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NUMERICAL SIMULATION AND EXPERIMENTAL STUDY OF THE PROPAGATION OF SOUND WAVES IN A PIPE FOR FLOW MEASUREMENT

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Abstract. *In this work one aims to determine the speed of sound and the flow velocity experimentally and the results are used to validate a numerical implementation. Acoustic signals generated by a loudspeaker of variable amplitude and frequency are used in the experiments. There is a need to measure, a priori, the speed of sound in the medium at flow conditions. Therefore, an experimental device was constructed, consisting of a horizontal tube, an excitation source, and pressure sensors, in order to determine the speed of sound and the flow rate. Experimental tests demonstrated the potential of the method in determining the speed of sound and flow rate. The numerical simulation was satisfactory when compared with the experimental results.*

Keywords: *single-phase flow, sound propagation, wave speed*

1. INTRODUCTION

Navier-Stokes equations are named after the studies of engineer and physicist Claude Louis Marie Henri Navier and mathematician and physicist George Gabriel Stokes, and designate the movement of viscous fluid substances. They are based on the amount of linear movement, caused by viscous forces and pressure difference, and the conservation of mass in an infinitesimal volume, in addition to other physical quantities.

Such equations are of paramount importance in several areas of knowledge. They can be used to model climate change, ocean currents, flow in pipelines, airfoil and aircraft projects, blood flow studies, among other applications (Elias, 2007). The movement of these fluidic substances can be understood as a flow. From thermodynamics, the phase of a substance is designated as a solid, liquid or pure gas (Lima, 2011). Therefore, it can be said that a multiphase flow is composed of two or more phases flowing simultaneously in a given domain at the macroscopic level. In addition, it is customary to refer to multiphase flow as the simultaneous flow of immiscible components or substances, even in cases where they might have different chemical compositions.

The fields of application are numerous, many of which are of interest to the oil and gas industry, such as in the case of flare gas in which there are liquid droplets (dispersed phase) in the gas (continuous phase). The continuous phase can be represented by a liquid or gaseous medium and the dispersed phase can consist of solid particles, gas bubbles or drops of liquid. Therefore, a multiphase system can be classified as a region or system where two or more non-miscible fluids coexist, separated by a continuous (stratified flow) or dispersed (dispersed bubbles or suspension flow) (Paiva, 2011) interface.

Multiphase flows have particular representations in relation to single-phase classification, which are the flow patterns. They are classified according to the arrangement between the phases within the control volume that are dependent on the physical properties of the phases, characteristics of the piping and, also, of operational conditions, such as: temperature, pressure, flow rates, etc. The proper modeling of a multiphase flow requires prior knowledge of the spatial distribution of the phases, that is, how the forces at the phase interface relate to the flow pattern (Ishii and Hibiki, 2007). Therefore, it is necessary to know the flow pattern so that the problem is modeled in a coherent way. Fig. 1 represents the most usual flow patterns for horizontal or slightly inclined flow.

The stratified smooth (SS) pattern is characterized by the total separation between the phases due to the difference in density of the phases, in which the fluid with higher density flows at the bottom of the cross section of the pipe. This pattern can be observed when the superficial velocities of liquid and gas are small enough that ripples on the interface between the phases are negligible or nonexistent.

The stratified wavy (SW) pattern is also characterized by the total separation between the phases due to the difference in density of the phases, with the most dense phase flowing below the less dense one. This pattern can be observed when small waves are formed on the interface between the fluids.

The annular pattern (AN) is characterized by gas flow on the central part of the pipe at high speeds and a liquid film formed on the tube wall. Due to the gravitational effect, the thickness of the liquid film at the bottom is greater than at the top, creating an asymmetry in the film profile.

The intermittent pattern, the buffer pattern and the sluggish pattern, are formed by the coalescence of the gas bubbles as the superficial velocity of the gas increases, generating elongated bubbles that flow in the upper part of the pipe, in general. As the gas speed is increased, the gas bubble occupies the entire cross section of the pipe, having an alternation between a gas bubble and a fraction of liquid being pushed by the gas.

The dispersed bubbles (DB) pattern is characterized by high liquid velocities, where there are gas bubbles dispersed in the liquid.

The mist-flow pattern is observed when the surface velocities of the liquid and gas are high, causing the drops of liquid dispersed in the gaseous nucleus to flow at speeds close to that of the gas.
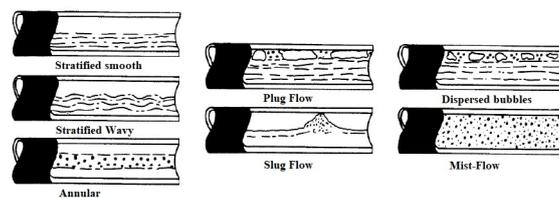

Figure 1. Flow patterns in horizontal ducts, adapted from (Silva e Costa, 2000)

As explained above, the flow pattern depends on operational conditions, such as temperature, pressure, surface velocity of the gas and liquid, in addition to geometric characteristics. Therefore, in a simplified way, the flow patterns can be plotted according to the surface velocities generating the flow map, according to the Fig. 2.
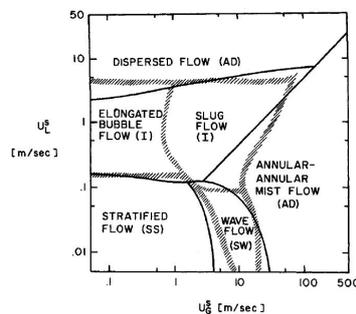

Figure 2. Flow chart for water-air in ambient conditions for a tube diameter of 2.5 cm, taken from (Taitel and Dukler, 1976)

Multiphase flows are frequently present in nature and, with this, it is necessary to understand the processes that occur for a better improvement of instruments for measuring the properties of these flows. Due to the high degree of complexity of the phenomena that encompass the problem, accurate predictions can only be found through instrumentation and/or numerical methods.

The basic principles of fluid mechanics characterize the study of sound propagation in fluids, and, with the appropriate initial and boundary conditions, they form a set of partial differential equations. With computational advances, mathematical models have been increasingly used for fluid analysis. Besides, significant advances in computational methods have been developed in recent decades, especially in areas involving fluid dynamics (Mittal and Tezduyar, 1995; Bazilevs *et al.*, 2013; Court and Fournié, 2015; Arslan *et al.*, 2017, 2018; Cohen, 2002). The Finite Element Method (FEM) is a method consolidated in its applications, gaining a special prominence in fluid mechanics due to the significant advances in computational tools.

Conventional flow meters inserted directly into the pipeline, such as an orifice plate, inverted cone, or turbine sensor, disturb the flow, causing pressure losses and decreasing the efficiency of the system (Gorny *et al.*, 2012; Raine *et al.*, 2015). A non-intrusive flow measurement method is desirable in this context, as proposed in this work, based on acoustic signals. In the current project, an experimental and non-intrusive numerical method is proposed for flow measurement based on the propagation of acoustic signals. This method aims to improve the efficiency of gas flow measurement systems and finds application in the oil industry and, in particular, in the measurement of flare gas flow. For this, there is a need to measure the speed sound of the medium, which is air (single-phase). In sequence, the air velocity is measured, and the

experimental results will be used to validate the implementation at numerical simulation. To circumvent the complexities of the problem, the public domain library FEniCS was used for the numerical implementation.

2. METHODOLOGY

When an acoustic wave passes through a fluid, its local density, pressure, and temperature vary periodically over time. In a first approximation, it is possible to assume that these variations occur in an adiabatic and reversible way due to the oscillation speed. Therefore, it can be assumed that a plane harmonic wave can travel without being significantly attenuated and with speed defined by the fluid state's compressibility and equation. A mathematical description for this movement can be constructed from the laws of conservation of mass, amount of movement, and energy, specified for an ideal non-viscous and non-absorptive fluid, with first-order variations in state and fluid dynamics variables (Martins, 2011; dos Santos, 2010). When considering a mean velocity, the same assumptions are considered. Thus, the sound wave equations with mean flow unidimensional are (Pridmore-Brown, 1958),

$$\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} = \left[1 - \left(\frac{V}{c}\right)^2\right]\frac{\partial^2 p}{\partial x^2} - \frac{2V}{c^2}\frac{\partial^2 p}{\partial x \partial t}, \text{ in } \Omega \times (0,T), \tag{1}$$

$$p = 0, \text{ in } \Gamma_{out} \times (0,T), \tag{2}$$

$$p = p_{in} = \sin\left(2\pi f t\right), \text{ in } \Gamma_{in} \times (0,T), \tag{3}$$

$$p^0 = 0, \text{ in } \Omega^0, \tag{4}$$

$$\frac{\partial p^0}{\partial t} = 0, \text{ in } \Omega^0, \tag{5}$$

where c is the sound speed, V is the mean flow velocity (considered constant), p is the acoustics pressure, f is the frequency of loudspeaker, and Eqs. (2) - (5) is the boundary and initial conditions, respectively. The group of equations was solved numerically using the finite element method, with support of the public domain library FEniCS.

The experimental approach is described in Fig. 3. The sound pulses were generated with a loudspeaker and captured by pressure sensors. The time interval in which the pulses pass through the pressure sensors (microphones) was analyzed. The distance between the microphones is measured, and the wave speed is determined by the speed definition, that is,

$$V_r = \frac{\Delta L}{\Delta t}. \tag{6}$$

When a sound wave propagates in a non-stationary medium ($V \neq 0$), the propagation speed of this wave is the addition or subtraction of the speed of sound in the medium with the flow velocity (Gorny *et al.*, 2012), determined by Eq.(6). Therefore,

$$c - V = \frac{\Delta L}{\Delta t}, \quad \text{Upstream} \tag{7}$$

$$c + V = \frac{\Delta L}{\Delta t}, \quad \text{Downstream} \tag{8}$$

where, c is the speed of sound in the medium, and V is the flow velocity. The loudspeaker was placed in the middle of the tube, and the sound pulses were generated and captured by pressure sensors upstream and downstream at the loudspeaker. The flow was generated by a fan coupled to an electric motor, in which a frequency inverter modified the rotation of the electric motor in order to vary the flow velocity. The reference velocity was obtained using an iCEL AN-3070 turbine type anemometer for validation, located in the tube outlet section. For temperatures ranging from 18°C to 28°C, the equipment's accuracy is $\pm (3\% + 0.2)$ m/s.

3. RESULTS

To calculate the sound wave propagation, an analysis of the signal over time is made, that consists of emitting a sound pulse that propagates through the tube in a waveform. When the microphones measure this pressure signal generated by the pulse, the acquisition system registers the signal in time, knowing the distance between the microphones positioned along the pipe, then speed of sound and flow velocity can be calculated. Flow velocity was measured for three different fan rotation, and the temperature was measured to compare the speed of experimental and theoretical sound. The reference velocity was obtained by positioning the anemometer, turbine type and model iCEL AN-3070, at the tube outlet. The
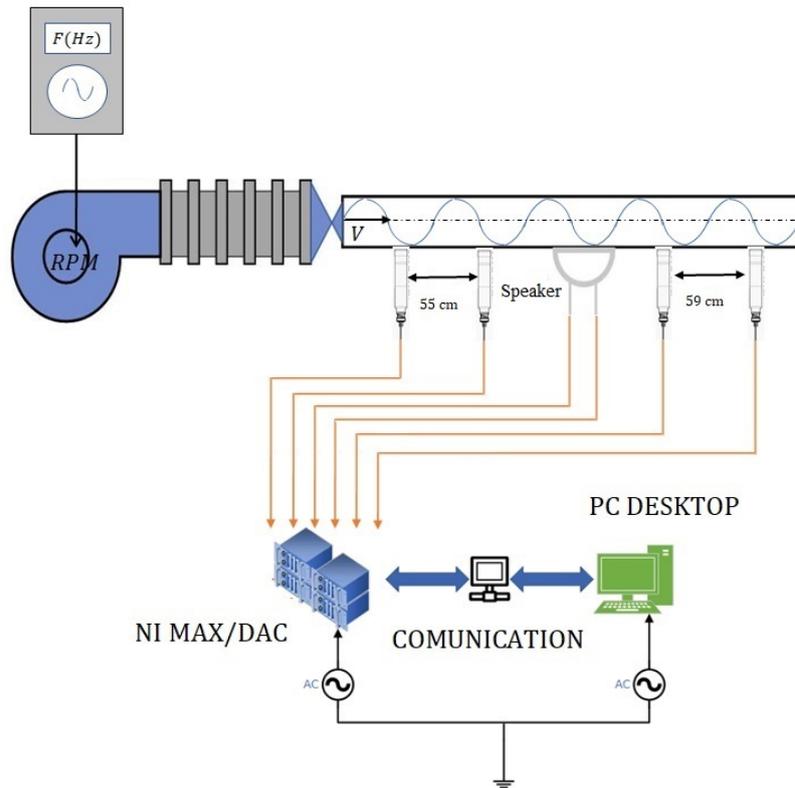
Figure 3. Schematic figure of the experiment

Table 1. Experimental results for three different fan rotation.

Fan (RPM)	T (°C)	V_{exp} (m/s)	V_{fan} (m/s)	C_{exp} (m/s)	C_{theo} (m/s)
583.33	24.5 ± 1	8.25 ± 2.13	6.78 ± 0.40	343.86 ± 2.13	346.02 ± 0.579
1167.67	24.9 ± 1	12.65 ± 2.16	14.08 ± 0.62	348.25 ± 2.16	346.25 ± 0.579
1750	26.6 ± 1	24.51 ± 2.14	21.38 ± 0.85	345.86 ± 2.14	347.23 ± 0.577

accuracy of the equipment is $\pm (3\% + 0.2)$ m/s for temperatures between 18 °C to 28 °C and, as the experimentally measured temperatures were within of this range, the measurement uncertainty using the anemometer could be determined. The results are presented in the Tab. 1.

It is noticed that the speed of sound, determined experimentally, is close to the theoretical one. The existing difference is related to experimental inaccuracies and air as an ideal fluid to determine the theoretical speed of sound. Furthermore, the method proved to be satisfactory for obtaining the flow velocity, reaching errors below 20%, and, considering the uncertainties, these errors are within the velocity range provided by the fan. However, there is a need for an improvement in the analysis. When greater distances between sensors are adopted, these errors tend to reduce since the uncertainties related to measurements start to influence the final result. We can also observe that, even with low errors for the speed of sound, the flow speed presents relatively larger errors. This is due to the large differences between the magnitudes of the velocities (sound and flow) since the equations used by the method are linear, Eqs. (7 - 8). Therefore, a small difference in determining the speed of sound results in a relatively larger error in the flow velocity.

The speed of sound in the medium is a function of the temperature. With this, the ambient temperature was measured, and the theoretical speed of sound was calculated for comparison with the experimental data obtained, as we can analyze in Tab. 1. In Fig. 5, the percentage error between the speed of sound measured experimentally, and the theoretical one is presented. We can assess that the results are consistent with the theory. The smallest error found is for the fan rotation in 1750 RPM, the most significant error being for the fan rotation in about 580 RPM. It is worth mentioning that the most significant error found was approximately 0.62%.

Equations. (1 - 5) were solved numerically using the finite element method, and the results obtained are presented in the Tab. 2. Considering the analysis analogous to the experimental tests, the temporal results were extracted in four different positions in the tube, and, with this, the flow and sound velocity were determined. For each experiment, the means and standard deviation are obtained, as shown in the Tab. 2. It can be seen that, for the flow velocity, the most
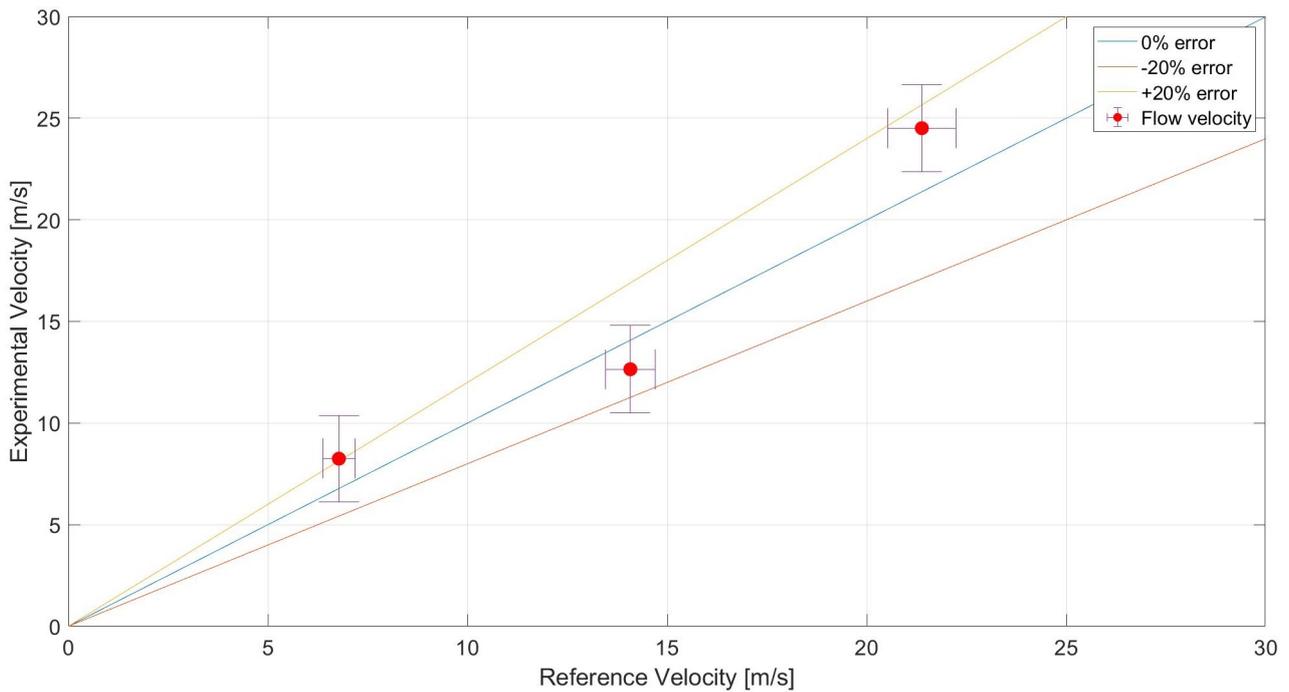
Figure 4. Experimental velocity compared to the reference velocity, along with the associated uncertainties and errors.
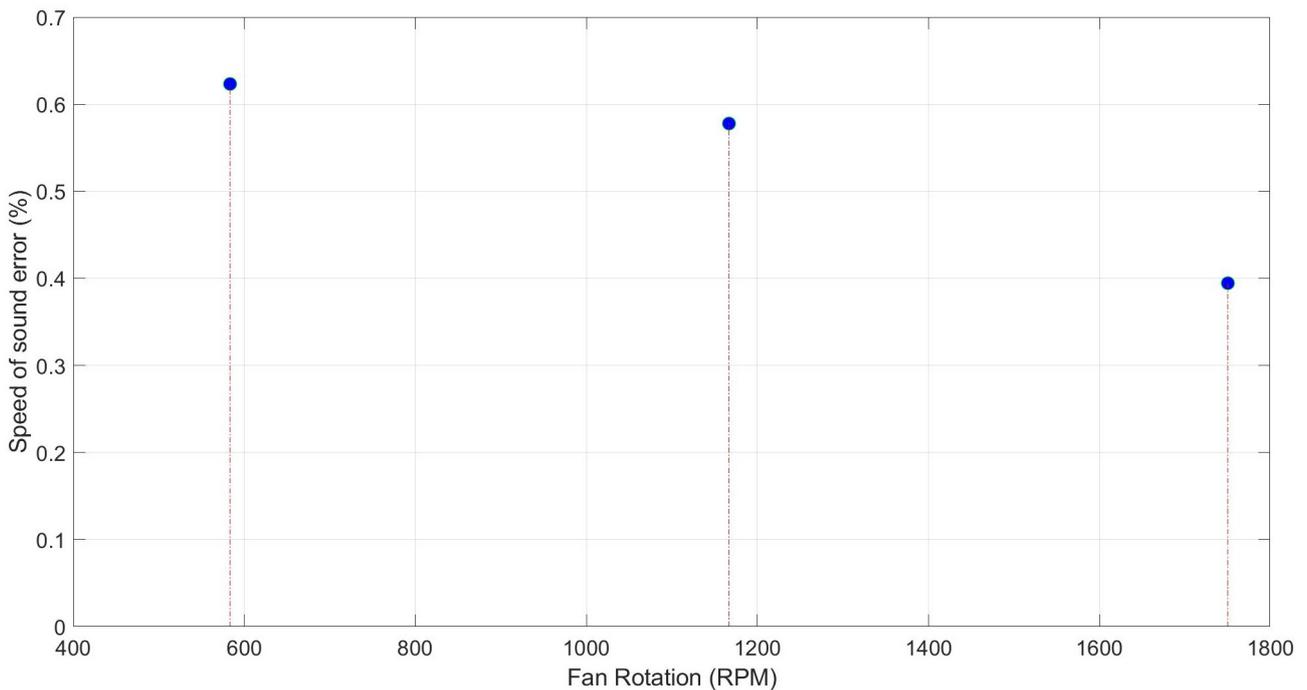

Figure 5. Percentage error between theoretical and experimental speed of sound

massive percentage error obtained between the experiment and the numerical simulation was approximately 6%. For the speed of sound, the most massive percentage error obtained was approximately 0.4%. Similar to the experimental procedure, it is noted that even obtaining a low error for speed of sound can be caused by a massive difference in the flow velocity. This is due to the massive difference between the magnitudes of the speed of sound and flow since the equations used are linear, Eqs. (7 - 8).

4. CONCLUSION

This work presented a numerical implementation using the finite element method and an experimental approach to study sound wave transit time between sensors.

Table 2. Numerical results using the experimental data, presented in the table1.

$C + V$ (m/s)	$C - V$ (m/s)	$\bar{\sigma}$ (m/s)	V_{num} (m/s)	C_{num}
352.20	335.40	0.053	8.40	343.80
361.00	334.93	0.084	13.03	347.96
370.41	318.29	0.19	26.06	344.35

The preliminary experimental results obtained are encouraging for both the speed of sound and the speed of flow. Errors less than 0.7% for the sound speed were obtained and the smallest error of 10% for the flow speed. However, slightly high differences were found in the flow speed. The ways to improve these results are: to adopt greater distances between the sensors and improve the post-processing of the signals.

The numerical implementation was consistent with the experimental data, resulting in errors of less than 6% for the flow velocity and 0.4 % for speed of sound. More realistic boundary conditions must be implemented to improve the results.

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