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Research on Fluidic Amplifiers Dimensional Modifications via Computer Simulation (CFD)

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Abstract: When studying active flow control applications, it is already stated that activating the boundary layer via using periodic flow produces better performance than when employing steady blowing or sucking. This is why studying the performance characteristics of devices like fluidic oscillators and zero net mass flow actuators is particularly interesting. In the present paper a particular configuration of fluidic oscillator is carefully analyzed, initially its dynamic performance is compared with experimental results undertaken by previous researchers, then the dimensional internal characteristics are modified in order to obtain how is the dynamic behavior being affected, in a third stage the evaluation of how fluidic oscillators scale is affecting its dynamic performance is also presented. Based on the results obtained it can be concluded that a given actuator working at a given Reynolds number, is capable of producing different frequencies and amplitudes when modifying some dimensional parameters.

Keywords: Active flow control, Fluidic oscillators, Computational Fluid Dynamics, Turbulence Modeling.

1 Introduction

Flow control actuators have long been in the focus of research in the fluid mechanics' field since they are able to reduce drag on bluff bodies, increase lift on aerofoils and enhance mixing. Their performance in real applications, must assure reliability and long lifetime. Among the different existing actuators, ZNMF (zero net mass flow), plasma actuators, MEMS (Micro-Electro-Mechanical Systems), fluidic oscillators and combustion driven jet actuators [1,2], only the plasma, fluidic and pulsed combustion actuators do not have moving parts, which a priory gives confidence regarding their reliability. At the moment, plasma actuators are not fully able to produce the needed momentum to modify the boundary layer in a real application, since it appears that the voltage differential used is not sufficiently ionizing the fluid to create the necessary fluid jet momentum. Pulsed combustion actuators provide a huge flow momentum, although, due to the combustion created temperatures, such actuators can just be used for very specific applications in which high fluid temperatures are tolerable. Fluidic oscillators, on the other hand, are able to produce a pulsating jet with the required momentum, although it appears that their design needs to be adapted to each particular application. It must be taken into consideration that nowadays the use of MEMS is steadily increasing, especially in the microfluidics field [3, 4], where small amount of flow is required. ZNMF actuators have been and are extensively studied and used, some relevant papers on development and applications are [5-9]. They provide enough momentum to modify the main flow boundary layer and thus to maintain high vorticity flux downstream. However, despite the fact that ZNMF are widely used, their reliability might be compromised due to their moving parts.

Original fluidic actuators design goes back to the 60s and 70s, left nearly unchanged for over 40 years. Their possible output frequency ranges from several Hz to KHz and the flow rate is usually of a few dm^3/min. Among their applications in flow control, it is worth to mention their use in combustion control [10-12], mixing enhancement and flow deflection [13], modifying flow separation in aerofoils [14], boundary layer control on hump diffusers used in turbomachinery [15], flow separation control on stator vanes of compressors [16], drag reduction on trucks [17] and cavity noise reduction [18].

Following the present introduction, it appears that fluidic actuators could be much widely used in the near future, and it is according to the authors, worth to better understand their behaviour in order to further improve their performance.

Regarding the fluidic oscillator design two main groups exist, the one based on Coanda effect [19], and the one based on a jet mixing chamber, also called vortex oscillators [20]. The former group had an early application as pressure, temperature and flow measuring devices [21-23], the latter group has recently been applied as a flow control device [24].

Depending on their application, fluidic actuators shall produce pulsating flow at a range of different frequencies and flow rates. To push forward such boundaries several fluidic oscillators' designs have been recently created. Uzol and Camci [25] studied experimentally and by Computational Fluid Dynamics (CFD) a fluidic oscillator based on two elliptical cross-sections placed transversally and an after-body located in front of them. Such configuration was in fact proposed by Bauer's patent [26, 27]. The device operates at frequencies of around 30 Hz and under laminar flow. The relation frequency versus Reynolds number was found to be perfectly linear.

Huang and Chang [28] performed a deep experimental study on a V-shaped fluidic oscillator. Playing with the dimensions and the internal oscillator circular cavity, they defined the regimes under which oscillation was generated and they proved that frequencies from few Hz to several KHz could be obtained by modifying oscillator parameters. Additionally, an analysis of the streamline patterns behind the oscillator was also presented. Khelfaoui et al [29], presented an experimental and numerical analysis of non-symmetrical mini and micro oscillators. The numerical analysis being based on a hybrid simulation, they simulated the central part of the oscillator by CFD, while the oscillator feedback was modelled analytically. They found a linear relationship between the actuator frequency and the feedback channel volume, and noticed that above a certain input pressure choked flow appeared. From this point on, the relation frequency versus pressure threshold difference decreased linearly. Gebhard et al [30] studied a micro-oscillator operated with water, finding a linear relationship between the output frequency and the input volumetric flow. Raman and Raghu [18] evaluated the decrease of a cavity tone by using fluidic oscillators. The main acoustic frequency was reduced by over 10 dB, concluding that fluidic excitation is a candidate in noise control applications. A numerical simulation of a two dimensional fluidic oscillator by using Navier-Stokes equations in laminar and incompressible flow, was performed by Nakayama et al [31]. They were able to visualize the periodical flow movement and measured the temporal axial and tangential fluid velocities, oscillation frequency being of 40Hz.

Gregory and Raghu [32], created quite recently a fluidic oscillator based on Coanda effect but driven by piezoelectric devices. One of the main interesting performances of such device is that the oscillating frequency can be decoupled from the input flow and pressure differential. Frequency just depends on input electrical signal, being the oscillator able to work at a range of velocities which goes up to sonic conditions.

At this point it seems clear that most of the research being done on fluidic actuators is focused in evaluating new configurations [20,25,28,29,32], performing numerical or CFD models, often under laminar conditions [25,29,30,31] and mostly on evaluating their performance experimentally whether by themselves or in a given application [10-17,20-25,28-29,18,32].

The present paper will present a numerical evaluation of a fluidic actuator. In fact, the fluidic actuator presented here was previously studied in [33-35]. They performed and extensive CFD model including the analysis of several turbulent models in order to find out which one was the most appropriate. Besides, they performed an experimental study obtaining a good agreement between experimental and CFD results. In the present paper, experimental results obtained in [33, 34] will be compared with the new CFD calculations. Finally, a discussion regarding how different fluidic oscillator parts and dimensions may affect its performance will be carried on. The idea is to give the reader some hints to be able to modify a given oscillator to fulfil a particular application.

It needs to be clarified that the information presented here has been obtained via 3D simulation, SpalartAllmarasDDES turbulence model was employed in all cases studied. Qualitative information obtained from the present study, allow to predict which modifications are worth studying in more detail due to their particular relevance.

2 Problem Statement

According to the information gathered, many of the CFD simulations performed on fluidic actuators were carried out in laminar flow. In fact, inside fluidic actuators, especially when high speeds are required, and or low density fluids are used, the flow is expected to be turbulent. In the present study flow will be considered as turbulent, incompressible and isothermal.
The fluidic oscillator used for the present work is presented in figure 1, notice that the representation is 3 dimensional, and in the present paper the CFD model studied will also be fully 3D. A very similar fluidic actuator design was already used in combustion control [10-12] giving notorious benefits regarding the combustor stability behaviour.

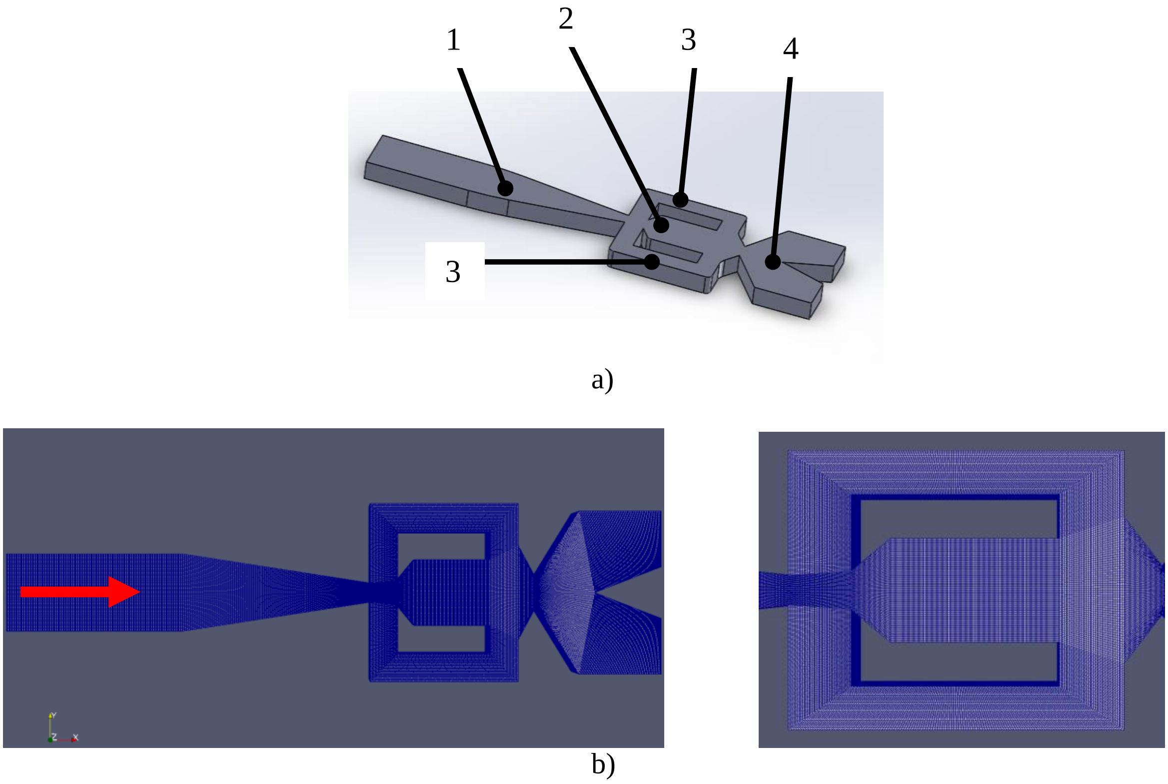


Fig 1 a) Main view of the fluidic actuator. b) Grid used in the present study and zoomed view of it.

Notice that the fluidic actuator consists of an input section (1), a mixing chamber (2), where the feedback channels (3) can be seen on both sides, and an external chamber (4), the two outlets are to be seen at the end of it. A zoomed view of the grid used to perform the simulations is presented in figure 1b. The grid it is of structured type and consists of 2242000 nodes. Boundary conditions employed were, fluid velocity at the entrance and absolute pressure $1.01978*10^5$Pa at the output, Dirichlet boundary conditions were set to all walls. A range of different input velocities from 0.758 to 1.23 m/s were studied, it's minimum and maximum Reynolds number associated was 8711 and 16034. The fluid employed was water and it was considered as incompressible. Fluid dynamic viscosity was chosen as 0.001003 Kg/(m s) and fluid density was 998.2 Kg/m^3. The characteristic length was chosen to be the inlet width, which value was $2.55*10^{-3}$m. The turbulence model used was the SpalartAlmaras DDES, which is a hybrid LES model. The Open Foam, version 3.0, open source package was employed for all 3D simulations, finite volumes approach was employed. Inlet turbulence intensity was set to 0.05% in all cases; PISO was used as a solution method, being the time step of 10^{-5}s, spatial discretization was set to second order. The initial tests were done employing two different grid sizes; the number of cells were respectively of 142000 and 2242000. Reynolds number was set to 8711. The course grid produced an oscillation frequency of 24.6Hz while the fine one gave an oscillation frequency of 22.7Hz. When comparing these values with the experimental results undertaken by [33, 34], it was noticed that for the coarse grid, the error produced was of 12.8%, while when using the fine mesh, the error reduced to 4.1%, the authors accepted the fine mesh as accurate enough and this particular mesh was used for the rest of the simulations presented in the present paper. Regarding the computational time, simulations required about 6 hours when using the course mesh, this time increased to 46 hours when the fine mesh was employed.

In order to further validate the CFD model, four different Reynolds numbers, 8711, 11152, 13593, 16034, based on the fluidic amplifier inlet dimensions and velocity, were simulated, the comparison with the experimental results obtained in [33, 34] is presented in figure 2. The agreement is very good, being the maximum frequency difference of 0.9Hz at the maximum Reynolds number evaluated.

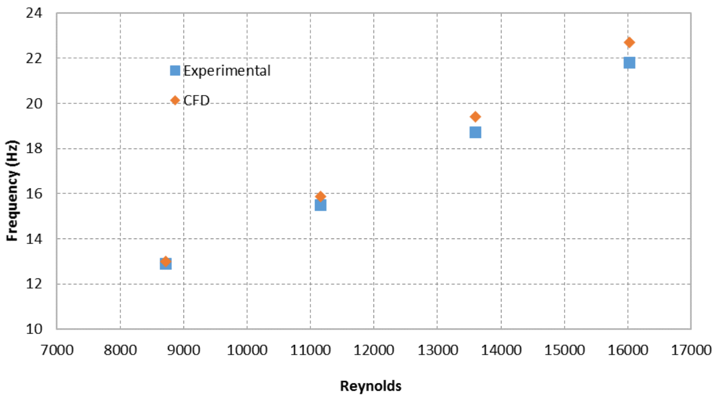


Fig 2 Comparison experimental and CFD results.

3 Fluidic amplifier internal dimensions' modifications

As stated in section 2, it seems that a fluidic actuator (FA), has a single outgoing frequency for a given incoming Reynolds number, yet, the authors believe, the internal geometry must play a role regarding this point. This is why in the present paper; it will be studied the output frequency dependency on the mixing chamber shape. In order to perform this evaluation, three different chamber modifications were undertaken, see figure 3. Inlet and outlet mixing chamber width's, as well as outlet inclination angle were modified while maintaining constant the incoming Reynolds number at 8711.

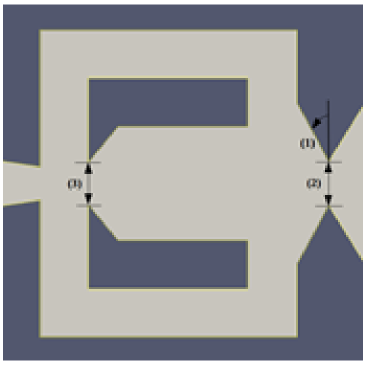


Fig 3 Main view of the fluidic oscillator mixing chamber.

Based on the generic dimensions, the outlet inclination angle, measured anticlockwise versus the vertical central axis, was increased and decreased by about 70%. Outlet mixing chamber width, was modified by approximately ±40%, and finally mixing chamber inlet width suffered modifications of around minus 65% and plus 108%. In the next section, the fluidic amplifier flow performance, when considering each of these three modifications, is evaluated.

4 Results

4.1 Output frequency and amplitude variation when modifying mixing chamber outlet angle.

Figure 4 presents the fluidic actuator output frequency and amplitude when modifying the mixing chamber outlet angle. This particular angle is measured anticlockwise from the vertical axis, see point (1) in figure 3. Notice that figure 4 is non dimensional, facilitating to understand the percentage variation of the different parameters versus the original ones. It is interesting to realize that as inclination angle increases, the output frequency tends to decrease, while the amplitude tends to increase. In percentage, it can be said that for an angle variation of ±70%, output frequency changes by approximately 30% while maximum amplitude variation hardly reaches 20%. Notice as well that this behaviour is not linear, the authors believe this may be due to the fluid nonlinear behaviour specially inside the mixing chamber. At this point it is important to recall that, for each case, output frequency and amplitude were measured based on the temporal output mass flow signal, obtained via integrating the fluid flow velocity across the fluidic amplifier output upper surface.

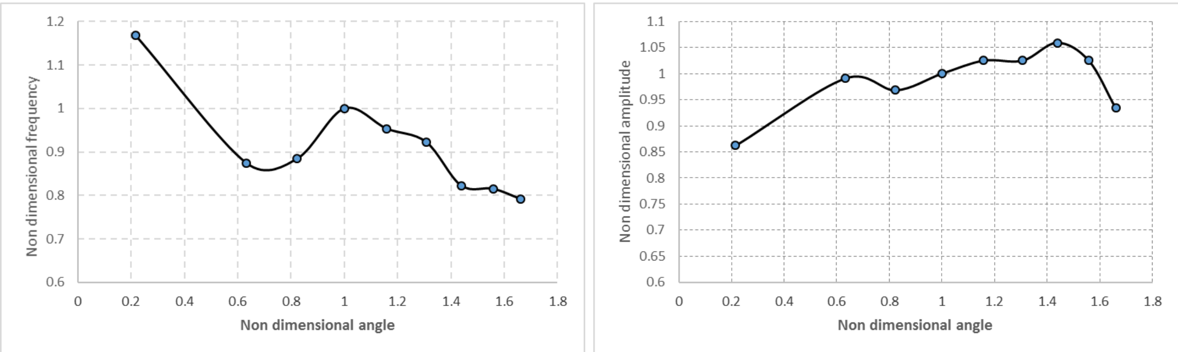
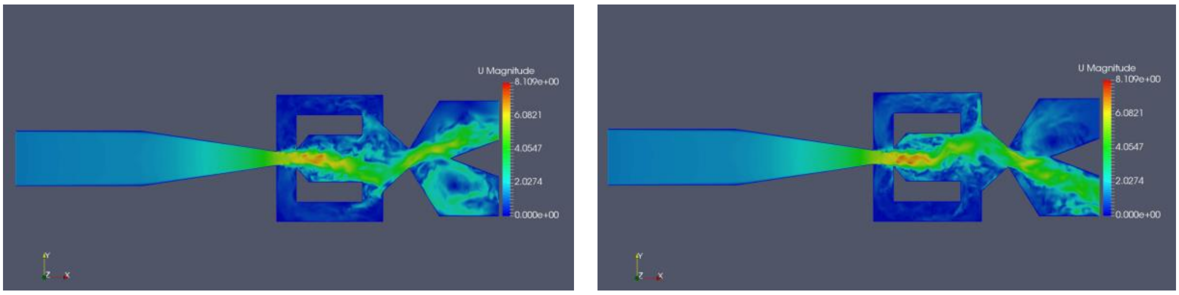
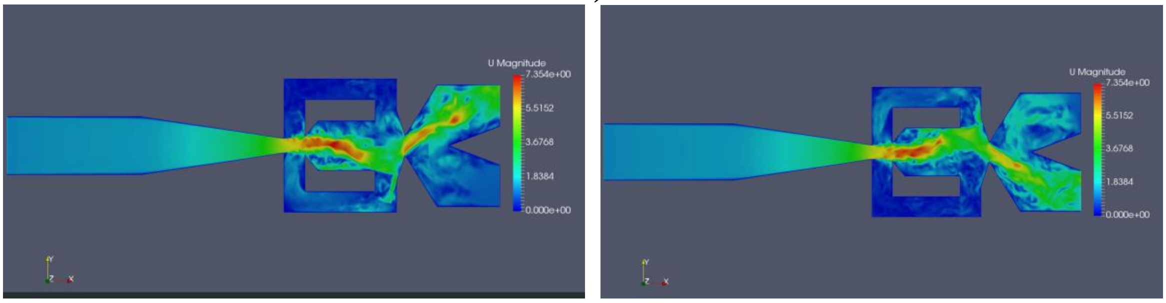


Fig 4 Non dimensional frequency and amplitude variation as a function of the mixing chamber outlet angle.



a)



b)

Fig 5 Fluidic amplifier internal flow visualization for a) maximum angle studied, b) minimum angle.

In order to better understand how the fluid evolves inside the fluidic actuator, in figure 5 is presented the fluid flow for the maximum and minimum angles studied. From this figure it is realized that as the mixing chamber output inclination angle increases, the jet oscillation inside the mixing chamber decreases, the flow is directed towards the output wedge and flows quite parallel to it, as a result, the fluid uses a small section of the outlet channel to leave the fluidic amplifier, allowing external fluid to enter the (FA) and generating a large alternative vortex on both sides of the wedge. Notice that for small angles, the jet undertakes a wide oscillation inside the mixing chamber and the fluid is mostly directed to the centre of the fluidic amplifier outlet channel section, external fluid finds now more difficult to enter, and the vortex generated alternatively on both lateral sides of the wedge, is for the present case, much smaller.

4.2 Output frequency and amplitude variation when modifying mixing chamber outlet width.

Another important parameter which it is expected to modify the amplifier internal flow and its output performance is the mixing chamber outlet width, represented as point (2) in figure 3. For the present study this parameter was modified ±40% versus its initial dimension, the outcome of the simulation is presented figures 6 and 7. Figure 6 introduces the FA exit frequency and amplitude variation as a function of the non-dimensional mixing chamber outlet width. Reynolds was maintained constant at 8711. Clearly the frequency increases with the outlet width increase, while the amplitude decreases. Notice that the opposite was happening in the previous sub-section when increasing the mixing chamber outlet angle. In reality the phenomena linking frequency and amplitude in both cases is the same. When decreasing the outlet width or increasing the outlet angle, the oscillation amplitude inside the mixing chamber tends to decrease, and in both cases the flow leaves the fluidic amplifier, quite parallel to the output wedge. In reality, for very small mixing chamber outlet widths, the fluid main stream is sliding over the external wedge walls, see figure 7, therefore using a very small fluidic amplifier outlet section to leave the amplifier, as a result very big alternative vortices are generated at both sides of the wedge. In both cases, the outgoing and incoming fluid flow velocity at the fluidic amplifier exit is maximum and so it is the mass flow amplitude associated. Regarding the frequency, as the incoming mass flow is time independent, to fulfil with the continuity equation, whenever exiting mass flow amplitude increases, the time required for a given fluid mass to leave the actuator has to reduce and the rest of the time to complete one cycle is used to allow external fluid into the actuator. Providing the amplitude associated to the outgoing mass flow is smaller, the fluid will require several cycles to transfer the incoming mass flow to the outlet, in each cycle, fluid will mostly leave the FA and some small flow will get into it. In both cases, for small and big amplitudes, after a given period of time, the fluid mass transferred from the inlet to the outlet must be the same. This explains why in figures 4 and 6, big amplitudes have associated small frequencies and vice versa.

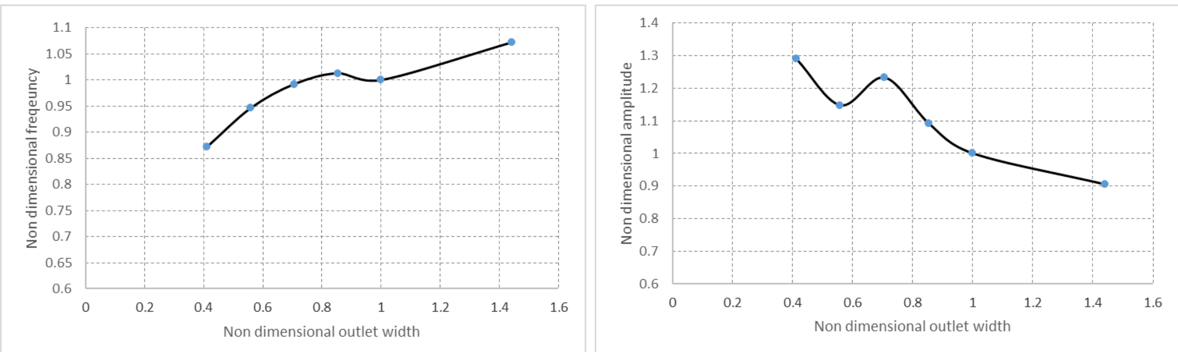


Fig 6 Non dimensional frequency and amplitude variation as a function of the mixing chamber outlet width.

A key difference when comparing the cases presented in sub-sections 4.1 and 4.2, is that, when the output width decreases the mixing chamber is being pressurized, mixing chamber stiffness increases, on the other hand, as outlet angle increases, the mixing chamber oscillating flow, is being directed towards the fluidic amplifier outlet, the maximum stagnation pressure point at the mixing chamber outlet reduces and so reduces the overall pressure inside the mixing chamber, mixing chamber stiffness decreases

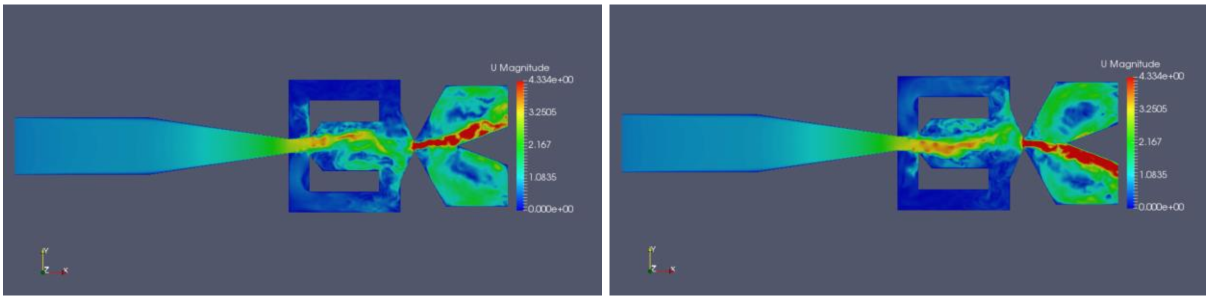


Fig 7 Fluidic amplifier internal flow visualization for the smallest outlet width studied.

Figure 7 is presenting the internal flow visualization for the minimum outlet width studied. It is observed that fluid has a small oscillation inside the mixing chamber and the fluid leaves the fluidic actuator sliding along the outlet wedge, therefore leaving a big part of the exit area unfilled and allowing external fluid entering the actuator.

4.3 Output frequency and amplitude variation when modifying mixing chamber inlet width.

In the present case, see fig 8, the first thing to be noticed is that whenever the inlet width falls below a minimum or is higher than a maximum, the actual fluidic amplifier is not producing any outgoing frequency, simply the flow crosses the amplifier as a jet. What it is also interesting to realize is that it exists a particular width at which the outgoing flow frequency falls to a minimum, a small increase or decrease from this width causes a slight increase of actuator frequency.
The explanation why there is no flow oscillation when the actuator inlet width overcomes a minimum value, is based on the fact that, at these small actuator inlet widths, the incoming jet partially impinges onto the left hand side mixing chamber walls, generating a pressure increase and forcing part of the fluid to move downstream along both feedback channels. For these particular cases, the feedback channels simply do not act as expected, which is, letting the flow and pressure waves to move from downstream to upstream, no feedback is allowed.
On the other hand, when the mixing chamber inlet width overcomes a maximum value, it appears a gap between the incoming jet and the inlet width borders, this small gap is enough to prevent the pressure increase at the feedback channel outlet, then it allows the fluid coming up from the feedback channel inlet, to escape though this gap, as a result, the momentum applied to the incoming jet lateral sides is drastically reduced, the jet cannot flip over. A second effect which is also tending to prevent the jet from oscillating inside the mixing chamber is, the low intensity static vortices generated alternatively on both sides of the mixing chamber. In other words, the vortices and the low pressure associated to the Coanda effect, appearing alternatively on both sides of the jet inside the mixing chamber, are too weak or simply do not appear.

In figure 8 it is also shown the effect on the oscillation amplitude, when modifying the inlet width. For inlet widths exceeding a limit in any direction, whether too big or too small, the flow stops oscillating and therefore the amplitude decays to zero. For the intermediate values it is seen that the amplitude is not much affected by the inlet width. It is also seen that the tendency is opposed to that of the frequency, and the highest amplitudes are found at the points where the frequencies are slightly smaller. Small frequencies are linked with high oscillation amplitudes and vice versa.
Figure 9 introduces the fluidic actuator internal flow visualization, for the cases of minimum and maximum inlet width, notice that in both cases no oscillation appears. Although not presented here, when evaluating one of the smallest diameters, it was obtained that the flow was just flowing through one of the (FA) exits, and the flow remained steady in this position.

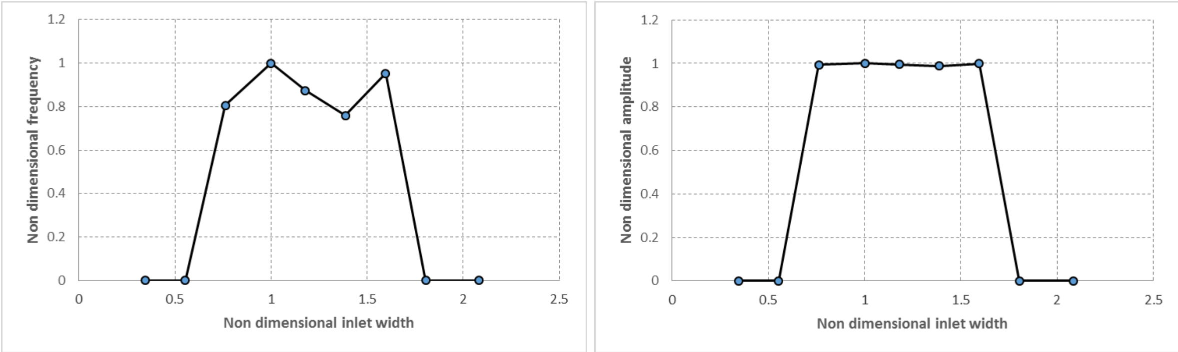
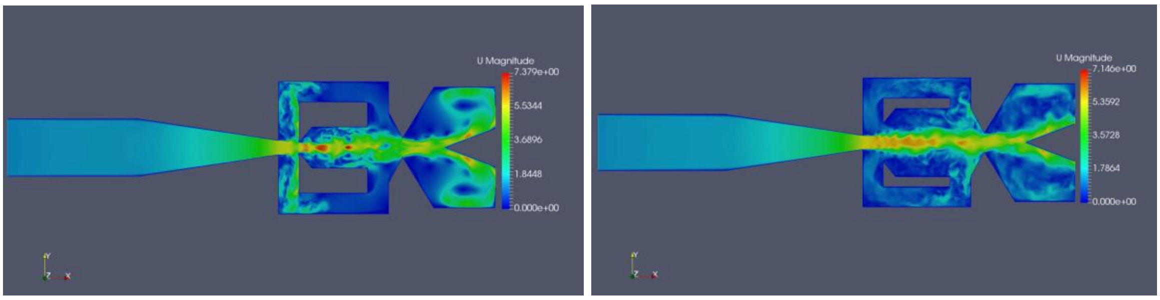


Fig 8 Non dimensional frequency and amplitude variation as a function of the mixing chamber inlet width.



a) b)

Fig 9 Fluidic amplifier internal flow visualization for a) minimum inlet width, b) maximum inlet width.

4.4 Output frequency variation when modifying fluidic actuator dimensions.

The latest test undertaken in the present study, relates the (FA) frequency with its dimensions, when maintaining constant the incoming fluid Reynolds number. For the present case the simulation was performed in 2D being the Reynolds number 8711. Five different fluidic actuator dimensions were evaluated, the original one, and another four which scale was, 0.1, 0.5, 5 and 10 times the original one. The output frequency obtained is presented in figure 10. The left hand side of figure 10, presents the oscillation pressure as a function of the scale measured in percentage, the right hand side introduces the same graph using double logarithmic axes, as expected, a linear relationship appears, indicating a common physical phenomenon.

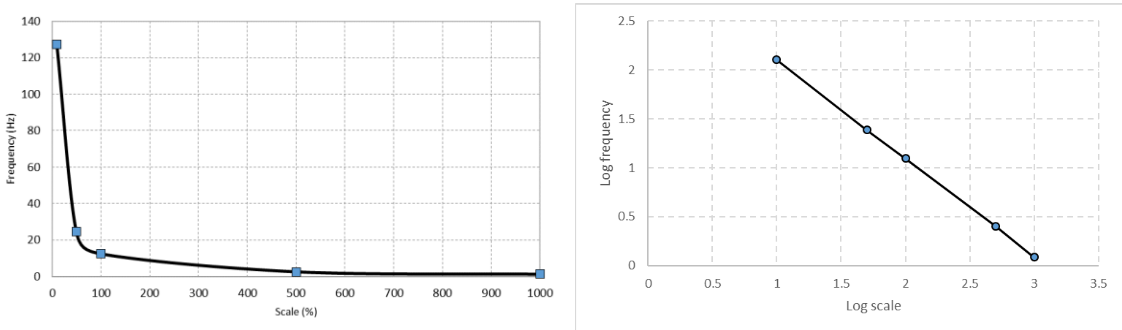


Fig 10 Variation of frequency when the fluidic actuator is scaled while maintaining constant the inlet Reynolds number.

5 Conclusion and Future Work

From the present study it is obtained that, when modifying the fluidic amplifier internal dimensions, while maintaining a constant input Reynolds, the output frequency and amplitude suffer appreciable modifications. It is noticed that two of the mixing chamber internal dimensions studied, the output angle and width, play an important role regarding the fluidic amplifier dynamic characteristics, modifying about ±10% the frequency and amplitude versus the original case. Regarding the scale associated to the actuator, it is observed that, when maintaining constant the input Reynolds number, as the scale increases, frequency sharply decreases. The relation is linear when using double logarithmic axes. The future work the authors are having in mind is to evaluate the effect of different Reynolds numbers on the actuator frequency and amplitude. The same parameters need to be evaluated when modifying the feedback channel dimensions, considering as well the compressibility effects.

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