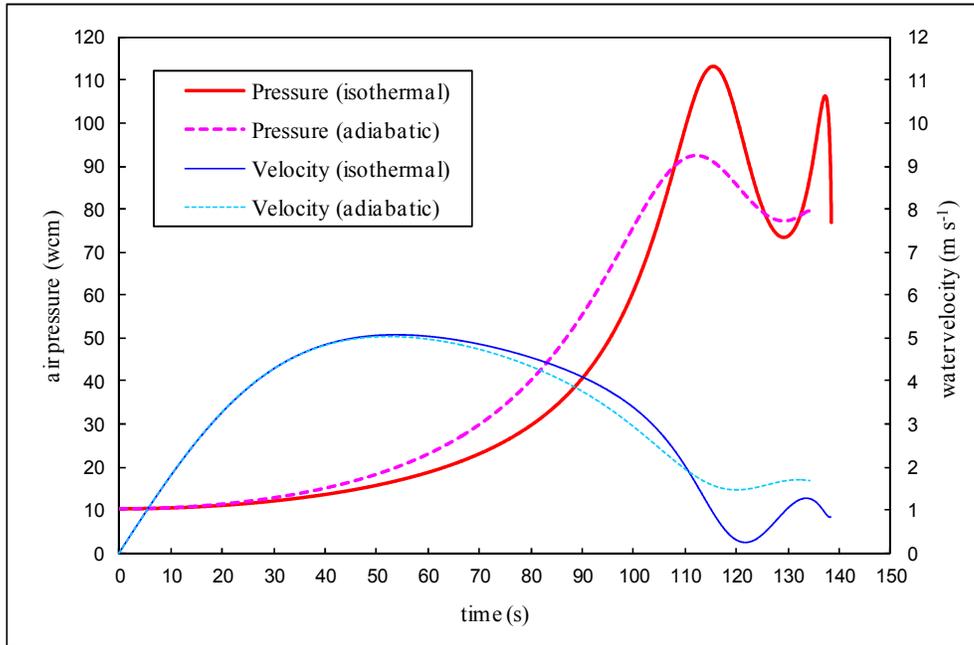
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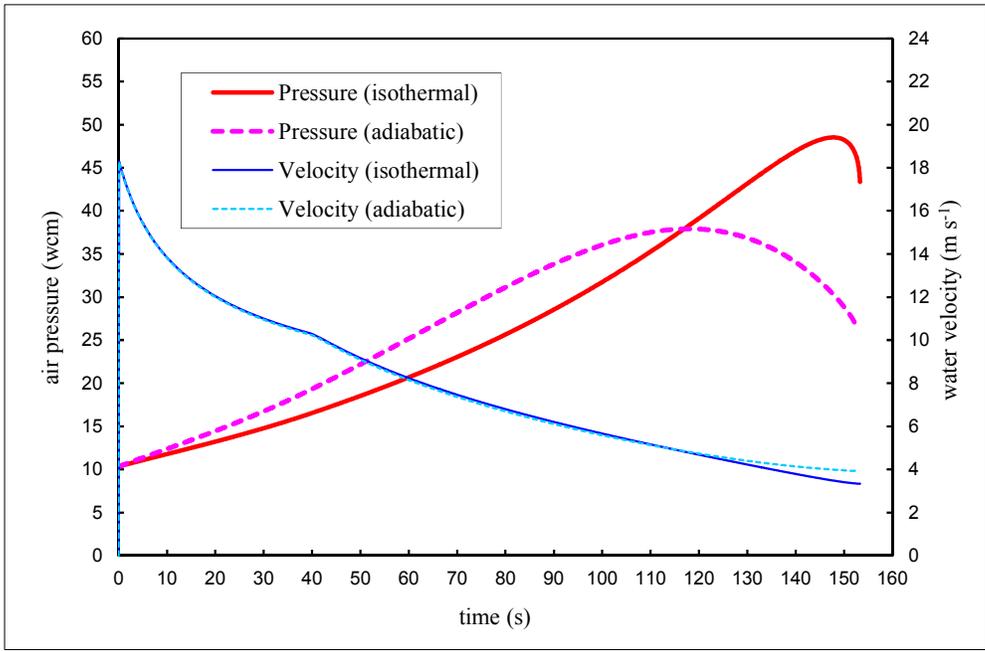
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