ABSTRACT

Some researchers have pointed out that air/vacuum valves performance curves provided by manufacturers need to be applied with caution, since inaccuracies might be present. In the present research, in order to understand the phenomenon of airflow through air valves and to establish the necessary characteristics of consistent curves, a theoretical model and data from experimental tests previously done by other researchers were considered. As a result, practical criteria were established for the evaluation of the consistency of air/vacuum valves performance curves. Additionally, 15 models of air/vacuum valves from 9 manufacturers from around the world were evaluated regarding their consistency. The diameters of the models ranged from 1/2 in to 24 in. It was concluded, in agreement with previous authors, that the performance curves of air/vacuum valves are frequently not entirely reliable. For example, for the evaluated models of air/vacuum valves, it is considered a single curve for both admission and expulsion in 33% of the cases. This is not acceptable, since expulsion and admission are different types of flow situations. Another important conclusion is that the development of standardized experimental procedures and graphic presentation for the characterization of air valves is necessary.

Keywords: air valves, airflow, water mains

1 INTRODUCTION

1.1 Air management in water mains

In a water main, the water flow is capable of transporting entrapped air downstream. It is not possible, though, to rely on this type of transportation for the totality of air removal operations (Koelle, 2000 [1]). Therefore, the use of air valves is indispensable. Additionally, air valves can be applied in hydraulic systems for air admission during extreme low pressures caused by transient phenomena (Aquino, 2013 [2]). As mentioned by Ramezani et al. (2015) [3], the three main types of air valves are: air-release valves (orifice diameters ranging from 1 mm to 5 mm), air/vacuum valves (orifice diameters ranging from 50 mm to 200 mm) and combination air valves. This type of device can increase mains security levels and energy efficiency of pumping operations (Escarameia, 2005 [4]). There are, though, lots of uncertainties associated with the use of air valves and, therefore, hydraulic engineers need to apply them with special care and with the knowledge of their limitations (McPherson and Haeckler, 2012 [5]).
1.2 The issue of inaccurate air/vacuum valves performance curves

In order to characterize the behavior of air/vacuum valves, manufacturers develop performance curves that relate airflow with differential pressure for admission and expulsion. Some authors point out that, since air/vacuum valves operate under dynamic conditions during transient phenomena, this type of characterization might not be totally appropriate (Lucca et al., 2010 [6] and Bergant et al., 2012 [7]). Despite this consideration, there are studies that show that, even under static conditions, the results of experimental tests often differ from the data presented by manufacturers in their catalogs (Fuertes et al., 2006 [8] and Iglesias et al., 2014 [9]). For this reason, in the present paper, an investigation about this issue was done.

2 AIRFLOW THROUGH AIR VALVES

2.1 The critical condition for airflow through an orifice

For the analysis of airflow through air valves, it is important to consider the velocity of air. In this regard, Equation 1 defines the Mach’s number $M$. In this equation, $V$ is the velocity of the fluid in m/s and $c$ is the sound velocity inside the fluid in m/s. If $M = 1$, there is sonic flow (critical condition); if $M > 1$, there is supersonic flow; and if $M < 1$, there is subsonic flow.

$$M = \frac{V}{c}$$  \hspace{1cm} (1)

Equation 2 can be applied for flows of compressible fluids through orifices opened to the atmosphere (Fox et al., 2006 [10]). In this equation, $P$ is pressure in Pa and $k$ is the specific heat ratio. Index $atm$ stands for atmospheric and index $i$ for internal.

$$\frac{P_{atm}}{P_i} = \left(1 + \frac{k - 1}{2} M^2\right)^{\frac{k}{k-1}}$$  \hspace{1cm} (2)

During air expulsion and air intake by air valves, isentropic and reversible processes take place ($k = 1.4$) because, in these situations, friction can be neglected and pressure variations occur rapidly with no time for heat exchanges. For sonic isentropic flow during expulsion, Equation 2 yields $P_{atm}/P_i = 0.528$. On the other hand, for sonic isentropic flow during admission, $P_{atm}/P_i = 1.893$ is obtained.

2.2 Theoretical Equations

Equations 3, 4, 5 and 6 are specifically used for air valves modeling, according to traditional books about hydraulic transients (Wylie and Streeter, 1983 [11] and Chaudhry, 1987 [12]). In these Equations, with slightly modified presentations for the sake of increasing the facility of comparison between inflow and outflow situations, $\dot{m}$ is the air mass flow in kg/s, $C_{adm}$ and $C_{exp}$ are the dimensionless admission and expulsion coefficients, $A_{adm}$ and $A_{exp}$ are the areas of the admission and expulsion orifices in m$^2$, $T_i$ and $T_{atm}$ are the internal and atmospheric temperatures in K, $P_i$ and $P_{atm}$ are the internal and atmospheric pressures in Pa, $\rho_i$ and $\rho_{atm}$ are the internal and atmospheric air densities in kg/m$^3$. 

$$\dot{m} = C_{adm} A_{adm} \sqrt{2 \rho_i T_i}$$  \hspace{1cm} (3)

$$\dot{m} = C_{exp} A_{exp} \sqrt{2 \rho_i T_i}$$  \hspace{1cm} (4)

$$\dot{m} = C_{adm} A_{adm} \sqrt{2 \rho_{atm} T_{atm}}$$  \hspace{1cm} (5)

$$\dot{m} = C_{exp} A_{exp} \sqrt{2 \rho_{atm} T_{atm}}$$  \hspace{1cm} (6)
\[ \dot{m} = C_{adm} A_{adm} \sqrt{7 P_{atm} \rho_{atm} \left[ \left( \frac{P_i}{P_{atm}} \right)^{1.4286} - \left( \frac{P_{atm}}{P_i} \right)^{1.714} \right]} \] (3)

\[ \dot{m} = C_{adm} A_{adm} \frac{0.686}{\sqrt{R T_{atm}}} P_{atm} = cte \] (4)

\[ \dot{m} = -C_{exp} A_{exp} \sqrt{7 P_i \rho_i \left[ \left( \frac{P_{atm}}{P_i} \right)^{1.4286} - \left( \frac{P_{atm}}{P_i} \right)^{1.714} \right]} \] (5)

\[ \dot{m} = -C_{exp} A_{exp} \frac{0.686}{\sqrt{R T_i}} P_i \] (6)

Equation (3) governs subsonic air admission when \( P_{atm} > P_i > 0.528 P_{atm} \). Equation (4) is for sonic air admission when \( P_i < 0.528 P_{atm} \). Equation (5) governs subsonic air expulsion when \( P_{atm} < P_i < 1.893 P_{atm} \). Equation (6) is for sonic air expulsion when \( P_i > 1.893 P_{atm} \). If we consider \( P_{atm} = 1 \) atm, then we have that 0.528 \( P_{atm} = 53,499.6 \) Pa and 1.893 \( P_{atm} = 191,808.23 \) Pa. Beyond the sonic limit, air velocity is limited to 343.4 m/s for atmospheric pressure and temperature of 20°C.

These Equations were plotted in Figure 1 for several orifice diameters of a model of commercial air/vacuum valve, considering, just for the sake of graphic visualization, \( C_{adm} = C_{exp} = 1.0 \). Considering absolute values, for a given differential pressure, air mass outflow is greater than air mass inflow. For air admission, air mass flow would increase with \( P_{atm} \rho_{atm} \) increments. For air expulsion, air mass flow increases with \( P_i \rho_i \) increments. In this regard, during admission, atmospheric air density and pressure are constants. However, during expulsion, internal air density increases with internal pressure increments, even after the sonic limit. In Figure 1 it can also be clearly noted the sonic limit for air mass inflow in which the curves become horizontal for internal pressure values lower than 53,499.6 Pa. The same does not happen for air expulsion after the sonic limit.

![Figure 1: Air mass flow curve according to the traditional theoretical model for a commercial air valve model](image-url)
2.3 Equations for airflow considering incompressible flow

A simplification of Equations 3 to 6 considering now incompressible flow, is done by Equations 7 to 10 respectively, according to Fuertes et al. (2006) [8]. The values of the constants $C_1$, $C_2$, $K_1$ and $K_2$ should be evaluated for each specific air valve.

\[ Q = C_1 \sqrt{(P_{atm} - P_i)P_{atm}} \]  
\[ Q = K_1 \]  
\[ Q = C_2 \sqrt{(P_i - P_{atm})P_i} \]  
\[ Q = K_2P_i \]

2.4 Expected shape of air/vacuum valves performance curves

Regarding the shape of air/vacuum valves performance curves, two general trends should be noted, according to the model presented by Equations 3 to 6:

- For a given value of $|\Delta P|$, $|\Delta P| = P_i - P_{atm}$ for air expulsion or $|\Delta P| = P_{atm} - P_i$ for air admission, the absolute value of air mass outflow is greater than the absolute value of air mass inflow. This trend is confirmed by the experiments performed by Iglesias et al. (2014) [9].

- Because of the occurrence of the sonic limit for inflow, the air mass inflow curve needs to tend to a horizontal line for values of $P_i$ near or lower than $0.528P_{atm}$. This trend is also confirmed by the results presented by Iglesias et al. (2014) [9]. It can be hard, though, to evaluate a performance curve in this regard if the differential pressure range for admission presented by the graph is much smaller than the subsonic one.

Regarding the units for airflow presented in performance curves, air volumetric flow in m$^3$/s, for example, is not acceptable, since air is compressible. It would be acceptable, though, the presentation of air mass flow in sm$^3$/s for example.

According to Iglesias et al. (2014) [9], the model presented by Equations 3 to 6 is more adequate to describe the behavior of air valves during air admission. The model proposed by Fuertes et al. (2006) [8], though, is more adequate to describe the behavior of air expulsion. Thus, a hybrid model Streeter-Fuertes might be a good option for air/vacuum valves modeling. This hybrid model would use Equations 3 and 4 for air admission and Equations 9 and 10 for air expulsion.

Examples of air/vacuum valves performance curves provided by manufacturers are presented in Figure 2. In this figure, performance curve a) follows the two trends, except for $DN150$ mm that doesn’t follow the second one. The unit used for airflow is inappropriate, since it is a volumetric unit. A reasonable supposition is that the unit m$^3$/s could be understood as sm$^3$/s.

For performance curve b), the first trend is not followed, since the same curve is used for air admission and expulsion. For the nominal diameter of 2 in, the second trend is not respected. However, the used unit for air mass flow scfm is acceptable.

The performance curve c) for air admission (right side) yields, for a given value of $|\Delta p|$, the same value of flow as the performance curve c) for expulsion (left side). This way, the first trend is not
Figure 2: Examples of performance curves provided by manufacturers catalogs

respected. The second trend, though, seems to be respected by these curves. The unit used for airflow is, as it happens for curve a), not acceptable.

3 RESULTS AND DISCUSSION

3.1 Evaluation of the consistency of manufacturers performance curves

Considering the expected shape of air/vacuum valves performance curves as described in the previous section, the curves of 15 models of air/vacuum valves from 9 manufacturers were evaluated. The results are presented in Tables 1 and 2.

For the evaluated models of air valves, it is considered a single curve for both admission and expulsion in 33% of the cases. This is not acceptable, since expulsion and admission are different types of flow situations and, thus, different curves should be obtained from experimental tests.

However, the presentation of distinct curves for admission and expulsion does not guarantee that the curves are actually different. From the ten cases in which the manufacturers present distinct curves for admission and expulsion, half of the time, the admission curve proves to be actually equivalent to the expulsion curve.

Considering the second general trend, the performance curves are consistent more often than not. For some cases, though, considering that the negative differential pressure is not near the sonic limit, there
Table 1: Analysis of air/vacuum valves performance curves provided by manufacturers considering the first general trend and the adopted units

| Manufacturer/Model | Same performance graph for outflow and inflow? | Is outflow > inflow for equal values of $|\Delta p|$? | Unit of airflow |
|-------------------|-----------------------------------------------|-----------------------------------------------|-----------------|
| Man. A, Mod. A, 1/2 in - 20 in | YES | NO | scfs |
| Man. B, Mod. A, DN50 mm - DN300 mm | NO | No. Same values, except for DN250/300 for which inflow > outflow. | m³/s |
| Man. B, Mod. B, DN50 mm - DN200 mm | NO | NO. Same values. | m³/s |
| Man. B, Mod. C, DN50 mm - DN150 mm | NO | YES | m³/s. As a contradiction, it is also indicated scfm as the unit. |
| Man. B, Mod. D, DN50 mm - DN200 mm | NO | NO. Same values. | m³/s |
| Man. C, Mod. A, 1/2 in - 16 in | NO | NO. Same values. | 1000 scfm |
| Man. D, Mod. A, 1/2 in - 16 in | YES | NO | 1000 scfm |
| Man. E, Mod. A, DN50 mm - DN200 mm | NO | NO. Inflow > outflow. | m³/s |
| Man. E, Mod. A, 1/2 in - 24 in | NO | YES | scfm |
| Man. G, Mod. A, 1/2 in - 6 in | YES | NO | cfm |
| Man. G, Mod. B, 1/2 in - 24 in | YES | NO | cfm |
| Man. H, Mod. A, 1 in - 6 in | YES | NO | cfs |
| Man. I, Mod. A, DN50 mm - DN150 mm | NO | YES | m³/h |
| Man. I, Mod. B, DN100 mm - DN300 mm | NO | YES | m³/h |

is no way to do a proper evaluation.

As described previously, there is a velocity limitation for sonic flow. Air expulsion becomes sonic when the differential pressure is equal or greater than 90.48 kPa. For air admission, the sonic limit happens when the differential pressure is equal or lower than $-47.83$ kPa.

For only 40% of the models, the graphs have ranges of differential pressures that comprise the sonic limit for air admission. For air expulsion, the range of the differential pressure comprises the sonic limit only for 33% of the cases.

Considering the units used, an acceptable air mass flow unit was applied for only 27% of the models.

### 3.2 Analysis of the results

A well characterized air/vacuum valve can be successfully simulated computationally. That is desirable when air valves are part of a water hammer protection system. The curves presented by manufacturers, though, frequently seem to lack consistency. The lack of consistency happens when a curve do not agree with the physical principles of the airflow phenomenon and with recent experimental data.

With the investigation here presented, it was established some characteristics that should be present in a consistent performance curve: 1) for a given value of $|\Delta P|$, the absolute value of air mass outflow should be greater than the absolute value of air mass inflow; 2) because of the occurrence of the sonic limit for inflow, the air mass inflow curve needs to tend to a horizontal line for values of $P_i$ near or lower than $0.528P_{atm}$; 3) the unit for air flow should be presented in terms of air mass flow like m³/s and scfm for example; and 4) the graphs should comprise at least the subsonic region.
Table 2: Analysis of air/vacuum valves performance curves provided by manufacturers considering the pressure ranges and the second general trend

| Manufacturer/Model | Maximum $|\Delta p| > 0$ (kPa) | Maximum $|\Delta p| < 0$ (kPa) | For values near the maximum $|\Delta p|$ when $\Delta p < 0$, does the inflow become constant? |
|---------------------|------------------|------------------|--------------------------------|
| Man. A, Mod. A, 1/2 in - 20 in | 34.47 | 34.47 | YES |
| Man. B, Mod. A, DN50 mm - DN300 mm | 40 | 47 | YES |
| Man. B, Mod. B, DN50 mm - DN200 mm | 89 | 47 | YES, except for DN150/200. |
| Man. B, Mod. C, DN50 mm - DN150 mm | 100 | 50 | YES, except for DN150. |
| Man. B, Mod. D, DN50 mm - DN200 mm | 89 | 47 | YES, except for DN150/200. |
| Man. B, Mod. E, DN50 mm and DN80 mm | 89 | 47 | YES, except for DN80. |
| Man. C, Mod. A, 1/2 in - 16 in | 103.42 | 34.47 | Only for small orifices of each graph. |
| Man. D, Mod. A, 1/2 in - 16 in | 34.47 | 34.47 | Only for small orifices of each graph. |
| Man. E, Mod. A, DN50 mm - DN200 mm | 40 | 45 | YES |
| Man. F, Mod. A, 1/2 in - 24 in | 41.37 | 41.37 | YES (logarithmic graph). |
| Man. G, Mod. A, 1/2 in - 6 in | 34.47 | 34.47 | Only for 1/2 in and 1 in. |
| Man. G, Mod. B, 1/2 in - 24 in | 34.47 | 34.47 | Only for 1/2 in and 1 in. |
| Man. H, Mod. A, 1 in - 6 in | 34.47 | 34.47 | Only for 1 in and 2 in. |
| Man. I, Mod. A, DN50 mm - DN150 mm | 80 | 40 | YES, except for DN150. |
| Man. I, Mod. B, DN100 mm - DN300 mm | 80 | 50 | Only for DN200. |

The aforementioned criteria for the evaluation of the consistency of air/vacuum valves performance curves rely on aspects that can be easily noted when a graph is consulted. These criteria are substantiated by the scientific literature and should be sufficient for a fast analysis of the data by a hydraulic engineer during the selection phase.

It was also observed that manufacturers often present curves with limited differential pressure ranges. Though an increase of the differential pressure range demands more expensive experiments, a range that comprises the whole subsonic regimen is desirable. With the information of the relationship between differential pressure and air mass flow for the subsonic range, it is possible to extrapolate for the sonic ranges. For sonic expulsion, the kinetic closure should be taken into account.

4 CONCLUSIONS

In the present paper, an analysis regarding air valves performance curves was done. It was found that, frequently, manufacturers do not provide acceptable curves. This might lead to wrongs in design. It was developed a set of criteria to determine the consistency of performance curves: 1) for a given value of $|\Delta P|$, the absolute value of air mass outflow must be greater than the absolute value of air mass inflow; 2) because of the occurrence of the sonic limit for inflow, the air mass inflow curve needs to tend to a horizontal line for values of $P_i$ near or lower than $0.528P_{atm}$; 3) the use of a proper air mass flow unit is necessary; and 4) the complete subsonic range should be presented.

For 33% of the analyzed curves, a single curve for admission and expulsion was considered. An
acceptable air mass flow unit was applied for only 27% of the models. Frequently, the graphs do not comprise the whole subsonic airflow regimen.

It is desirable the development of standardized experimental procedures for the characterization of air valves. In addition to that, a standardized presentation of the data would also be positive. This way, the information presented by manufacturers would be more reliable and easy to understand by hydraulic engineers.

Further research should be developed to evaluate the possible consequences of the use of inaccurate air valves performance curves in the design phase of water mains. Possibly, the use of inaccurate curves could lead to improper and even dangerous design decisions.

The phenomenon of air expulsion and admission during hydraulic transients might have different characteristics when compared to the steady regimen of air mass flow. This observation is important, since the development of air mass flow curves by manufacturers is usually done for steady flow conditions.

References