

Prediction of flow-control devices' noise with modified acoustic perturbation equations

Luca Fenini and Stefano Malavasi

ABSTRACT

Fluid-dynamic noise emissions produced by flow-control devices inside ducts are a concerning issue for valve manufacturers and pipeline management. This work proposes a modified formulation of Acoustic Perturbation Equations (APE) that is applicable to industrial frameworks where the interest is addressed to noise prediction according to international standards. This formulation is derived from a literature APE system removing two terms allowing for a computational time reduction of about 20%. The physical contribution of the removed terms is discussed according to the literature. The modified APE are applied to the prediction of the noise emitted by an orifice. The reliability of the new APE system is evaluated by comparing the Sound Pressure Level (SPL) and the acoustic pressure with the ones returned by LES and literature APE. The new formulation agrees with the other methods far from the orifice: moving over nine diameters downstream of the trailing edge, the SPL is in accordance with the other models. Since international standards characterize control devices with the noise measured 1 m downstream of them, the modified APE formulation provides reliable and faster noise prediction for those devices with outlet diameter, d , such that $9d < 1$ m.

Key words | computational aero-acoustic, flow-control devices, flow-induced noise

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INTRODUCTION

The regulation of noise emissions is a very common issue that affects several fields, ranging from industrial processes to civil infrastructure. Fluid-dynamic noise is a particular kind of noise induced by the excitation of a fluid due to its turbulent motion and it is a common phenomenon, for instance, in Oil and Gas (O&G) systems that convey natural gas. When the flowing fluid is air or gas, such noise is also referred to as aero-dynamic noise. Other examples taken from everyday life are the sound produced by wind turbines or that coming from Heating, Ventilation and Air Conditioning (HVAC) systems. The main issues associated with this type of noise are the possible harmful exposure of either the public or work personnel to sound levels which may be dangerous to their auditory system. Inside industrial plants the aero-dynamic noise adds to that coming from machinery such as pumps, internal combustion engines, fans and rotors. In pipelines, close to singularities such as

bends, junctions and valves, the aero-dynamic noise may reach higher peaks than that coming from the other sources, thus becoming the most potentially dangerous source.

Furthermore, fluid-dynamic noise plays an important role in the emerging field of vibro-acoustic emission (VAE) analysis for leak detection in water distribution networks. As water flows through a leak, turbulence is generated around the hole and transmitted in the form of pressure fluctuations through the fluid and in the form of vibrations along the pipe walls. The pressure fluctuations are then picked up by hydrophones installed inside the pipeline and the vibrations by accelerometers positioned on its wall (Butterfield *et al.* 2018). The power spectra of such measurements provide useful information for the prediction of the size and shape of the leak (Ahadi & Bakhtiar 2010).

It is important to stress the fact that the generation of sound follows the same mechanism (i.e. turbulence) both

in gases and liquids up to cavitating conditions in the latter. In such conditions, vapor bubbles transported downstream of the control-device by the flow implode, generating strong pressure oscillations which rapidly become the main contribution to the measured fluid-dynamic sound. The international standard regulating flow-induced noise emissions in control-devices (IEC 60534-8-4 2015) considers this fact by combining both the contribution due to turbulence and the contribution due to cavitation.

Studies on aero-dynamic noise began in the 1950s thanks to the work of James Lighthill (1952, 1954) on the acoustic emissions of commercial planes' jet engines. He pointed out that the turbulence is itself the acoustic source that dissipates the mechanical energy of the flow, converting part of it into noise. He also proposed an acoustic analogy based on the resolution of an inhomogeneous wave equation whose source term is derived directly from the Navier–Stokes (NS) equations governing the fluid's motion. His work was subsequently extended in order to take into account the effects of the inhomogeneity of the flow, pressure gradients, variations in the speed of sound, presence of solid bodies inside the domain and sound-flow interaction (Curle 1955; Ffowcs & Hawkings 1969; Howe 1975). All acoustic analogies are based on the resolution of a wave equation with a proper source term that depends on the modelling assumptions. Their solution is analytically computable and allows the evaluation of the acoustic pressure in the far-field with an integral. However, their numerical solution requires the storing of the whole temporal series and may result in too much of a computational effort, especially in the case of high Mach number flows. However, no analytical solution is available for the computation of the noise emitted by regulating valves because of the complexity of the flow close to the devices. The prediction of their noise emissions can be performed through numerical methods such as *Direct Numerical Simulations* (DNS) or *Large Eddy Simulations* (LES) (direct approach) and *Acoustic Perturbation Equations* (APE) (hybrid approach). Direct methods evaluate the acoustic field together with the fluid-dynamic one, but their application is not feasible in the applied engineering world because of their demanding computational burden. Hybrid methods, on the contrary, allow the saving of time and resources since they solve the flow

field and the acoustic one in two different steps of lower complexity. The APEs manage the propagation of the acoustic wave but need as input the flow field obtained with computational fluid dynamics (CFD) simulations, either using Reynolds Averaged Navier–Stokes (RANS) equations or LES. Different formulations of APE have been developed in the literature (Bechara *et al.* 1994; Ewert & Schroder 2003) according to the modelling assumptions introduced in their derivation.

The lower computational resources required by the implementation of the APE makes them affordable for the industrial evaluation of valves' noise emissions. Aiming for the greatest time reduction without affecting the quality of the result, in this work a new faster APE formulation is presented for the computation of the noise in the position indicated by the international standards, i.e. 1 m downstream of the device. The study has been performed on a transversal perforated plate (orifice) installed inside a pipe, a fundamental configuration for both flow measurement and regulation in civil and industrial pipelines (Niedźwiedzka *et al.* 2017). This geometry is chosen because the jet created downstream of the orifice is representative of the flow at the outlet of most valves, which consists of a combination of one or more jets coming from different paths. In the literature, experimental studies on the noise emission of perforated plates were conducted by Kirkwood (1992) who worked on the characterization of low-emission orifice plates analysing the influence of different design parameters such as the number of perforations, their diameter and their thickness.

In the next part of the paper, the mathematical and modelling background of the APE is introduced. A new formulation of the APE, based on the simplification of the one found in the literature, is then presented. The modelling of the APE source terms and of the effects of turbulence is also discussed.

The following section investigates the acoustic features downstream of an orifice in order to understand the differences in the resolution of the modified APE with respect to other literature models (LES and APE).

The next section is about the discussion of the evidences described in the previous section and about the feasibility of the application of the modified APE to practical pipeline engineering problems followed by a section summarising

the treated problem and giving a range of applicability for the modified APE.

ACOUSTIC NUMERICAL MODELLING

In this work, a numerical modelling of the noise production of a control device is performed through a second order partial differential equation (PDE) system based on the resolution of the acoustic particle velocity and the acoustic pressure within the domain. [Bechara et al. \(1994\)](#) proposed a system of APEs derived from the Euler equations by means of flow decomposition and proper linearization. The flow is split in three parts, i.e. a mean component $\overline{(\cdot)}$, a turbulent fluctuation $(\cdot)_t$ and an acoustic one $(\cdot)_a$. Considering the fact that the magnitude of the acoustic fluctuations is usually much lower than that of the other contributions, [Bechara et al. \(1994\)](#) applied a linearization of all second order terms containing acoustic variables. The derived system describing the generation and propagation of an acoustic wave in an ideal gas and in a turbulent non-uniform flow is then:

$$\begin{cases} \frac{\partial p_a}{\partial t} + \bar{\mathbf{u}} \cdot \nabla p_a + \gamma \bar{p} \nabla \cdot \mathbf{u}_a + \gamma p_a \nabla \cdot \bar{\mathbf{u}} + \mathbf{u}_a \cdot \nabla \bar{p} = S_p \\ \frac{\partial \mathbf{u}_a}{\partial t} + \bar{\mathbf{u}} \cdot \nabla \mathbf{u}_a + \mathbf{u}_a \cdot \nabla \bar{\mathbf{u}} + \frac{1}{\bar{\rho}} \nabla p_a - \frac{\rho_a}{\bar{\rho}^2} \nabla \bar{p} = \mathbf{S}_m \end{cases} \quad (1)$$

where \mathbf{u} stands for the velocity field, p for the pressure field, ρ for the density field and S_p and \mathbf{S}_m represent the source terms.

The left-hand side of system (1) is a wave operator for the propagation of an acoustic wave in a medium with non-uniform velocity and pressure, conditions commonly found in pipelines.

The right-hand side acts as an acoustic source and contains all the terms that do not include acoustic quantities: it consists of the products of mean quantities with turbulent fluctuations (or their derivatives).

Modified APE

A different formulation of APE is proposed in this work in order to reduce the computational time associated with the resolution of system (1) and to make its numerical resolution feasible in industrial applications. The algorithm for

the algebraic resolution of the discretized formulation of system (1) is based on a predictor-corrector approach with the resolution of the equations for acoustic particle velocity and acoustic pressure in two different steps.

In particular, the matrix associated with the algebraic system for the acoustic particle velocity (Equation 1) is a sparse matrix whose elements out of the diagonal come from the contributions:

$$\bar{\mathbf{u}} \cdot \nabla \mathbf{u}_a + \mathbf{u}_a \cdot \nabla \bar{\mathbf{u}} \quad (2)$$

If the matrix for Equation (1) were diagonal, faster numerical approaches could be used for its numerical resolution reducing the overall computational time, thus meeting one of the most important needs of industrial applications. This is the reason why the purpose of the work presented in this manuscript is the verification of the feasibility of adopting a simplified and faster APE version to be used in the applied engineering issue of predicting the noise emitted by control devices. According to the discussion on the sparsity of the matrix for the acoustic particle velocity, the modified APE system is obtained by removing the terms in Equation (2) and it is written as:

$$\begin{cases} \frac{\partial p_a}{\partial t} + \bar{\mathbf{u}} \cdot \nabla p_a + \gamma \bar{p} \nabla \cdot \mathbf{u}_a + \gamma p_a \nabla \cdot \bar{\mathbf{u}} + \mathbf{u}_a \cdot \nabla \bar{p} = S_p \\ \frac{\partial \mathbf{u}_a}{\partial t} + \frac{1}{\bar{\rho}} \nabla p_a - \frac{\rho_a}{\bar{\rho}^2} \nabla \bar{p} = \mathbf{S}_m \end{cases} \quad (3)$$

The simplification introduced implies that the production and propagation of noise do not consider some effects connected to the interaction of the acoustic particle velocity with the mean flow. A deeper insight into the physics of the neglected terms (2) was presented by [Ewert & Schroder \(2003\)](#) who derived the original APE system through filtering operations on the Euler equations' source terms. In his work, part of the noise production is linked to the term:

$$\nabla \Phi_1 = \bar{\mathbf{u}} \cdot \nabla \mathbf{u}_a + \mathbf{u}_a \cdot \nabla \bar{\mathbf{u}} \quad (4)$$

which corresponds to the contributions neglected in the modified version of the APE proposed in this study.

By exploiting the irrotationality of \mathbf{u}_a , Ewert & Schroder (2003) reformulated Equation (4) as:

$$\nabla\Phi_1 = \nabla(\bar{\mathbf{u}} \cdot \nabla\mathbf{u}_a) + \bar{\boldsymbol{\omega}} \times \mathbf{u}_a \quad (5)$$

where $\bar{\boldsymbol{\omega}}$ is the vorticity of the mean velocity.

Equation (5) highlights the role played by the vorticity of the mean flow in noise generation. In particular, its contribution is most relevant in regions of the flow where shear is important (e.g. boundary layers, free shear flows). In fact, the high gradients of the mean velocity involved in such regions interact with the acoustic field, changing its directivity and thus affecting noise generation (Ewert & Schroder 2003). Neglecting the terms in Equation (5) therefore implies a less refined model for noise generation, whose contribution in the case of valves' noise emissions must be studied in more detail. In fact, the proposal of the international standards is the evaluation of the noise at some distance away from the orifice, where the vorticity of the mean flow and the spatial gradients of \mathbf{u}_a are lower. Therefore, it is assumed that the noise generated by the terms in Equation (5) is local in nature, affecting noise emissions only in the first few diameters downstream of the device.

The next sections are devoted to the investigation of the relative importance of the terms in Equation (5) and to whether the modified APE here proposed can be used in the practical prediction of the aero-dynamic noise emissions of control devices without the loss of fundamental information. The international standards (IEC 60534-8-3;4) regulating the noise emissions of control devices refer to the noise measured at 1-meter distance from the line and 1-meter downstream of the device: the reliability of the modified APE thus needs to be measured on this value.

ACOUSTIC FEATURES OF A PERFORATED PLATE

In this section, the APEs (1) and (3) are tested on the computation of the noise emitted by a perforated plate (orifice) with the perforation's axis aligned with the pipe's axis. The orifice diameter d is equal to 30.5 mm and the ratio β between it and the nominal diameter of the pipe D is equal to 0.4 (D is equal to 3", i.e. 77.5 mm). The edges of the perforated plate are designed with 90° angles and the ratio of the orifice thickness over its diameter t/d is equal to 1.

The computation through the solution of the APE system of the fluid-dynamic and acoustic fields generated by the device is influenced by the turbulence modelling. In particular, the source terms for the acoustic particle velocity and the acoustic pressure in Equation (3) have been described in several ways in the literature. In this work, the approach presented by Bailly *et al.* (1995) is adopted. This model considers the relevance of the acoustic pressure's source term negligible, while it describes that for the acoustic particle velocity as:

$$\mathbf{S}_m = \overline{\mathbf{u}_t \cdot \nabla \mathbf{u}_t} - \mathbf{u}_t \cdot \nabla \mathbf{u}_t \quad (6)$$

It can be seen how such a term depends on both the average and the instantaneous values of the turbulent velocity field. In this work, the mean flow is computed through a time-average of the output of a LES computation, performed on the domain visualized in Figure 1. The turbulence field is then reconstructed from the mean flow according to the Stochastic Noise Generation and Radiation (SNGR) algorithm presented by Bailly *et al.* (1995). This model produces a synthetic turbulent velocity field as a finite sum of Fourier

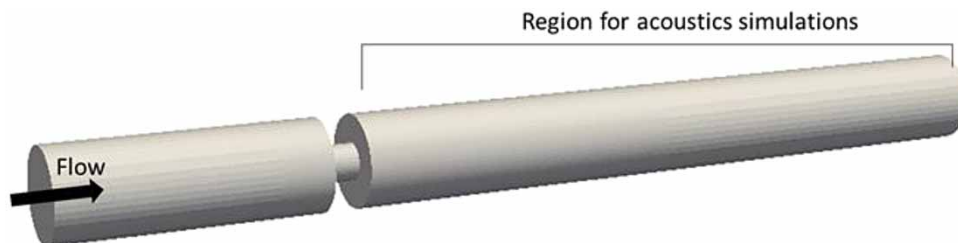


Figure 1 | Domain for LES simulation.

modes correlated both in space and in time according to the turbulence statistics obtained from the averaged flow.

The choice of performing a time-average on the results of a LES simulation rather than employing directly a RANS solver (thus saving computational resources) was completed to better numerically validate the results obtained through the new modified APE. In fact, the model for turbulence employed in the RANS would introduce an additional factor to be considered in the comparison of the results with the other methods. Furthermore, an LES computation provides the acoustic pressure field without the need of a model for the propagation of sound; its drawback being the much higher, and often unfeasible in the industrial world, computational resources required.

The numerical simulation is run on a three-dimensional domain built with an inlet placed $15d$ upstream of the resistor and the outlet $25d$ downstream of it, far enough to avoid the influence of the outlet on the computed fields. The modelling of the subgrid turbulence is made through a one-equation model based on the resolution of a transport equation for the kinetic energy k .

The simulation is run on a uniform, Cartesian grid with a spacing that allows the computation of the sound propagation up to the frequency of almost 70 kHz according to the formula $f_{max} = c_0/5\Delta x$ (Bogey *et al.* 2003) where c_0 is the speed of sound in the free field and Δx is the spacing of the grid.

The dimension of the filter for the application of the sub-grid model is computed as the cube root of the cells' volume. The time spacing is adapted at each time step in order to keep the Courant number lower than 0.75, and the whole simulation is run for about 200 characteristic periods D/U_j where U_j is the velocity of the jet developed at the outlet of the orifice.

The movement of the fluid is imposed through the assignment of boundary conditions with the pressure values on the inlet patch (total pressure equal to 5 bar) and at the outlet (static pressure equal to 4 bar).

With this pressure drop, a Mach number equal to 0.7 is reached inside the jet and the Reynolds number referred to the mean velocity in the pipe is 5.3×10^5 .

The chosen fluid is dry air and, considering the high Mach number involved, the compressibility of the flow is taken care of by selecting an equation of state relating

pressure, velocity and temperature. In particular, the ideal gas law is chosen as a model. Since an additional term has been introduced (namely, temperature) an equation for the conservation of energy is added to the set of coupled equations to solve.

Wave transmissive boundary conditions are also added on these patches in order to avoid numerical reflection of the incident waves. On the other boundaries of the domain (walls of the pipe and orifice) a no-slip condition is imposed while their thermodynamic behaviour is modelled as adiabatic.

The external temperature is imposed equal to the ambient one at 300 K.

The length of the jet (estimated through the time-averaged velocity) is about $4d$ and the mixing layer around the potential core can be clearly seen looking at the instantaneous vorticity field $\vec{\omega}$ (Figure 2).

Vorticity is a fundamental quantity for the fluid-dynamic noise because, according to the Powell–Howe theory of vortex sound (Howe 1975), it is a necessary condition for noise generation. The identification of the acoustic source region is made based on the turbulent kinetic energy κ and it is defined as the region where κ is higher than a percentage of its maximum value assumed in the domain. In this study, such a percentage has been set equal to 20% in accordance with Mesbah (2006). The evaluation of the source region is made with the mean κ computed as half of the trace of the averaged sub-grid scale stress tensor:

$$\kappa = 0.5 (\overline{u'u'} + \overline{v'v'} + \overline{w'w'}) \quad (7)$$

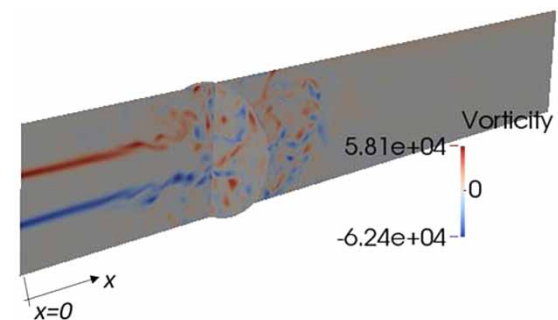


Figure 2 | Instantaneous vorticity field: visualization of the components of $\vec{\omega}$ orthogonal to the surfaces (ω_y in the longitudinal section and ω_x in the transversal section). Flow from left to right.

where u' , v' and w' represent the fluctuating components of the velocity field \mathbf{u} .

Figure 3 shows the distribution of the turbulent kinetic energy so calculated and its intensity suggests that the largest noise generation is located downstream of the potential core of the jet and not in the mixing layer where the vorticity on the contrary reaches its peak.

Sound pressure level on the walls

The flow field described in the previous paragraph is used for the definition of the averaged quantities necessary for the application of the SNGR algorithm and for the APE resolution. In this work, the APE are solved on the same mesh used for the LES and are run for a duration T of the simulation equal to 2 ms, with a time step Δt equal to 10^{-7} s. The duration time is chosen in such a way to allow the acoustic pressure to reach a steady state condition from the initial configuration. Totally reflective boundary conditions are imposed on the walls of the pipe and non-reflecting ones on the numerical boundaries.

The fundamental quantity on which the comparison among the APEs and the LES is performed is the sound pressure level (SPL) evaluated on the walls of the pipe (quantity of interest for the international standards). The SPL is defined through the root mean square (RMS) in time of the acoustic pressure computed in the centre of a given cell:

$$p_{RMS} = p_{RMS}(x_i, y_i, z_i) = \sqrt{\overline{p_a^2}} \quad (8)$$

Such a quantity is in fact capable of capturing both the average intensity of a signal (thus playing the role of a time average) and of the fluctuations around the average (thus playing the role of a standard deviation). The SPL is

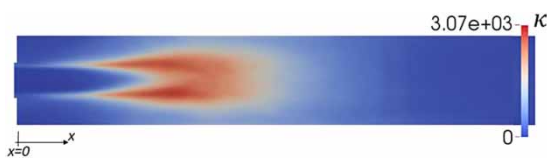


Figure 3 | Time-averaged turbulent kinetic energy k from LES downstream of the orifice. Flow from left to right.

then defined as:

$$SPL = 20 \log_{10} \left(\frac{p_{rms}}{p_{ref}} \right) [dB] \quad (9)$$

where $p_{ref} = 2 \times 10^{-5}$ Pa is the lowest pressure value detectable by the humans' auditory system.

The results in terms of the SPL evaluated at the walls of the pipe downstream of the orifice are shown in Figure 4 for the three different approaches mentioned, i.e. the LES computation, the literature APE system (1) and the modified APE (3) proposed in this work.

The LES and the APE (1) return curves similar in shape: an initial increase of noise starting at around the length of the jet is followed by an initially sharp decrease which gradually softens farther from the orifice. Such a noise distribution along the pipe agrees with the results highlighted by Kirkwood (1992), who performed experimental studies on the noise emitted by different flow-control devices.

The SPL returned by the modified APE (3) shows instead a trend different from the previous two, since it does not display a marked peak value in the position where the others do. Nonetheless, sufficiently far from it, the noise distribution starts to fit the curves obtained with the other two approaches and has their same decrease rate.

The SPL trend highlighted by the three approaches suggests that the main difference between the modified APE and the other models is in the description of the noise not too far downstream of the control device, where

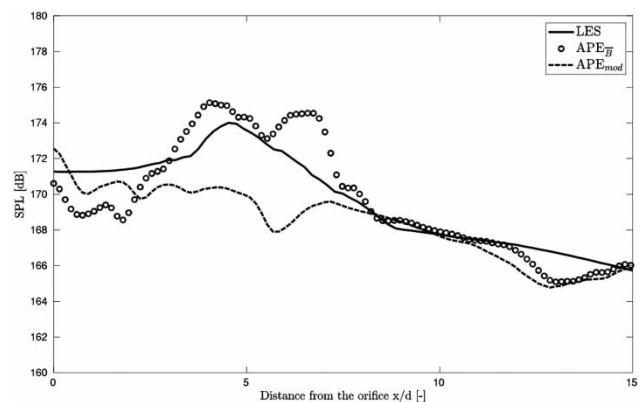


Figure 4 | Sound pressure level downstream of the orifice computed with LES, literature APE (1) and the new APE system (3).

the shear layer of the jet originating from the orifice is dominant. Far enough from the orifice however, the modified APE is capable of capturing the trend of decreasing noise along the pipe similarly to the other two methods.

A more in-depth analysis of the main features of the modified APE system and on its differences with the other models is performed in the next paragraph by looking at the internal acoustic pressure field, which provides a clearer view of the noise generation and propagation mechanisms.

Acoustic pressure inside the pipe

The SPL measured on the walls of the pipe is a quantity which is derived from the acoustic pressure field computed in the internal domain. As already mentioned in the previous sections, both the APE systems and the LES evaluate the acoustic pressure field in all of the regions within the pipe downstream of the orifice using different modelling assumptions. The temporal evolution of the acoustic pressure from a silent initial configuration identifies the areas within the pipe where the acoustic fluctuations are generated and describes their propagation as part of the acoustic wave. A graphical visualization of this information is achieved by looking at the acoustic pressure field on a longitudinal plane at a given time instant, as shown in Figure 5.

The distribution of the acoustic pressure and the properties of the mean flow define four regions downstream of the orifice, characterized by different features.

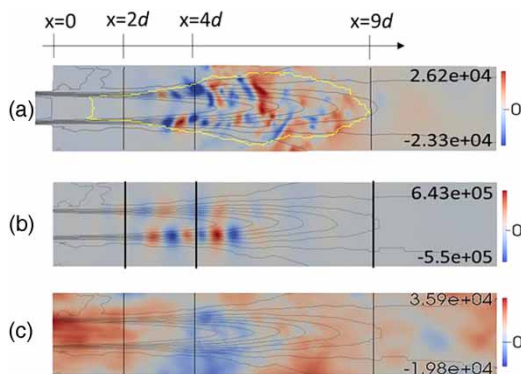


Figure 5 | Instantaneous acoustic pressure p_a on a longitudinal plane downstream of the orifice (in black the contours of the mean axial velocity): (a) Field returned by LES and acoustic source region in yellow; (b) Field returned by literature APE (1); (c) Field returned by modified APE (3). Please refer to the online version of this paper to see this figure in colour: <http://dx.doi.org/10.2166/hydro.2020.156>.

A first region may be identified by looking at the acoustic pressure field returned by the LES and the literature APE approaches. In the area going from immediately downstream of the device to about $2d$, both of the two methods show a substantially flat behaviour for the acoustic pressure field before displaying a highly spatial variability in the following region. From a fluid-dynamic point of view, such a region corresponds to the first half of the inner potential core of the orifice-generated jet, where the shear layer is still limited in thickness.

The second region coincides with the one showing a mostly regular spatial variability of the acoustic pressure for the results returned by the LES and the literature APE, which goes from $2d$ to about the end of the jet's inner core at $4d$. In this section, structures of negative and positive acoustic pressure follow each other in a manner which suggests that they may be caused by the interaction of the increasing thickness of the jet's shear layer and an acoustic field generated further downstream.

The acoustic wave that generates the previous structures most probably comes from the third region $4d < x < 9d$ which, according to the turbulent kinetic energy field displayed in Figure 3, is the most energetic one. The distribution of the acoustic pressure computed with LES and the literature APE (Figure 5(a) and 5(b)) shows a rather different evaluation of the intensity of the spatial structures produced in the shear layer amongst the two approaches. These structures, after their generation in region two, are convected downstream by the mean flow and initially grow in intensity, only to show a fast decay within the same region.

Both LES and APE (1) describe this phenomenon, the difference being that the latter computes the structures' intensity an order of magnitude higher than the LES. Indeed, in APE (1), once those structures move to $4d < x < 9d$, they are more intense than any other possible acoustic contribution coming from other regions, thus hiding it. Contrarily, such contributions are visible in the LES approach.

As already stated, in both cases the noise in the region $4d < x < 9d$ shows a fast decay after reaching its peak intensity. This decay continues up to the limit of the acoustic source region, i.e. about $9d$ downstream of the orifice.

The fourth region may thus be identified as that for $x > 9d$, where no noise is generated, and the acoustic wave is just propagated along the duct.

The acoustic pressure field returned by the new modified APE (3) is qualitatively different from those returned by the two methods discussed above and no structures are generated in the shear layer. Such a difference is expected since the removed terms of Equation (2) are most important in regions where high gradients are present, that is, exactly in the shear layer. Therefore, noise generation in region two is not caught by the modified APE and the acoustic pressure field within the pipe is just influenced by the propagation of the noise generated in the region between $4d$ and $9d$.

DISCUSSION

The results presented in the previous sections have highlighted the main features to consider in the evaluation of the reliability of the modified APE system proposed in this work, i.e. the noise measured on the walls and the evolution of the acoustic pressure within the pipe.

The comparison with the LES and with the literature APE (1) suggests that, far from the orifice, the modified APE (3) returns a noise estimation in agreement with more complex and complete models. With the studied device, the limit over which the modified APE may be used in industrial applications is identified as the distance of $9d$ where d is the diameter of the orifice's perforation. As shown in Figure 4, over that distance, the SPL perfectly matches the results from literature APE (1) suggesting that no errors connected to the removal of the terms in Equation (2) are introduced in that region.

Conversely, in the region $x < 9d$ the computed noise is affected by modelling errors and the modified APE system is not able to describe the local increase of noise nor the intensity of its peak located at about $4d$ or $5d$.

In industrial applications, control devices' manufacturers are interested in their acoustic characterisation according to international standards. These regulations identify the noise 1 m downstream of the device and 1 m far from the pipe as the quantity characterizing their acoustical emissions. The variable of interest in a numerical simulation is thus the internal noise intensity 1 m downstream of the device (transmission through the walls and propagation in the external space are managed by analytic equations not relevant in the context of this work). Since

the modified APE formulation (3) returns the same SPL prediction as APE (1) downstream of $9d$ with a 20%-reduced computational cost, it can be applied in industrial frameworks for the characterization of those control devices with an outlet diameter d such that $9d < 1m$.

A reason for the SPL behaviour is given by the acoustic pressure fields shown in Figure 5. The most important feature is the evolution of the structures of alternate positive and negative acoustic pressure that are generated in the shear layer. Their propagation in the domain must be analysed looking both at the SPL and at the acoustic pressure: the comparison between LES and APE (1) on the SPL suggests that no effective differences are present in the noise behaviour in the region between $2d$ and $9d$. On the other side, the intensity of the acoustic pressure structures in the shear layer computed with the APE (1) is one order of magnitude higher than the one computed with LES. These two aspects imply that, even though the APE (1) overestimates noise production in this region, the structures are dissipated so fast in the inner section of the flow that their effect on the walls of the pipe is negligible.

The fast decay of the noise generated in the shear layer is the reason why the SPL computed with APE (3) is basically coincident with the one returned by APE (1) further than $9d$, even though the acoustic pressure field is substantially different close to the orifice. The noise in the shear layer has only a local effect that is evident in the hill-shaped profile of the SPL close to the orifice. Moving downstream, the effective noise that propagates through the pipe is only the one generated in the region between $4d$ and $9d$. The irrelevance of the noise contained in the structures on the evaluation of the SPL far from the jet therefore allows to neglect those terms in the new APE system and to apply them for those devices with an outlet diameter such that $9d < 1m$.

CONCLUSIONS

In this work, a new APE system has been presented for the prediction of the noise emitted by control devices in the position indicated by the international standards. The new system is derived from a literature APE system neglecting two terms with a simplification of the numerical system that involves a reduction of computational time of about 20%.

Literature studies suggest that those two terms are responsible for the description of noise generation in sheared regions, i.e. where high velocity gradients and high vorticity are present. The feasibility of the application of the modified APE to industrial problems has been investigated on an orifice. It has been shown that the noise prediction is in accordance with the one returned by more complex and accurate methods far enough from the control device and, in particular for the tested orifice, for $x > 9d$.

Therefore, according to the international standards, the modified APE can be used for those devices whose outlet diameter d is such that $9d < 1m$.

Further developments should consider the use of simpler approaches for the computation of the average flow field than LES such as RANS and the influence of the chosen turbulence model.

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