Practical Formulas for the Dimensioning of Air Valves

Alberto Bianchi[1]; Stefano Mambretti[2]; and Paola Pianta[3]

Abstract: On the basis of theoretical considerations, the technical note presents two practical formulas for the dimensioning of air valves when filling a pipe with water. One is to be used for designing air valves on the basis of the maximum allowed water hammer overpressures; the other when the maximum in pipe water velocity is set. The reliability of these formulas was tested with a numerical model based on the same hypothesis, which was in turn verified with experimental tests.

DOI: 10.1061/(ASCE)0733-9429(2007)133:10(1177)

CE Database subject headings: Air flow; Design; Laboratory tests; Water pipelines; Water hammer; Water flow.

Introduction

The presence of air entrained by a liquid flow in a pipeline under pressure often gives rise to operational problems in plant management, either in the form of concentrated headloss due to section reduction, or in the form of amplified overpressures in the case of unsteady flow after plant operations like pump startup or stop. Given these considerations, there is an evident need to insert air sections in order to allow the air trapped inside the pipeline to escape, and particular attention should be paid to the correct dimensioning of the air release sections.

Various authors have studied the transient in pipes due to air efflux (Benfratello 1957; De Martino et al. 1986) even with experimental comparisons (Gisonni 1998) and considering the related numerical problems (Roth and Nucera 1988). Lingireddy et al. (2004) studied the surges resulting from air release, paying particular attention to their dimensions. While a number of empirical formulas can be found in the literature, this technical note is an attempt to provide simple practical formulas based on fundamental principles; it reports a theoretical study intended to develop a method with which to design appropriate dimensions of the air valves, after which laboratory tests were performed in order to verify the correctness of the mathematical model. Practical formulas were derived and were checked by comparison with numerical and experimental studies.

[1]Associate Professor, DIIAR—Politecnico di Milano, Piazza Leonardo da Vinci 32, 20133 Milano, Italy. E-mail: alberto.bianchi@polimi.it

[2]Associate Professor, DIIAR—Politecnico di Milano, Piazza Leonardo da Vinci 32, 20133 Milano, Italy (corresponding author). E-mail: stefano.mambretti@polimi.it

[3]Post Graduate Student, DIIAR—Politecnico di Milano, Piazza Leonardo da Vinci 32, 20133 Milano, Italy. E-mail: piantapaola@libero.it

Note. Discussion open until March 1, 2008. Separate discussions must be submitted for individual papers. To extend the closing date by one month, a written request must be filed with the ASCE Managing Editor. The manuscript for this technical note was submitted for review and possible publication on January 17, 2006; approved on March 27, 2007. This technical note is part of the ***Journal of Hydraulic Engineering***, Vol. 133, No. 10, October 1, 2007. ©ASCE, ISSN 0733-9429/2007/10-1177–1180/$25.00.

Basic Hypothesis

The basic hypothesis is that these processes are isentropic (Mironer 1979; Whitaker 1968). The process of air expulsion through an air valve can be described by the polytropic equations of ideal gases and the Bernoulli equation. The equation with which to compute the sonic mass discharge m from a valve with cross section $A_{\text{air valve}}$ is

$$m = 0.685 \cdot C_d \cdot A_{\text{air valve}} \cdot \frac{\rho_{\text{atm}}^{0.5}}{p_{\text{atm}}^{0.357}} \cdot p_{\text{in}}^{0.857} \quad (1)$$

In this equation p=pressure and ρ=air density; the subscript atm=values at atmospheric pressure; subscript in=quantities inside the pipe; and C_d=coefficient of discharge. The ratio of specific heats has been implicitly set equal to 1.4, in accord with the isentropic hypothesis.

Practical Formula—Condition on Maximum Pressure

Complex models, which are still the best way to simulate networks, are not immediately applicable. As a consequence, a simple formula is derived that allows the easy design of air valves so that maximum pressure values can be acceptable for the given pipe. If the valve diameter is too small, the time taken to fill the pipe is too long. On the other hand, if the valve diameter is too large, air efflux does not give enough resistance and, as a consequence, the discharge in the pipe increases and the velocity becomes too high. At the end of the air efflux transient, this velocity suddenly becomes zero, water stops when it collides with the valve outlet, and this generates high water hammer pressures.

As is well known, the water hammer pressure Δp due to unsteady flow transient in the case of sudden closure of the valve downstream is given by the Allievi–Joukowski equation and is equal to

$$\Delta p = c \cdot \rho_{\text{water}} \cdot v_{\text{water}} = c \cdot \rho_{\text{water}} \cdot \frac{Q_{\text{in volume of water}}}{A_{\text{pipe}}} \quad (2)$$

where c=water hammer wave velocity; ρ_{water}=water density; v_{water}=water velocity in the pipe; $Q_{\text{in volume of water}}$=volume water discharge; and A_{pipe}=area of the pipe section.

If it is required that the maximum water hammer pressure value be limited to a fixed Δp value, it is necessary for the discharge in the pipe to be, as a maximum, equal to

$$Q_{\text{in volume of water}} = \frac{\Delta p \cdot A_{\text{pipe}}}{c \cdot \rho_{\text{water}}} \tag{3}$$

If $p_{\max}$=maximum pressure value in the pipe, the pipe must be filled at a pressure equal to $p_{\text{filling}} = p_{\max} - \Delta p$, and, during the transient, the corresponding density ρ_{filling} must be constant, i.e., the following ratio must be constant

$$\rho_{\text{filling}} = \frac{\text{Air mass in pipe}}{\text{Available volume}} \tag{4}$$

and then

$$m = \rho_{\text{filling}} \cdot Q_{\text{in volume of water}} \tag{5}$$

Substituting Eq. (3) into Eq. (5)

$$m = \frac{\Delta p \cdot A_{\text{pipe}} \cdot \rho_{\text{filling}}}{c \cdot \rho_{\text{water}}} \tag{6}$$

and, taking into account the polytropic equation, we can obtain the following

$$m = \frac{A_{\text{pipe}} \cdot \rho_{\text{atm}}}{c \cdot \rho_{\text{water}} \cdot p_{\text{atm}}^{0.714}} \cdot p_{\text{filling}}^{0.714} \cdot \Delta p \tag{7}$$

and under sonic conditions [Eq. (1)]

$$A_{\text{air valve}} = \frac{A_{\text{pipe}}}{c \cdot \rho_{\text{water}}} \cdot \frac{1.46 \cdot \rho_{\text{atm}}^{0.5}}{\text{Cd} \cdot p_{\text{atm}}^{0.357}} \cdot p_{\text{filling}}^{-0.143} \cdot \Delta p \tag{8}$$

Sonic conditions become established when $p_{\text{in}} \geq 189$ kPa, i.e., in most practical cases.

On this hypothesis Eq. (8) can be written as

$$D_{\text{valve}} = K \cdot D_{\text{pipe}} \cdot (C_d \cdot c)^{-0.5} \cdot p_{\text{filling}}^{-0.072} \cdot \Delta p^{0.5} \tag{9}$$

with $K = 5.128 \times 10^{-3}$ $\text{m}^{0.5}$ $\text{s}^{-0.5}$ Pa^{0428}

Eq. (9) is valid for single pipeline systems.

Practical Formula—Condition on Maximum Velocity

It is in some cases preferred to constrain the maximum velocity v_{filling} when filling a pipe. In this case Eq. (2) becomes

$$\Delta p = c \cdot \rho_{\text{water}} \cdot v_{\text{filling}} \tag{10}$$

so that Eq. (3) is now

$$Q_{\text{in volume of water}} = v_{\text{filling}} \cdot A_{\text{pipe}} \tag{11}$$

And substituting Eq. (11) into Eq. (5) yields

$$m = \rho_{\text{filling}} \cdot v_{\text{filling}} \cdot A_{\text{pipe}} \tag{12}$$

Taking the polytropic equation into account, Eq. (12) becomes

$$m = \rho_{\text{atm}} \cdot \frac{p_{\text{filling}}^{0.714}}{p_{\text{atm}}^{0.714}} \cdot v_{\text{filling}} \cdot A_{\text{pipe}} \tag{13}$$

and under sonic conditions [Eq. (1)]

$$0.685 \cdot \text{Cd} \cdot A_{\text{air valve}} \cdot \frac{\rho_{\text{atm}}^{0.5}}{p_{\text{atm}}^{0.357}} \cdot p_{\text{in}}^{0.857} = \rho_{\text{atm}} \cdot \frac{p_{\text{filling}}^{0.357}}{p_{\text{atm}}^{0.357}} \cdot v_{\text{filling}} \cdot A_{\text{pipe}} \tag{14}$$

In this case, the equation for dimensioning air valves becomes

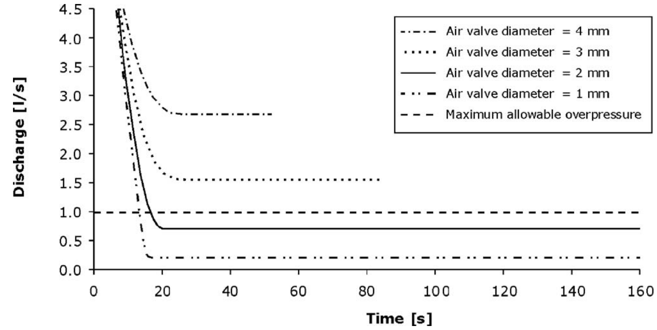

Fig. 1. Discharge versus time: simulation with different air valve diameters and comparison with discharge associated with maximum allowable overpressure

$$D_{\text{air valve}} = 9.886 \cdot \text{Cd}^{-0.5} \cdot p_{\text{filling}}^{0.072} \cdot v_{\text{filling}}^{0.5} \cdot D_{\text{pipe}} \tag{15}$$

Eq. (15) is also valid only for single–pipeline systems.

Numerical Model to Validate Practical Formula

In order to validate the practical formula, simulations were performed using a rigid–water–column numerical model similar to that presented by Liou and Hunt (1996). Different systems were assumed. The maximum allowed pressure and pipe dimensions were assigned to each of them, and the air valve dimensions were computed with practical formula Eq. (9) [or Eq. (15) when required]. The systems were then simulated with the mathematical model, assuming different dimensions of the air valve, and the maximum allowable discharges were compared with those carried out from the model.

For instance, suppose that a valve is being designed with a maximum allowable overpressure $\Delta p = 50$ m, a filling pressure $p_{\text{filling}} = 10$ m, a pipe diameter $D = 5$ cm, and a celerity $c = 1{,}000$ m/s. Eq. (9) gives $D_{\text{valve}} = 2.23$ mm. From the Allievi–Joukowski equation it is evident that the maximum allowable discharge is equal to 1 L/s.

Fig. 1 shows the results of the simulations for this case, assuming air diameters equal to 1, 2, 3, and 4 mm. Eq. (9) correctly estimates the maximum D_{valve}, because if $D_{\text{valve}} > 2$ mm then higher velocities result, which imply higher water hammer pressures. On the other hand, smaller D_{valve} entails lower water velocities, which imply more time to fill the pipe.

Laboratory Tests to Validate Complete Mathematical Model

In order to verify the validity of the mathematical model used to check the practical formula, a comparison was made between the results obtained by the model itself and those acquired by a number of laboratory tests performed in the Laboratory of Hydraulics of the Politecnico of Milan, Italy.

The laboratory system, as shown in Fig. 2, consisted of an iron spiral pipe 90 m long, diameter equal to 52 mm, and Manning's roughness coefficient (measured in steady flow conditions) equal to 0.009 $\text{m}^{-1/3}$ s. The pipe was fed by a constant head tank positioned 5 m above the outfall, with an air valve positioned at its end. Air was present in the duct from the beginning of each trial, and it could be easily evacuated by the water entering through a

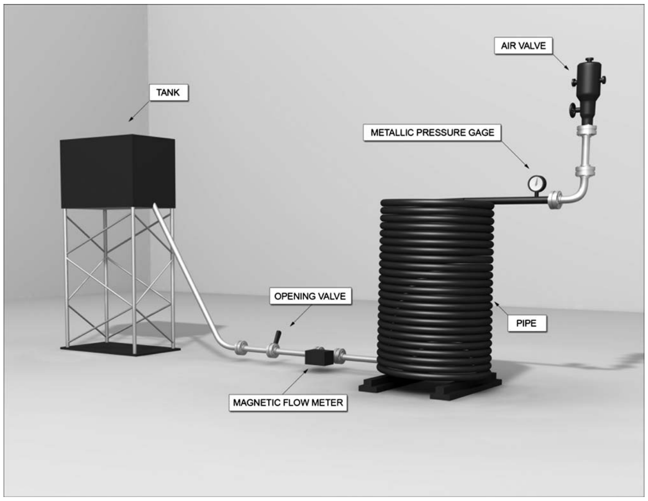

Fig. 2. Three–dimensional schematic of laboratory model

valve which opened instantaneously. In the meantime, both the values of the flow rate of the water entering the pipe and the air pressure just before it escaped from the air valve itself were measured. A magnetic flow meter recorded the flow rate values, while the air pressure values were measured with a metallic pressure gauge.

Using Eq. (9), for sonic flow, an optimal diameter of 2 mm was obtained. The numeric simulations and the laboratory tests carried out with the above-calculated air valve diameter gave good results, examples of which are shown in Fig. 3 and 4.

We can conclude that the model agrees well with reality with regard to both the flow rate and the pressure.

Conclusions

Given the operational problems due to the presence of air in the pipes that may arise when a plant is being driven, a theoretical study was conducted in order to determine a method for the optimization of air valve dimensions. The "optimal" dimensions are those with which: (1) the time of air expulsion is shortest; and (2) the allowable water hammer overpressures, generated when the water column impacts with the opening of the air valves, is satisfied. An alternative formulation, based on the maximum filling velocity allowable, was also given.

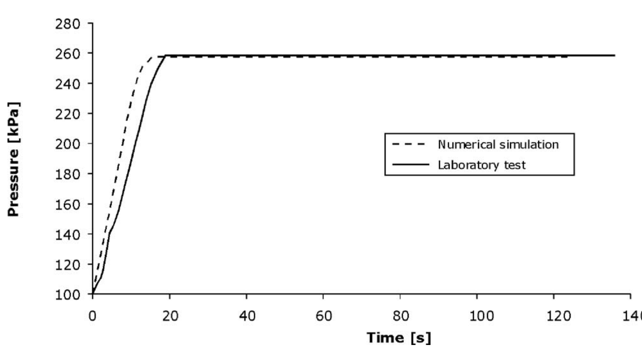

Fig. 3. Comparison between numerical simulation and laboratory test. Air valve diameter equal to 2.0 mm—pressure versus time.

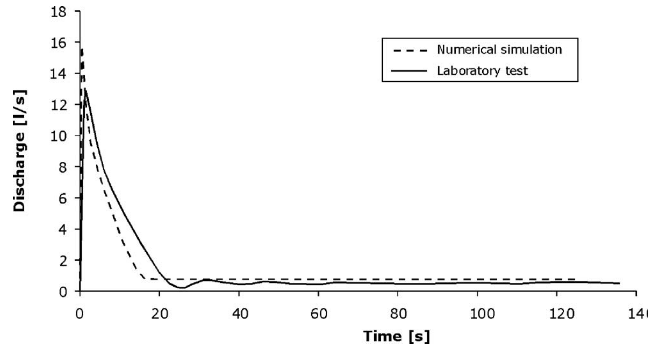

Fig. 4. Comparison between numerical simulation and laboratory test. Air valve diameter equal to 2.0 mm—discharge versus time.

A direct method was established for the dimensioning of the air valves, the purpose being to determine the dimension of the orifice, given the pipe diameter, the value of the suitable overpressure in the pipe, the velocity of pressure wave in the system water pipe, and the discharge coefficient of the orifice [Eq. (9)]. The alternative formula [Eq. (15)] allows determination of the dimension of the orifice given the pipe diameter, the value of the suitable velocity in the pipe, and the discharge coefficient of the orifice. These can be considered valid formulas for a sonic air flow: that is to say when, during the discharge, the pressure in the duct exceeds or at least reaches the value of 1.89 atm, which is what occurs in most real cases.

A complete mathematical model of the laboratory plant was also developed, under the same hypothesis as the practical formula. Laboratory tests were performed to verify the validity of the mathematical model. The results of the numeric simulations were confirmed: the diameter of the air valve calculated with the practical formula was found to be the most suitable one; in particular, it was possible to verify that air valves dimensions smaller than the dimension yielded by the practical formula extend the time necessary for the air expulsion, while larger dimensions may give rise to very high water hammer overpressures.

The experimental tests reproducing the theoretical model and carried out with laboratory equipment confirm that the embraced mathematical model well represent the real phenomenon from a global point of view, so that it is possible to apply the accepted practical formula in order to dimension the required air valves.

Notation

The following symbols are used in this technical note:

A = surface;
C_d = coefficient of discharge;
c = water hammer wave velocity;
D = diameter;
K = empirical constant;
m = air mass discharge;
p = pressure;
Q = water volume discharge;
v = velocity of flow; and
ρ = density.

Subscripts

atm = values at atmospheric pressure;
filling = quantities in pipe during filling transient;
in = quantities inside pipe;

pipe = quantities related to pipe;
valve = quantities related to valve; and
water = quantities related to water.

References

Benfratello, G. (1957). “Riempimento di una condotta elevatoria alimentata da una pompa centrifuga.” *Proc., 5th Convegno di Idraulica*, Stamperia Artistica Nazionale, Torino, Italy, 263–272 (in Italian).

De Martino, G., Giugni, M., and Viparelli, M. (1986). “Transitori in condotte idriche causati da fuoriuscita d’aria.” *Idrotecnica*, Vol. 2, Associazione Idrotecnica Italiana, Rome, Italy, 125–138 (in Italian).

Gisonni, C. (1998). “Confronto teorico-sperimentale di transitori in condotte in pressione causati da fuoriuscita d’aria.” *Proc., 26th Convegno di Idraulica e Costruzioni Idrauliche*, Vol. III, CUCEM, Catania, Italy, 529–540 (in Italian).

Lingireddy, S., Wood, D. J., and Zloczower, N. (2004). “Pressure surges in pipeline systems resulting from air releases.” *J. Am. Water Works Assoc.*, 96(7), 88–94.

Liou, C. P., and Hunt, W. A. (1996). “Filling of pipelines with undulating elevation profiles.” *J. Hydraul. Eng.*, 122(10), 534—539.

Mironer, A. (1979). *Engineering fluid mechanics*, McGraw-Hill, New York.

Roth, G., and Nucera, D. (1988). “Sull’utilizzo di metodi di calcolo numerico nell’analisi di fenomeni transitori attraverso sfiati.” *Proc., Convegno di Idraulica e Costruzioni Idrauliche*, Vol. II, L’Aquila, Maggioli, Rimini, Italy, 527–541, (in Italian).

Whitaker, S. (1968). *Introduction to fluid mechanics*, Prentice-Hall, Englewood Cliffs, N.J.