Unsteady pulsed jets using pneumatic valves for flow separation control: effect of internal acoustic waves on external flow structure

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Abstract:
Flow control for enhancement of aerodynamic performances may be achieved by use of pulsed jets provided by fast valves. In this paper, the flow properties at the pulsed jet nozzle exit are studied for an actuation at low-frequency; these properties are shown to be influenced by pressure waves that develop in the pipe downstream of the valve, and also upstream of it. These pressure waves induce a marked sensitivity of the flow to small modifications of the classical command parameters such the actuation frequency and the duty cycle, for a given pressure.

Résumé :
Le contrôle d'écoulement pour l'amélioration des performances aérodynamiques peut faire appel à l'utilisation de jets pulsés commandés par des électrovannes rapides. Cet article montre au travers d'une étude de l'actionnement à basse fréquence que les propriétés de l'écoulement dans la section de sortie du jet de contrôle sont influencées par le développement d'ondes de pression qui transitent dans la conduite en aval de l'électrovanne, mais aussi en amont de celle-ci. L'existence de ces ondes de pression induit une sensibilité particulière de l'écoulement à de faibles modifications des classiques paramètres de commande que sont le rapport cyclique d'ouverture et la fréquence d'actionnement, à pression d'alimentation fixée.

Key words: Flow control; pulsed jets; acoustic waves.

1 Introduction

In several active flow control applications, unsteady jet generators are mounted at the surface of a model (wing, bluff body …) in order to interact with the external flow in such a way it promotes recovery of some aerodynamics performances like lift or drag, or reduces or suppresses flow detachment. Different kind of unsteady jet actuators are used in the literature: pulsed jets using valves or MEMS, plasma actuators, synthetic jets, self-oscillating actuators. The present paper is focused on the study of pulsed air jets generated using an electro-pneumatic valve and connected to a 2D exit slit. One key feature of the efficiency of such pulsed jets is not only the periodic injection of flow momentum, but also the formation during blowing of patches of vorticity on both edges of the jet. The intensity of these vorticity patches is related to the time evolution $U_j(t)$ of the jet exit velocity. In an ideal way, the pulsed jet generator produces a binary boundary condition consisting in a blowing phase with a constant value of $U_j(t)$, followed by a no blowing phase with $U_j(t)=0$. Nevertheless, the opening and closing dynamic behaviours of the valve produce deviations of the actual boundary condition $U_j(t)$ from this binary scheme. Moreover, the fast opening and closing of the valve generate acoustic waves producing pressure and velocity oscillations in the air contained in the connecting pipes upstream and downstream the valve. Such velocity $U_j(t)$ oscillations have been observed near the jet exit slit by several authors [1, 2]. For high actuation frequencies

larger than the frequency of acoustic waves, Barros [3] noticed very large variations of the jet exit velocity pattern during one cycle for small changes of the duty cycle; such a high sensitivity is expected to be related to the interaction of acoustic waves between successive actuation cycles when the period of actuation is smaller than the persistence of acoustic waves. The knowledge of velocity overshoot amplitude is of primary importance when dealing with the interaction of the pulsed jet and the outer flow. During closing, one important issue is also to know if the amplitude of negative pressure fluctuations inside the downstream pipe between the valve and the exit slot can be high enough to produce some suction of the external fluid.
For pulsed jets generated by electro-pneumatic valves, the primary control parameters are the actuation frequency f_0, the upstream pressure P_{inlet}, and the duty cycle DC. Therefore, it is of primary importance to understand the influence of these parameters on the properties of acoustic waves. The length of the connecting pipes and the sound velocity are also presumed to play an important role on the properties of velocity oscillations at the jet exit. In the present paper, the experiments are restricted to low actuation frequencies f_0 in order to characterize the intrinsic properties of pressure and velocity oscillations due to acoustic waves, without interaction between successive cycles or between opening and closing phases. In this case, the characteristic time of oscillations damping is small enough for the flow to reach a quasi-steady state at the end of both opening and closing of the valve.
In this paper, the experimental setup is presented first, then experimental results are analysed. The effect of the valve time delay is underlined before focusing on the observed oscillations of the velocity and the pressure according to operating conditions.

2 Experimental setup

2.1 Pneumatic setup

The working fluid is air. A sketch of the setup is shown in figure 1. The solenoid valve (5) is supplied via a 0.5 litre tank (3) whose pressure is regulated at a specified value by using a pressure regulator (2); the temperature inside the upper tank is constant during all the experiments around 20°C. Connecting pipes upstream and downstream the valve have the same internal diameter (2r = 6mm) and can be changed to study the influence of the length L_{up} between the tank and the valve inlet (4) and the overall length L_{down}, including both downstream connecting pipe (6) and diffuser lengths (7).

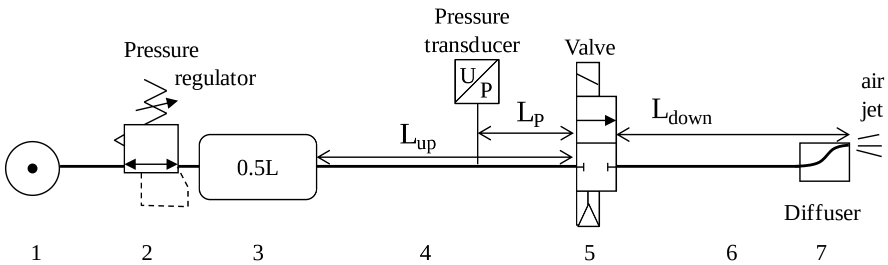


Fig. 1 - Sketch of the setup

The inlet section of the diffuser (7) is circular with a diameter of 5mm, while its exit section is rectangular, forming a narrow slit of width w = 40 mm, and thickness e = 1 mm. The diffuser has an internal complex S-shape in order to satisfy geometrical constraints of the model supporting the pulsed jet device and to smooth the transition between the inlet circular shape and the outlet rectangular shape.

2.2 Valve and control board

The pneumatic valve is a high speed two-port solenoid valve (SMC SX11-AJ). Its nominal flow rate is 50L/min (under standard reference atmosphere - ANR) at 3.5barA and it is given to work up to 1200Hz [4]. This very high dynamics is obtained thanks to a pressure return force to close the valve instead of the classical return spring usually used in this kind of valves. Its working principle is very similar to that described in [5] and requires a minimum working pressure (2.5barA as stated by the manufacturer) to ensure the correct closing of the valve.

The actuation of the valve is done thanks to a control board specifically developed to pilot the valve from a TTL signal whose frequency and duty cycle can be tuned. The typical signals are illustrated in Fig.2. The electronic power board provides 24V during enough time (about 0.5ms) to develop an electromechanical force sufficient to move the mobile part and open the valve. After this phase, the control is switched to limit the current at a level sufficiently low to not heat the valve but still maintain it fully open. When switched off, the electronics enables to shut off the current very rapidly to decrease the electromechanical force, the mobile part can then be moved by the pressure force to close the valve. This circuit board enables also the measurement of the current on the valve coil.

2.3 Pressure and velocity measurements

Single hot-wire anemometry is performed using a Dantec 55P01 probe combined with a Dantec Dynamics miniCTA measurements system. The wire is located at the exit section of the diffuser, in the middle of the slit. The wire direction is parallel to the slit largest dimension in order to minimize errors due to the finite length of the wire.

Inlet pressure measurements are made using a pressure sensor located at a fixed distance L_P of 50mm upstream the valve (figure 1) whatever the total upstream length L_up is. A high frequency miniature pressure transducer (Kulite ETL-1-140) is flush-mounted in a flange to be as close as possible to the pipe in order to avoid any distortions between the actual pressure and its measurement.

3 Experimental results

Tests were conducted for a fixed actuation frequency f_0 = 10Hz and a duty cycle DC = 50%. The frequency is low enough for the flow to reach a quasi-steady state at the end of both opening and closing phases. Pressure and velocity signals were recorded for two values of the upstream length (L_{up} =206 and 450 mm), three different pressure levels (P_{in} =2.5, 2.88 and 3.7barA), and different downstream lengths in the range of 155 to 250 mm. Table 1 summarizes the tested configurations.

The flow properties upstream the valve will be characterised by the analysis of the pressure measurements whereas the velocity at the diffuser exit will be used for the analysis of the flow downstream the valve. In most cases, the study at opening is separated from the one at closing. The origin of time is first chosen as the time at which the control signal sent to the electronic board is switched on (respectively, off). The signals presented in the following are obtained from a time-averaging over 30 periods at same experimental conditions (particularly at stabilized upstream pressure).

Regulated pressure [barA]	L_up [mm]	L_down [mm]
2.5	450	185, 210 and 250
2.88	206	155, 169, 177 and 200
3.7	206	155 and 177

Table 1 – Tested configurations (see sketch of Fig.1 for definition of lengths L_{up} and L_{down})

3.1 Example of pressure and velocity oscillations

Figure 2 shows an example of measurements. On the top, the control signal sent to the electronic power board together with the measured coil current of the valve are plotted. The upstream pressure and local velocity at outlet are shown below.

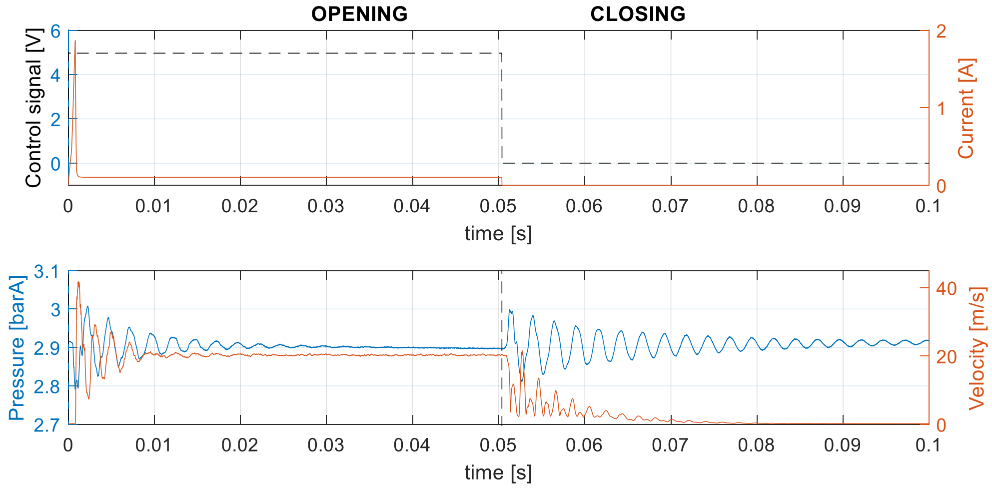
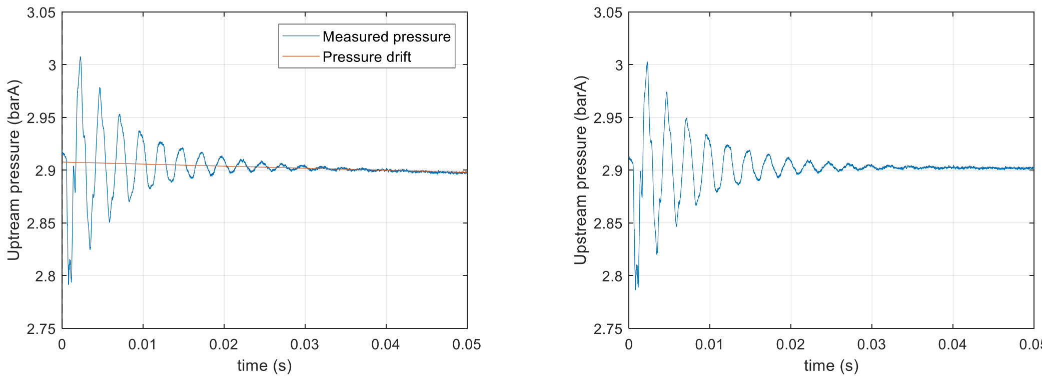


Fig. 2 - Example of measurements (P = 2.88barA, L_{up} =206mm and L_{down} =177mm)

Noticeable oscillations on both pressure and velocity signals can be observed at valve opening and closing. These oscillations, suspected to be related to acoustic waves, are the subject of the following analysis. One should first note that, after the valve closure, the velocity signal do not exhibit regular oscillations but a rectified waveform, which will be explained in the following.

The pressure signal in figure 2 has a slight decrease at opening and a slight increase at closing. This is attributed to the pressure regulator time response which is larger than the half period of observation. The amplitude of these drifts during the half period depends on the pressure level and the upstream pipe length. As a consequence, in order to make the analysis easier when comparing the pressures obtained for different test conditions (pressure level, upstream length), a correction is applied to the measured pressure signals. It consists in correcting the raw pressure signal from the slope of the drift obtained by a linear approximation of the pressure evolution as shown in figure 3a. This enables to obtain oscillations centered on the pressure mean value at opening (as the signal shown in figure 3b), respectively at closing.



a – Drift of pressure *b – Corrected pressure signal*

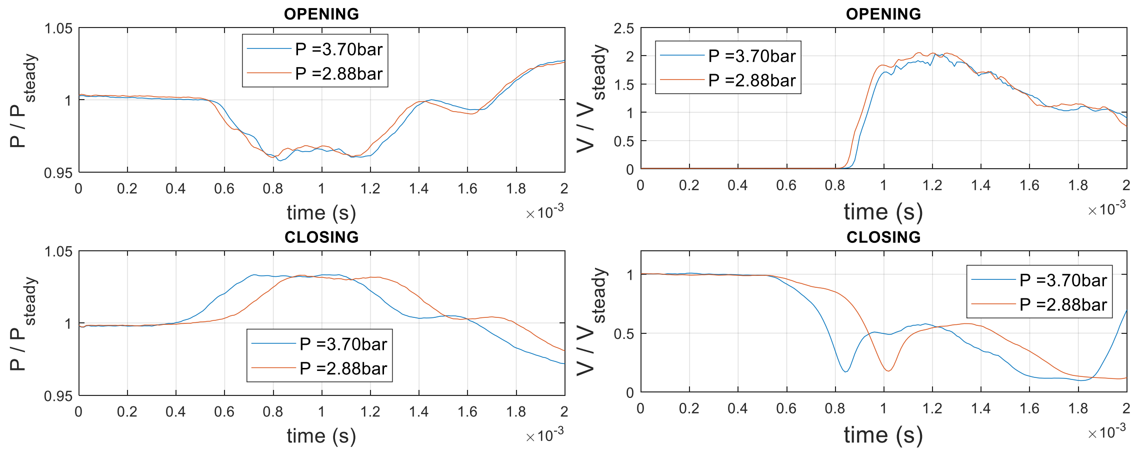
FIG. 3 - Correction of upstream pressure at opening (P=2.88 bars, L_{up}=206mm, L_{down}=177mm)

3.2 Identification of the time delay

Experimental recordings show that after the switching of the valve command, both pressure and velocity oscillations are delayed. To better visualize this time delay, figures 4 and 5 use signals normalized by their steady state value. For the same geometric configuration but for two different inlet pressures, the time evolution of pressure and velocity over the 2 first milliseconds after a change of command state (figure 4) leads to the following conclusions:

- the time delays at opening and at closing are different,
- at opening, the time delay is nearly independent of the inlet pressure P_{in},
- at closing, the time delay decreases with the increase of P_{in}.

The dependency on the inlet pressure at closing only can be attributed to the working principle of the valve which needs pressure force (pneumatic spring) to close: the higher the pressure is, the faster the closing of the valve. As the opening of the valve is done by the electromechanical force of the solenoid, it is less sensitive to pressure: only a very slight increase in the time delay can be seen in figure 4b, when the pressure level is increased.



a – Normalized pressures *b – Normalized velocities*

FIG. 4 – Influence of upstream pressure on the delays - zooms over 2ms (L_{up} =206mm, L_{down} =177mm)

Upstream the valve, as the pressure sensor is at a fixed distance from the valve, the time delay of the pressure is independent of the upstream length. But at downstream, since the hot-wire probe is at a variable distance of the valve, the time delay of the velocity increases linearly with the downstream length L_{down} for a given pressure (figure 5). Since the movement of the mobile part of the valve is very fast, it could be assumed that the delay is a delay due to the valve opening (resp. closing) plus a delay due to the wave propagation from the valve to the sensor (pressure at upstream, and velocity at downstream). It has been verified that this assumption is consistent with the time delays t_d identified experimentally for all experimental data.

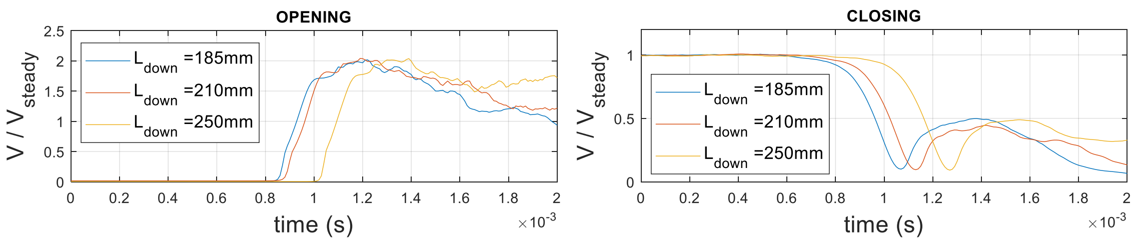


FIG. 5 – Influence of downstream lengths on the velocity delay - zooms over 2ms (Lup=450mm, P=2.5bar)

3.3 Characterization of acoustic waves using dimensionless parameters

Figure 6 shows for the nine configurations listed in Table 1, the time evolution of both normalized pressure P/P_{steady} and velocity V/V_{steady} versus the dimensionless time $(t-t_d)*f$, where t_d and P_{steady} refer to the time delay and the quasi-steady pressure as defined previously. f is the measured frequency of oscillations, and V_{steady} is the value of the velocity at the end of the opening phase, when the velocity has reached its asymptotic value.

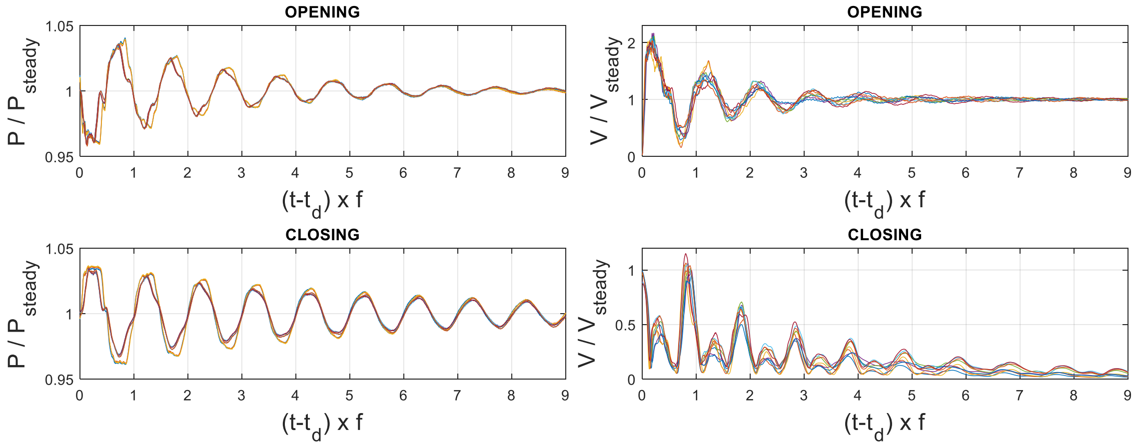


FIG. 6 - Evolution of the normalized pressure and velocity

From the set of experimental data, we can draw the following conclusions:

- the upstream (resp. downstream) oscillation frequency is independent of the downstream (resp. upstream) length;
- the oscillation frequency f of the upstream pressure (resp. downstream velocity) is a decreasing function of the upstream length L_{up} (resp. downstream length L_{down}). Moreover the upstream (resp. downstream) oscillation frequencies are the same at opening and closing;
- both frequencies are independent of the inlet pressure supply P_{in};
- the amplitude of the first peaks of pressure (resp. velocity) are nearly proportional to P_{in} (resp. V_{steady}), and independent of the lengths of the connecting pipes;
- the oscillation damping coefficient is larger at opening than at closing.

These results highlight the decoupling between upstream and downstream oscillations. If this decoupling is intuitively justified during the closing phase, it can be roughly justified during the opening phase by the very small valve throttle which induces a very strong geometric restriction that inhibits propagation of upstream acoustic waves through the valve in the flow direction.

3.4 Physical modeling

The aim in this section is to refer to existing analytical models to identify the physical parameters that define the frequency and the damping coefficient related to the oscillations in each part of the system (downstream and upstream of the valve).

3.4.1 Frequency

First, the frequency of the oscillations is remarkably independent of the state of the valve (open or closed). As stated above, acoustic waves in the pipes are presumed to be the cause of these oscillations. When the valve is closed, the pipe upstream the valve has one end closed, and the other opened on the pressure tank. To some extent, it corresponds to a classical closed-end pipe, whose resonance frequencies are provided by the formula given in Table 2. For comparison, the results obtained for an open-end pipe are also provided. The fundamental frequency of the closed-end pipe is thus half that of the open-end pipe. Considering the sound speed c = 340 m/s under normal atmospheric conditions, the closed-end pipe formula provides an estimate with a maximum deviation of 7% from the experimental results, which is quite satisfactory knowing that the measurement of the real lengths is not trivial in practice. Interestingly, when the valve is open, the relevant model remains the closed-end pipe, as the frequency of the oscillations obtained in this case is far from being twice the one obtained when the valve is closed. This is linked with the dimension of the valve opening section, which is small with respect to the pipe diameter. For the oscillations observed on the velocity downstream the valve, the closed-end pipe model still applies for the same reasons. Moreover it can be added that the valve opening corresponds to a sonic throat in the range of the NPR tested. These results confirm the acoustic nature of the waves in the upstream and downstream pipes of the valve.

	Closed-end pipe	Open-end pipe
Acoustic resonance frequencies	$F_{2n+1} = (2n+1)c/4L$	$F_n = nc/2L$

Table 2: Frequencies of acoustic modes in pipe configurations

3.4.2 Damping coefficient

As stated above, we can also observe a damping effect of the oscillations. Two sources of attenuation may be considered: the acoustic radiation at the pipe ends, and the viscous effects along the pipe wall. This latter appears to be predominant here, because of the high value of the ratio between the pipe length and its diameter. This is also confirmed by considering that, the relevant model being a closed-end pipe model, be the valve open or not, no change in the acoustic radiation at the open-end of the pipe should be expected. Thus the difference between the two damping coefficients comes from viscous effects which are more pronounced when the valve is open. Considering the case of a closed valve, thus with no mean flow in the pipe, the expression of the time absorption coefficient α may be expressed by [6]:

$$\alpha = \frac{\omega\, d}{2\, r}\left(1 + \sqrt{\frac{\chi}{\nu}}\left(\frac{Cp}{Cv} - 1\right)\right)$$

where ω is the angular frequency, ν is the kinematic viscosity, *Cp* and *Cv* the specific heat capacities, χ the thermal diffusivity for air and *d* is seen as a "penetration depth" for viscous effects. A simplified model of this quantity is given by [6]:

$$d = \sqrt{\frac{2\nu}{\omega}}$$

Evaluating these terms for the pipe upstream the valve provides the results illustrated in figure 7, where a good agreement is obtained for the logarithmic decrement δ between the measurements and the theoretical decay ($\delta = \alpha\frac{2\pi}{\omega}$) of the oscillations. The slight dependency of δ with the pipe length is linked with the dependency of the penetration length d with ω, which depends on the pipe length. It is likely that, in the case of an open valve, the mean flow enhances the penetration depth d which leads to larger values of the attenuation coefficient.

Let us note that the viscous effects are visible not only for the decay of the oscillation at the fundamental frequency f1, but also for higher harmonics at frequencies (2n+1) f1. These are associated with higher values of damping coefficient because of the higher frequency values. The oscillations signals, presented in figure 6 which are far from sinusoidal on the first few periods progressively tend to a sinusoidal signal after damping of the higher harmonics contributions.

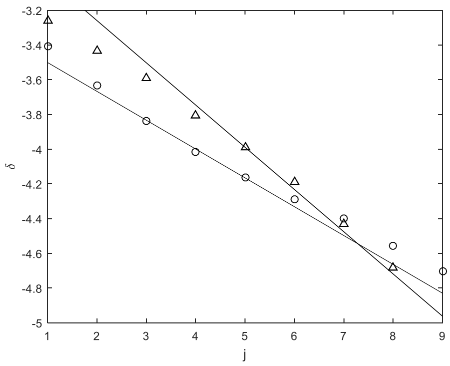


Figure 7: Logarithmic decrement δ estimated from the measurement data (Δ: L_{up} =450mm, o: L_{up} = 206mm) and from theoretical model (solid line), as a function of the index j of local maximum in the time series of pressure oscillations upstream of the valve. Valve closed.

4. Conclusions

For the applications of the pulsed jet generation to flow control of the wake of a bluff-body, the experiments of the unsteady flow generated by a fast electro-pneumatic valve reveal a noticeable influence of acoustic waves in the pipes connected to the valve. In order to characterize the intrinsic properties of the acoustic waves producing oscillations of the upstream pressure and of the velocity at the pulsed jet exit, the valve has been operated at low frequency in order to achieve a quasi-steady flow at the end of both opening and closing phases. Acoustic waves upstream and downstream the valve appear to be decoupled. Their frequency is mainly defined by the length of the inlet and exhaust connecting pipes. Oscillating pressure and velocity measured for a set of operating parameters and geometric configurations collapse using dimensionless parameters, provided that a time delay combining valve characteristics and wave propagation is introduced. Oscillation frequencies predicted by a model of the acoustic wave propagation characteristics in a single ended-pipe are in good agreement with the experiments. The damping coefficient of these waves is also satisfactorily predicted by taking viscous dissipation into account.

For future flow control applications, the operating frequency of the pulsed jet will be increased up to 900 Hz. At such high actuation frequencies, the damping of oscillations will be negligible and will produce complex interactions between acoustic waves generated at opening with waves generated at closing. We may suspect that some resonance can be achieved if the operating frequency of the valve is close to the intrinsic frequency of the acoustic waves: this will produce large blowing velocity peaks while keeping constant the inlet pressure. It is also expected that very different blowing velocity patterns at the jet exit can be achieved when the ratio between acoustic waves frequency and actuation frequency varies in a range of values centered around unity. The effect of such a ratio will be studied with the help of PIV measurements performed at the jet exit, in order to capture the unsteady flow structure near the pulsed jet exit.

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