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A NEW EXPERIMENTAL ALGORITHM FOR THE EVALUATION OF THE TRUE SONIC CONDUCTANCE OF PNEUMATIC COMPONENTS USING THE CHARACTERISTIC UNLOADING TIME

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Abstract

In this paper an alternative method for obtaining the sonic conductance of pneumatic valves, *C*, is presented. The method uses the characteristic unloading time defined in a transitory discharge process and supposes an experimental cost lower than the ISO 6358 procedure. With this method the test rig needed is not so large and a precise measure of the variables involved in the discharge, pressure, mean temperature or specific volume, is not required either. Furthermore, the author has found out experimentally that *C* depends on the geometric factor *L*/*D* of the chamber that impulses the mass flow rate and not only on the effective section *A* of the valve element. The sonic conductance obtained by the characteristic unloading time method is smaller than the one obtained by the ISO 6358, and finally explains some experimental points. Firstly, the effective mass flow through some valves is significantly inferior to the one expected when considering the *C* ISO estimation. And secondly, it looks like if the valve would conduce a different mass flow depending on the system to which it is connected.

Keywords: sonic conductance, mass flow rate, pneumatic valves and nozzles, characteristic time

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1 Introduction

An accurate method of calculating the mass flow rate through nozzles, valves or vents is essential in several industrial pneumatic applications. Pneumatic flow valves are employed both in discontinuous and in continuous control. In the industry domain they are in general used with pneumatic actuators in applications of automation in manufacturing, for pieces handling, in grippers and in robots. Although the two types of processes, on-off and continuous, differ, they both require an accurate analysis of the instantaneous delivered flow through the control valve.

The International Standard ISO 6358 (1989) introduces a simple analytic method for controlling, measuring, conducting or guiding the air flow in pneumatic circuits which is directly applicable to all types of pneumatic components. This pattern establishes a simplification of the theoretical equation first obtained by Saint-Venant in 1839 for the isentropic mass flow rate of compressible fluid through convergent nozzles, which adapts the real characteris tics of the element by using two experimental coefficients: the sonic conductance *C* and the critical pressure rate *b*. The International Standard shows how to obtain these parameters by means of tests at steady flow.

2 Ideal Isentropic Flow in a Convergent Nozzle

Saint-Venant and Wentzel were the first to develop a theoretical model for the calculation of the mass flow rate at permanent regimen through nozzles. However, they observed that the analytic expression obtained was different from reality, since it seemed to indicate that the gas could not flow into a vacuum. Their work was rediscovered 30 years later by other researchers who obtained identical results, thus reinforcing their findings. According to them, the theory was refuted when the downstream pressure dropped below a certain value. In 1856, Joule and Thompson in England and Weisback in Germany derived their own equations, and in 1867 Napier indicated for the first time that the mass flow rate was independent from the downstream pressure when the pressure factor was lower than approximately 0.5. In 1886, Osborne Reynolds made probably the most important contribution to the phenomenon since the work of Saint-Venant and Wentzel, by recognizing that the gas velocity in the point of the flow with minimum area, whose pressure drops continuously, is equal to the sound velocity at that point. Aurel Stodole proved, at the beginning of the century, the existence of supersonic flow in the Laval convergent-divergent nozzle (Sullivan, 1981).

The method is based on the assumption that the process is adiabatic and reversible, i.e. isentropic, and it makes use of the polytropic equation with the respective γ factor and the state equation for ideal gases. If the drive tank volume were big enough to prevent the exhaust gas passing through the nozzle from causing an appreciable decrease in the gas pressure, we would have a nearly stationary process in which the average outgoing velocity would be:

$$c_2 = \left\{ \int_{p_2}^{p_1} \frac{2}{\rho} dp + c_1^2 \right\}^{1/2} = \left\{ \frac{2\gamma}{\gamma - 1} \frac{p_1}{\rho_1} \left[1 - r^{\frac{\gamma-1}{\gamma}} \right] \right\}^{1/2} \tag{1}$$

where the flow velocity on entry, c_1, is not considered, because of its magnitude in comparison to c_2 and the r is the pressure ratio

$$r = \frac{p_2}{p_1} \tag{2}$$

With the ideal gas state equation and Eq. 1 it is possible to deduce that

$$G = A \frac{p_1}{\sqrt{T_1}} \left\{ \frac{2\gamma}{R(\gamma - 1)} \left[r^{\frac{2}{\gamma}} - r^{\frac{\gamma+1}{\gamma}} \right] \right\}^{1/2} \tag{3}$$

Expression in Eq. 3 reaches a maximum when the pressure ratio r has the critical value b, which makes the flow take the maximum value

$$b = \frac{p_{2s}}{p_1} = \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma}{\gamma - 1}} \tag{4}$$

This expression shows that the value of the critical ratio r depends on γ, although this dependence is weak: when $\gamma = 1.4$ then $b = 0.53$, whereas when $\gamma = 1.5$ then $b = 0.55$. The maximum mass flow rate is easily calculated by replacing Eq. 4 in Eq. 3, resulting in:

$$G_s = A \frac{p_1}{\sqrt{T_1}} \left\{ \frac{\gamma}{R} \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}} \right\}^{1/2} \tag{5}$$

The velocity corresponding to this condition is found in the same way, with Eq. 1, giving:

$$c_s = \sqrt{\frac{2\gamma}{\gamma + 1} \frac{p_1}{\rho_1}} = \sqrt{\gamma \frac{p_{2s}}{\rho_{2s}}} \tag{6}$$

which is also the local velocity of the sound in the Laplace's equation for perfect gases. The above expression substantiates the divergence of theory from experience. When a gas expands in a convergent nozzle it is not possible to take a pressure at outlet p_2 lower than the value p_{2s}, which corresponds to the maximum mass flow rate, because the pressure wave for the decrease of p_2 does not come back upstream. The flowing gas does not know that the downstream pressure has decreased, and because of this, the flow is maintained constant and with the maximum value.

3 ISO Standard 6358 for the Calculation of the Mass Flow Rate

The non-ideal nature of real gases and the fact that the expansion process cannot be regarded as isentropic in all senses deflect the real behaviour of the nozzle from the theoretic characteristic in Eq. 3 and 5. It is usually accepted that the nozzle geometry is a very important local factor in the mass flow rate characteristic. In this paper some experimental results obtained with a non-convergent nozzle are used in order to illustrate the method.

As a result of the acceptable approximation at the low working pressures of the pneumatic components, the International Standards can be applied. Figure 1 shows a real characteristic of the pneumatic nozzle taken from pneumatic mass flow rate tests.

In the subsonic path where $p_2 > b\, p_1$, the flow-mass rate drops nearly elliptically with increasing r, and so it is reasonable to simplify the Eq. 3 by defining the term

$$\omega = \sqrt{1 - \left(\frac{r - b}{1 - b} \right)^2} \tag{7}$$

giving $$G = A K \frac{p_1}{\sqrt{T_1}} \omega \tag{8}$$

where K is a characteristic constant of the gas, function of γ and R. Another advantage of using Eq. 7 is that it allows rapid observation of the influence of b.

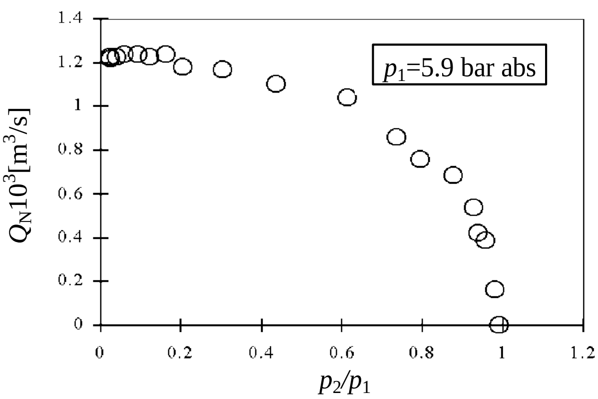


Fig. 1: *Characteristic flow versus r for the pneumatic nozzle with a 1.5 mm diameter*

When $bp_1 \geq p_2$ the flow is choked, resulting in $\omega = 1$ and the mass rate having a maximum value, specifically

$$G_s = AK \frac{p_1}{\sqrt{T_1}} \tag{9}$$

The flow characteristics of the pneumatic components can be defined in normal conditions. Using the normal conditions, $T_N = 293$ K, $p_N = 100$ kPa, at which

$$G_s = \rho_N Q_N = \rho_{2s} Q_{2s} \tag{10}$$

we have $$Q_N = AK \frac{p_1}{\rho_N \sqrt{T_1}} = \frac{\rho_{2s}}{\rho_N} Q_{2s} \tag{11}$$

The above equations indicate that the $Q_N \sqrt{T_1}$ term is proportional to the upstream pressure in choked flow. The mass flow rate is calculated unequivocally from expression in Eq. 10, so there is no possibility of confusion.

The relation between the volumetric flow at normal test conditions and the incoming pressure of the nozzle is called the conductivity or, commonly, conductance of the pneumatic element, that is,

$$C = \frac{Q_N}{p_1} \tag{12}$$

which combined with Eq. 11 indicates that, by definition,

$$C = \frac{K}{\rho_N \sqrt{T_N}} A \tag{13}$$

and because of this, the only intrinsic factor of the pneumatic element that affects the conductance C is the effective area A and hence the nozzle geometry. Figure 2 shows the experimental dependence of C on section A.

It is fundamental to notice that the definition of C in the International Standard is based on test conditions at steady flow in which the upstream pressure is maintained constant, and consequently, the nozzle perceives a system with infinite capacity whose geometry is indifferent.

It is necessary to correct the measured flow with the factor ($T_1^{1/2}$) when the temperature is far from the normal conditions. This is because of the combination of Eq. 11 with the definition in Eq. 13, it can be found that

$$C = \frac{Q_N}{p_1} \sqrt{\frac{T_1}{T_N}} \tag{14}$$

If we know the upstream pressure and temperature, Eq. 14 allows us to determine the conductance of the nozzle by a test at steady flow regimen with a nominal flow rate Q_N. Generally, the method presents the following formulation: The mass flow rate is:

$$G = C K_T \rho_N p_1 \omega \tag{15}$$

being $\omega = 1$ in the choked case and ω from Eq. 7 in the subsonic case, with

$$K_T = \sqrt{\frac{T_N}{T_1}} \tag{16}$$

the stagnation temperature correcting ratio.

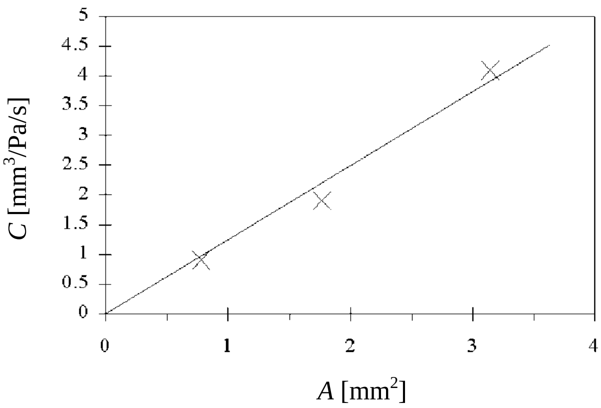


Fig. 2: *Experimental relation of C and the nozzle area tested following ISO 6358*

4 Introduction to the Unloading Process

Suppose a cylindrical tank of volume V as shown in Fig. 7, full of gas to a pressure of p_{10} and at a temperature T_{10} in thermal equilibrium with the tank wall. Unloading the gas into the atmosphere through a nozzle provokes the relaxation of the gas contained to atmospheric pressure p_{atm}. Obviously, the measured flow rate is not constant during the unloading process. From the ideal gas state law and admitting the conservation of the local equilibrium inside the chamber, it can be deduced that the internal mass changes with the rate

$$\frac{dm_1}{dt} = \frac{V}{R} \frac{d}{dt} \left(\frac{p_1}{T_1} \right) \tag{17}$$

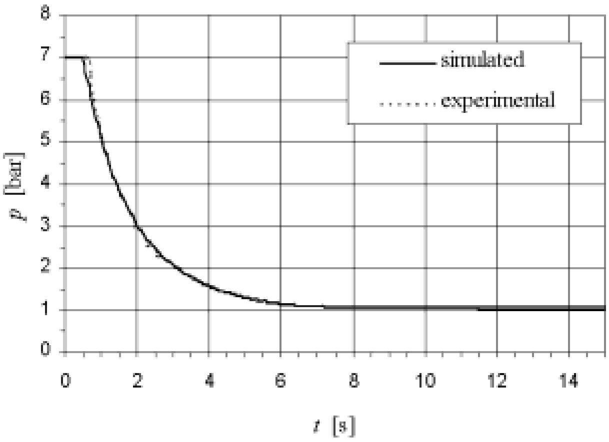
The local equilibrium hypothesis means that the process occurs as a succession of states in nearly static equilibrium. This is acceptable when the relation of diameters between the nozzle and the container volume is small, and, consequently, the gas in the cylinder is stable except in a small region near the outlet vent. This local effect can also occur in the stationary flow, and so, although it is incorrect to suppose that the unloading is nearly static, the continuous variation of the average pressure can be admitted.

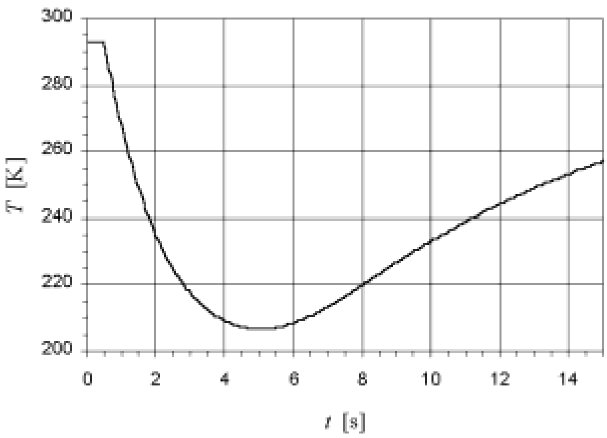
Although the inside pressure is uniform, the same is not true for the temperature because of the local differences of density occasioned by the appearance of flow and the effect of the heat transfer near the wall. However, following the explanations of Deckker and Chang (1968), it seems that the instantaneous temperature as calculated from the experimental pressure corresponds acceptably well to the average of the local temperatures experimentally measured.

It is often assumed that the process is isothermal near the wall and tends towards isentropic further towards the center. The thermal boundary layer width δ is being estimated by

$$\delta \approx \frac{L}{(\mathrm{Re\ Pr})^{1/2}} = \left(\frac{k_0 \tau}{\rho_0 c_{\mathrm{p}0}} \right)^{1/2} \tag{18}$$

Changing thermodynamics variables and parameters while unloading, δ can be calculated with variables in the stagnation condition. In this expression τ is the discharge time and it usually depends on valve size, tank volume and the stagnation condition. For example, with air at normal conditions (Pr = 0.69), a volume of characteristic length 0.1 m and being τ equal to 1 second (Re = 660), the wall thermal effect penetrates the vessel 5 millimeters. With air at 7 bar abs, δ becomes only 2 mm and the process would be nearly isentropic





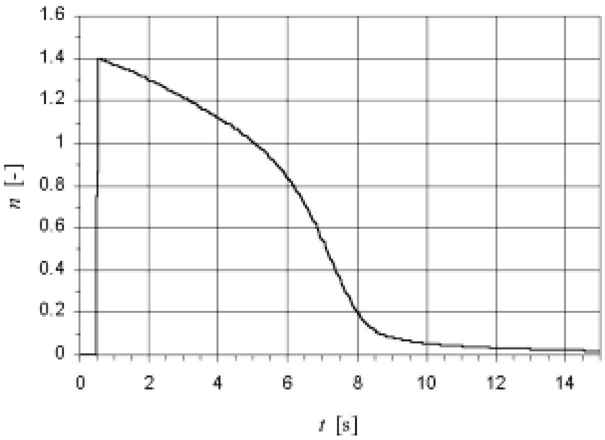


Fig. 3: *Temporal evolution of some thermodynamic variables during an unloading process*

inside. In fact, in an unloading process, the isentropic condition is quite approximate while the flow is choked and the thermal time is still bigger than the process time. In Fig. 3 both simulated and experimental pressures are compared for ideal gas. Averaged gas temperature and the instantaneous polytropic index of the process are represented as they have been obtained by simulation. It is also shown in Fig. 3 that the polytropic index is 1.4 just at the beginning, decreasing with time while discharging. Obviously, when n = 1 the temperature reaches its minimum value. The basis of the bond-graph simulation method are explained by de las Heras (1997) and applied by Sorli and de las Heras (1999) in the dynamic analysis of pneumatic actuators.

So, if the air inside the tank expands isentropically (at least at the beginning), it is

$$T_1 = T_{10} \left(\frac{p_1}{p_{10}} \right)^{\frac{\gamma-1}{\gamma}} \tag{19}$$

Equations 17 and 19 can be combined to give

$$-G = \frac{dm_1}{dt} = \frac{V}{\gamma R T_1} \frac{dp_1}{dt} < 0 \tag{20}$$

Using Eq. 15 with ω = 1, which is valid during the unloading process while $p_1 > p_{\mathrm{atm}}/b$, it can be deduced that,

$$\frac{dp_1}{dt} = -C K_{\mathrm{T}} \rho_{\mathrm{N}} \frac{\gamma R T_1}{V} p_1 \tag{21}$$

Furthermore, by combining and integrating Eq. 19 and 21, results

$$t = \frac{V}{C} \sqrt{\frac{T_{\mathrm{N}}}{T_{10}}} \frac{1}{p_{\mathrm{N}}} \frac{2}{\gamma - 1} \left[\left(\frac{p_{10}}{p_1} \right)^{\frac{\gamma-1}{2\gamma}} - 1 \right] \tag{22}$$

This equation is the basis of the SSE approach suggested by Xu Wen-Can (1989). Using the SSE method it is possible to obtain C from the temporal evolution of p_1 during the process under choked conditions. If the unloading process is stopped after a short time t^*, the system relaxes to the thermal equilibrium while pressure rises to a $p_{1\infty}$ located between p_{10} and p_{atm}. For

$t > t^*$, it is $G = 0$, and can be demonstrated that Eq. 22 can be written as

$$t^* = \frac{V}{C}\sqrt{\frac{T_N}{T_{10}}}\,\frac{1}{p_N}\,\frac{2}{\gamma-1}\left[\left(\frac{p_{10}}{p_{1\infty}}\right)^{\frac{\gamma-1}{2}} - 1\right] \tag{23}$$

Of course, it is clear that the SSE test rig is simpler than the ISO one, but the system pressures p_{10} and $p_{1\infty}$, and temperature T_{10}, are still needed in order to evaluate the system conductance with Eq. 23.

The method exposed by Han et al (1999) is similar to the SSE and can be used to discuss the pipe influence on the system. The pneumatic RC system behaviour seems to have been recognized but surprisingly it has not been completely understood. An intent was made by Eula, Ivanov and Viktorov (1996) when they used a time constant to evaluate the conductance of a photoacoustic cell in a simple way and with low level pressure signals.

The following is an explanation of a new algorithm for the evaluation of the sonic conductance of pneumatic valves and fittings that uses the characteristic unloading time, really simplifies the experiments and acquisition, and at last identifies the relation between the RC time and the valve sonic conductance.

5 A New Procedure: the Characteristic Unloading Time Method

What follows is a new algorithm for calculating the sonic conductance of pneumatic valves and nozzles proposed by de las Heras (1998).

It is clear from experience that the pressure inside the tank which is unloading into the atmosphere relaxes exponentially. Ideal gases and the hypothesis supposed in the preceding section yield Eq. 22, which can also be written as

$$p_1 = p_{10}\left\{1 + \frac{t}{\dfrac{V}{p_N C}\,\dfrac{2}{\gamma-1}\sqrt{\dfrac{T_N}{T_{10}}}}\right\}^{\frac{-2\gamma}{\gamma-1}} \tag{24}$$

Being

$$K_{T0} = \sqrt{\frac{T_N}{T_{10}}} \tag{25}$$

and assuming that $e^{ax} \approx (1+x)^a$, Eq. 24 can be written in first approximation as follows

$$p_1 \approx p_{10}\, e^{-\frac{t}{\frac{V}{\gamma p_N}\frac{1}{C}K_{T0}}} \tag{26}$$

Obviously, this expression is only validated while it is true that

$$t \ll \frac{V}{p_N C}\,\frac{2}{\gamma-1}K_{T0} \tag{27}$$

but it enables us to understand the unloading process by means of an exponential decrease. Without doubt, it is true at the initial instant, so it is possible to define from Eq. 26 the characteristic initial or starting time as:

$$\tau_0 = \frac{V}{\gamma p_N}\frac{1}{C}K_{T0} \tag{28}$$

The unloading time can also be defined as:

$$\tau = \mathrm{R^*C^*} \tag{29}$$

where R* is defined from Eq. 15 for choked flow as:

$$\mathrm{R^*} = \frac{1}{K_T \rho_N C} = \frac{p_1}{G_s} \tag{30}$$

and C* is defined from Eq. 20 as:

$$\mathrm{C^*} = \frac{V}{\gamma \dfrac{p_1}{\rho_1}} \tag{31}$$

Therefore, the pressure inside can be calculated by

$$p_1 = -\int_t \frac{G}{\mathrm{C^*}}dt \tag{32}$$

Strictly speaking, R* and C* are not the characteristic resistance of the nozzle and the capacity associated with the chamber that unloads because they do not have pressure/volumic flow and volume/pressure units. They are pseudo magnitudes (see units), while the gas normal density cancels when forming the time constant.

It is true that C* initially has the value

$$\mathrm{C_0^*} = \frac{V}{\gamma \dfrac{p_{10}}{\rho_{10}}} \tag{33}$$

while the initial resistance becomes

$$\mathrm{R_0^*} = \frac{1}{K_{T0}\rho_N C} = \frac{p_{10}}{G_{s0}} \tag{34}$$

Considering Eq. 25 and the ideal gas law, it is also true that

$$\frac{p_{10}}{\rho_{10}} = \frac{1}{K_{T0}^2}\frac{p_N}{\rho_N} \tag{35}$$

so Eq. 28 is being corroborated with

$$\tau_0 = \frac{V}{\gamma p_N}\frac{1}{C}K_{T0} \equiv \frac{1}{K_{T0}\rho_N C}\,\frac{V K_{T0}^2}{\gamma \dfrac{p_N}{\rho_N}} = \mathrm{R_0^*\ C_0^*} \tag{36}$$

As the unloading process advances, Eq. 27 becomes invalid and so does the exponential decay defined by τ_0. However, it is possible to suppose an expression such as Eq. 26 where the characteristic time is τ from Eq. 29, a function of time.

One can obtain an approximate value for τ experimentally by supposing that the system responds as a first order RC system. So, in the case of unloading into the atmosphere, the following should be true:

$$\left.\frac{dp_1}{dt}\right|_{t=0} = -\frac{p_{10} - p_{\mathrm{atm}}}{\tau_{\mathrm{exp}}} \tag{37}$$

and together with Eq. 21, 28, 30 and 31, it is possible to deduce that the average time constant of the unloading process is

$$\tau_{\mathrm{exp}} = \frac{p_{10} - p_{\mathrm{atm}}}{p_{10}} \frac{K_{\mathrm{T0}}}{C} \frac{V}{\gamma\, p_{\mathrm{N}}} = \frac{p_{10} - p_{\mathrm{atm}}}{p_{10}} \tau_0 \tag{38}$$

from where

$$C = \left\{ \frac{p_{10} - p_{\mathrm{atm}}}{p_{10}} \frac{K_{\mathrm{T0}} V}{\gamma\, p_{\mathrm{N}}} \right\} \frac{1}{\tau_{\mathrm{exp}}} \propto \frac{1}{\tau_{\mathrm{exp}}} \tag{39}$$

For instance, expressions in Eq. 38 and 39 offer a simple method for calculating the conductance of the nozzle, C, that only needs the measure of the experimental unloading time τ_{exp} and the initial state given by p_{10} and K_{T0} (usually 1).

A direct consequence of Eq. 39 is the C dependence on the τ_{exp} time that is the RC system time. Since τ_{exp} depends on the system pseudo resistance, R* in Eq. 30, and pseudo capacity, C* in Eq. 31, it is obvious now that C also depends on R* and C*.

The experimental unloading time in the RC analogy is the time that takes pressure to drop the 63.2 % of the complete pressure change. That is, the time when the absolute pressure reaches the value

$$p^* = p_{\mathrm{atm}} + (1 - 0.632)(p_{10} - p_{\mathrm{atm}}) \tag{40}$$

as is shown in Fig. 4. Furthermore, a curve fitting procedure could be used in order to have a correlation coefficient at will.

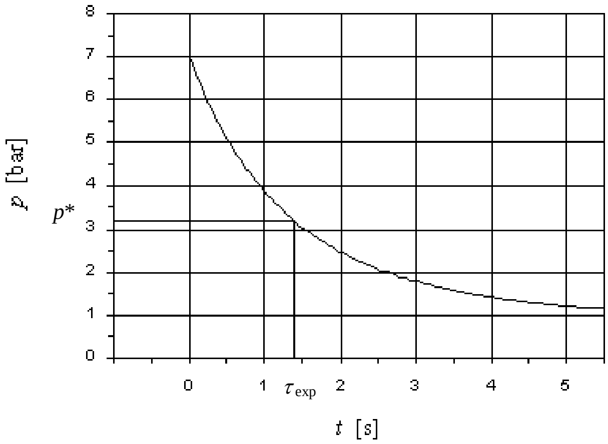


Fig. 4: *Experimental determination of the unloading time τ_{exp}. In the example, being p_{10} = 7 bar and p_{atm} = 1 bar, it results p* = 3.208 bar and τ_{exp} = 1.4 s*

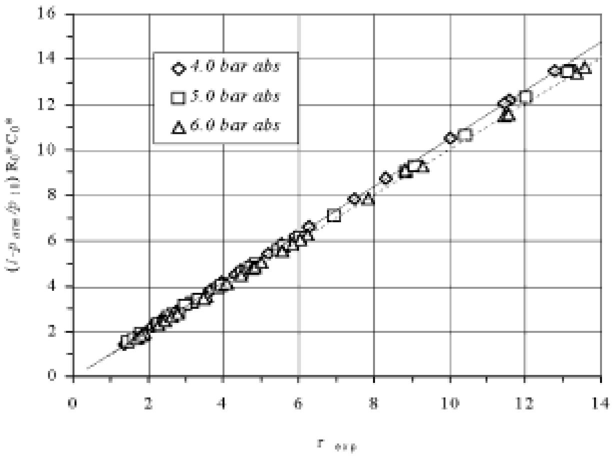


Fig. 5: *τ of Eq. 38 against the experimental τ_{exp} for different nozzle diameters and stagnation pressures p_{10}*

Anyway, in a practical point of view it is unnecessary to relate the decay curve to Pascal or other pressure units. If you use a pressure transducer in the test rig of Fig. 7 to monitor the pressure evolution while discharging, this one needs no adjustments, just linearity. The unloading time τ_{exp} could be measured on a volts-time chart or amperes-time chart.

The experimental time τ_{exp} does not coincide with τ in Eq. 29 because the system is not exactly like the one described by Eq. 37. It has been said that τ changes in a continuous manner while τ_{exp} assumes linearity and takes only one value. The first order approach is just a way of understanding the system behaviour. Anyway, the pattern of the unloading time constant makes it possible to predict the system transitory response and linearity, as is shown in Fig. 6 for different nozzle sections and initial states.

As the obtained results confirm by varying the initial pressure p_{10}, C does not depend only on section A of the nozzle or the equivalent area A of the valve, but also on the geometry of the unloading chamber. Nozzle conductance is represented in Fig. 6 as a function of the geometric ratio L/D for different nozzle diameters and initial pressures, and it can be seen to grow asymptotically as L/D does. The maximum is situated near the value of C found by the ISO steady flow tests, see Fig. 2.

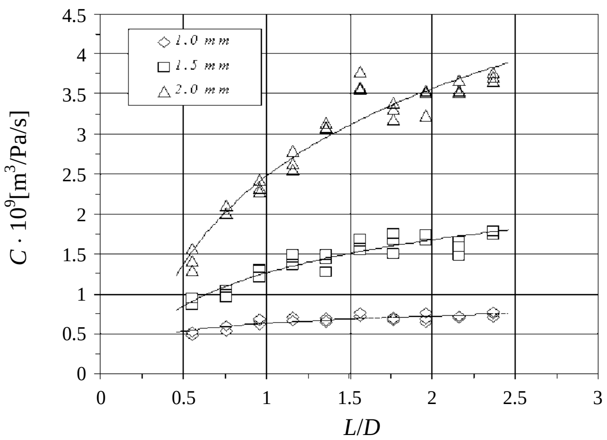


Fig. 6: *Nozzle conductance C against the L/D ratio for different diameters and pressures*

6 Experimental and Test Rig Methodology

The installation in Fig. 7 allows the unloading of the air contained in a cylindrical tank. The *L/D* geometric ratio can be changed at will between tests. After pre-loading the tank to a chosen pressure p_{10}, the air is expulsed into the atmosphere through the tested E nozzle. A quick directional control valve 2/2 is preferred for not affecting the unloading time by the valve opening time. The air should be in thermal equilibrium with the tank wall. Only initial pressure p_{10} and temperature T_{10} are needed for the evaluation of the conductance *C*, because the experimental unloading time can be observed in tendency by means of an equivalent electrical signal. One can use a not calibrated but linear pressure transducer for this.

The figures in this paper have been obtained varying the initial pressure, nozzle diameters and the *L/D* ratio. The working relative pressures were 3, 4 and 5 bar and nozzles of 1, 1.5 and 2 mm diameters were used. The *L/D* ratio was changed from 0.6 to 2.4.

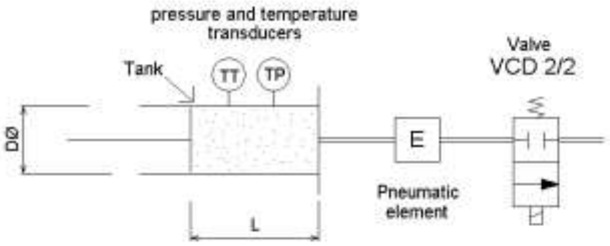


Fig. 7: *Installation test rig (schematic) for determining C by the discharge time method with unloading tests*

7 Conclusions

The mass flow rate through pneumatic fittings is usually calculated with Eq. 15 in stationary and also in transitory circumstances. Usually the conductance *C* of valves and other pneumatic elements is calculated by means of steady flow tests, as indicated by ISO 6358. However, when the unloading method is used, it can be seen that *C* depends on the geometry of the chamber, which impulses the mass flow rate and not only on the effective section *A* of the element. The geometric ratio *L/D* of the chamber plays an important role, as indicated in Fig. 6. Results show that *C* tends asymptotically to a maximum value when *L/D* increases and that, at least by its appearance, it seems the maximum is situated near the value of *C* found by the ISO steady flow tests (the longer the *L/D* ratio is, the more similar the flow paths inside the chamber are). In this way, it is not possible to determine the true conductance of the element by the steady flow test indicated by ISO 6358.

If the study is carried out by means of the unloading characteristic time, τ in Eq. 29, the contributions of the adjacent chamber, C* in Eq. 31, and the resistance associated with the system nozzle-conduct, R* in Eq. 30, are included in the determination of *C*. Accepting the experimental correlation shown in Fig. 5, it is also possible to find the value of *C* using unsteady flow tests measuring only the experimental time τ_{exp} and the initial conditions of the tank. The unloading method presented in this paper does not need an exact measure of pressure during the unloading process; what is required are the relative values showing their tendency and allowing us to visualize τ_{exp}, with Eq. 37, as in Fig. 4, in order to calculate *C* with Eq. 39. The initial stagnation pressure, p_{10}, can be measured with a common manometer.

An immediate evidence of these studies is that any given pneumatic element will conduct different flow rates depending on the system to which it is connected. The system, basically the impulsion or up-stream chamber and the nozzle, fittings or valves, must be coupled in order to define the mass flow rate in choked and also in subsonic conditions. Even for the same upstream/downstream pressure ratio, the mass flow rate depends on the chamber geometry and capacity, C*. This is a direct consequence of the *C* dependence on the RC system time that can not be undervalued but definitely accepted.

Nomenclature

a	constant	
A	equivalent valve area, nozzle area	[m^2]
b	critical pressure ratio	
c	gas celerity	[m/s]
c_p	gas specific heat at constant pressure	[J/kg/K]
C	sonic conductance	[m^3/s/Pa]
C*	characteristic pseudo capacity	[ms^2]
D	tank diameter	[m]
G	mass flow rate	[kg/s]
k	gas conductivity	[W/m/K]
K	gas characteristic constant	[sK$^{1/2}$/m]
L	tank length, characteristic length	[m]
m	mass of gas	[kg]
n	mean instantaneous polytropic index	
p	mean absolute gas pressure	[Pa]
Pr	Prandtl number, Pr = $c_p\,\mu/k$	
Q	volume flow rate	[m^3/s]
r	pressure ratio	
R	gas constant	[J/kg/K]
R*	characteristic pseudo resistance	[m^{-1}s^{-1}]
Re	Reynolds number, Re = $\rho L^2/(\mu\tau)$	
t	time	[s]
T	spatially averaged absolute gas temperature	[K]
V	tank volume	[m^3]
x	variable	
δ	thermal boundary layer width	[m]
γ	gas specific heat ratio	
μ	gas dynamic viscosity	[kg/m/s]
ω	subsonic term defined in (7)	

ρ	spatially averaged gas density	[kg/m^3]
τ	unloading characteristic time	[s]

Subscripts

0	stagnation condition, initial condition
1	up-stream
2	down-stream
atm	atmosphere
exp	experimental
N	normal condition
s	sonic condition, sound
T	temperature

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