A STUDY OF FLUIDIC OSCILLATORS AS AN ALTERNATIVE
PULSED VORTEX GENERATING JET ACTUATOR
FOR FLOW SEPARATION CONTROL

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Nomenclature

\( D_0 \)  
Output orifice diameter

\( b_s \)  
Power nozzle width

\( b_c \)  
Control nozzle width

\( B \)  
Offset distance of an attachment wall from the power nozzle

\( L \)  
Feedback loop length

\( h \)  
Power nozzle height

\( D_h \)  
Hydraulic diameter

\( AR \)  
Aspect ratio of power nozzle

\( A_o \)  
Cross sectional area of Output orifice

\( A_{ref} \)  
Cross sectional area of the fluidic element output aperture conduit at the location of the output port

\( U_S \)  
Power jet mean velocity at the exit plane of the power jet nozzle

\( \overline{u} \)  
Time-averaged jet velocity

\( \overline{u}_m \)  
Time-averaged jet mean velocity

\( \Delta u \)  
Difference between the high state velocity and the low state velocity of the oscillating jet

\(<u> \)  
Phase-averaged velocity

\( f \)  
Frequency of jet oscillation

\( f_{sat} \)  
Maximum attainable frequency

\( P \)  
Pressure

\( \Delta P \)  
Differential pressure

\( T \)  
Period of jet oscillation

\( x_R \)  
Power jet attachment distance

\( q_w \)  
Mass flow rate

\( q_V \)  
Volume flow rate
M  Local Mach number
GF  Flow gain
nF  Fanout ratio
Tu  Turbulence intensity
St  Strouhal number based on $U_S$ and $L$
Re$_{b_S}$  Reynolds number based on $b_S$
Re$_{D_h}$  Reynolds number based on $D_h$
Re$_{D_o}$  Reynolds number based on $D_o$
Re$_c$  Critical Reynolds number based on $D_h$
Romod  Modified Roshko number

Greeks
\( \xi \)  Ratio of feedback loop length to power nozzle width
\( \rho \)  Density of fluid
\( \mu \)  Dynamic viscosity of fluid
\( \nu \)  Kinematic viscosity of fluid
\( \eta_{\text{In}} \)  Ratio of mass flow rate to supply flow rate
\( \tau_p \)  Propagation time
\( \tau_S \)  Switching time
\( \tau_n \)  Fundamental switching time
\( \tau_L \)  Switching delay time
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Declaration

No portion of the work referred to in this thesis has been submitted in support of an application for another degree or qualification of this or any other university or other institution of learning
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Abstract

The current study experimentally examines a control loaded bistable fluidic oscillator and an array of bistable fluidic amplifiers driven by the control loaded fluidic oscillator to validate their potential as an alternative flow control actuator. These fluidic devices are considered to be able to generate a periodically oscillating jet without a necessity of any mechanical moving parts and their prototype designs are provided by Professor Vaclav Tesar from Academy of Sciences of the Czech Republic.

The output jet characteristics of the control loaded fluidic oscillator has been experimentally evaluated by means of hotwire measurements with single wire probes oriented normal to the plane of output orifices for supply flow rate range of 0.7g/s ~ 6g/s with a feedback loop length as a parameter which varied from 0.3m ~ 1.78m. The experiment have successfully demonstrated that the prototype fluidic oscillator can generate periodically oscillating jets from its conjugating pair of orifices in an alternating manner with a 50% pulse duty cycle, which its amplitude varies proportionally to the supply flow rate. The period of the pulsation found to vary linearly with a supply flow rate up to a power nozzle Reynolds number of 3.0X10^4 where the frequency saturation initiates giving a constant Strouhal number of ~0.7 with further increase in supply flow rate. The experimental results demonstrate an existence of a strong correlation between the propagation time, \( \tau_p \), and the switching time, \( \tau_s \), in determining the saturation frequency, \( f_{sat} \), for a chosen feedback loop length. Furthermore the current study demonstrates a simple way of possibly approximating the natural switching time of a control loaded fluidic oscillator without making an internal flow measurements, which may require further verification though, and found it to be approximately 1ms for the current test model. In general, the frequency response of the oscillating output jet reduced linearly with increasing feedback loop length when running at a relatively low values of \( Re_{D_h} \), while changing inverse proportionally with \( L \) at high \( Re_{D_h} \) values.

The experimental study of an array of bistable fluidic amplifiers verified that there exists a load mismatching due to a concentric reduction in the output orifice diameter for its intended function of delaying flow separation over an aerofoil, which caused attenuation in the output pulse characteristic. Although this change in the output characteristic may not be crucial when it comes down to an actual flow control application or can even be advantageous in certain cases the author attempted to remove the disparity in the output jet characteristic with a normal operation condition and suggests a design modification as described in chapter 5. As a by-product of this study it was found that the ratio of the output orifice diameter to the cross sectional area of the exit aperture conduit of a device should be greater than \(-0.5\) to avoid attenuated output jet characteristic and as a supporting evidence it was found that amplifier with a smaller height, i.e., lower power nozzle aspect ratio, can accommodate a smaller orifice diameter without disturbed output jet characteristic.
1 Introduction

1.1 The need for flow separation control
Flow separation on an aircraft body and wing sections, such as what is shown in Figure 1.1, can cause a significantly increased pressure drag and a reduced lift hence a great loss in performance. In an extreme case, this can result in a loss of control on the aircraft. For these reasons, our ability to control flow separation is of immense technological interest in the realm of aerospace engineering.

![Figure 1.1 Separated flow over an aerofoil at an angle from Schlichting and Gersten (2000). Original picture taken by Prandtl and Tietjens (1931).](image)

Flow separation control can provide potential benefits such as improved lift/stall characteristics, which lead to a reduced landing speed and a greatly enhanced manoeuvrability. It also allows the high-lift system and the control surfaces to be simplified leading to a reduced take-off gross weight hence a reduced fuel requirement or an increased endurance. For an aircraft optimised for supersonic missions, this can improve their poor subsonic performance. As a summary, flow control enables an aircraft designer to work with a broader flight envelope and it can offer a reduction in the direct operational cost to the aircraft transport industry.

1.2 Methods for flow separation control
Owing to the extensive research work undertaken during mid 1900s various types of flow separation control devices and methods have been developed (Gad-el-Hak 2000). The classic approaches include passive/active tangential blowing which directly increases the momentum in the near-wall flow (steady VGJs), active wall suction which removes the slow moving low energy flow near the wall and vane vortex generators (VGs) which energise the boundary layer flow by enhancing the
mixing with the high energy outer flow. Some of these devices are now routinely employed on aircrafts (Joslin 1998). The passive and active classification used in this context categorise flow control methods depending on the energy expenditure requirements in their actuation. Figure 1.2 provides a reader with an idea of the level of modification of the flow that can be achieved by the application of a flow separation control with a classic active suction method.

![Figure 1.2 Separation control of flow in a diffuser by suction](image)

Figure 1.2 Separation control of flow in a diffuser by suction. a) Separation at both walls; a') Suction applied only at the upper wall; a'') Suction at both wall (Schlichting and Gersten (2000)). Original picture taken by Prandtl and Tietjens (1931).

Since the demonstration of the capability in delaying the flow separation of Pulsed Vortex Generating Jets (Pulsed VGJs) in late 1950’s (Wallis and Stuart 1958; Pearson 1961), Pulsed VGJs have been regarded as a promising alternative for accomplishing the control. Generation of pulsed VGJs are normally achieved by using solenoid/siren valves or cylindrical pump. Also, advanced actuators incorporating a piezoelectric diaphragm or plasma electrode are frequently used. Extensive efforts on verifying their effectiveness and feasibility have been made (Zhang and Sheng 1987; Katz et al 1989; Seifert et al 1993; McManus et al 1994, 1996; Amitay et al 1998; Glezer et al 2005). Compared to its predecessors such as the vane vortex generators (VGs) and the steady VGJs, Pulsed VGJs actuators exhibit an improved performance in suppressing flow separation. This improvement arises from the fact that they can generate convective vortical structures which penetrate further into the boundary layer than those of steady VGs resulting an improved mixing with an outer layer. Also as a result of an impulsively started unsteady pulsation, a much greater mixing can be achieved compared to that of steady VGJs due to the generation of large scale turbulent vortex rings, which are much more effective in mixing, in addition to the customary streamwise vortices
(Johari and McManus 1997). Furthermore, Unlike the VGs, Pulsed VGJs produce zero parasite drag and they also require much less air supply than the steady VGJs.

In more recent years, the research emphasis has been shifted from optimising the classic methods to developing more efficient Micro-Electro Mechanical System (MEMS) based actuation methods (Warsop et al, 2007). This is a result of the increased need for the separation control devices to be controlled on demand so as to deliver an optimised performance at all flight conditions. The advances in the MEMS technology and the even increasing demand for energy saving have led to a global interest in searching for more effective and robust actuators for flow control.

1.3 Fluidic oscillators

The most ingenious feature of fluidic oscillators would be their capability of generating periodically oscillating pulsed jets without a necessity of any mechanical moving parts. A fundamental principle of their operation is based on the well known Coanda effect and this will be explained in details in Chapter 2.

A fluidic oscillator usually consists of one supply port, two output ports and two control ports along with their feedback loop. They can be generally categorised into two classic designs as shown in Figure 1.3, in which the oscillator in Figure 1.3 a) is named as the control loaded fluidic oscillator and that in Figure 1.3 b) as the vent-fed oscillator.

![Fluidic Oscillators Diagram](image)

a) Control loaded fluidic oscillator  b) Vent-fed fluidic oscillator

S – supply port; C – control port; O – output port

**Figure 1.3 Classic designs of fluidic oscillator.** (Morris (1973)).
Both types of the above fluidic oscillators share a common principle at the very beginning stage of their operation which can be summarised as following. As the power jet leaves the power nozzle (throat), owing to the Coanda effect it will choose to attach to one side of the attachment walls. For a symmetric bistable oscillator, the jet can attach equally well on to one of the both sides of the attachment walls, hence the jet attachment would solely governed by a naturally introduced momentary disturbances. However, in an asymmetric monostable oscillator the power jet will always attach to the side where the entrainment is more restricted. This attached jet will then initiate the habitual oscillation mechanism in the fluidic oscillator. Figure 1.34 shows schematic drawings of a symmetric bistable and symmetric oscillators.

![Schematic drawings of a symmetric bistable and asymmetric monostable oscillator](image)

Figure 1.4 Schematic drawing showing the geometrical difference of a symmetric bistable and asymmetric monostable oscillator.

In a control loaded fluidic oscillator, the separation bubble formed on the attached side wall further increases the pressure difference hence a rarefaction wave (expansion) is generated on that side of the control port while a pressure wave (compression) grows on the opposite side. The continuous generation and interaction of these waves through the time delay feedback loop is the fundamental principle of operation of these types of oscillators. Due to this switching characteristic, they are more appropriate to be regarded as a pressure driven devices.

The switching mechanism in a vent-fed fluidic oscillator partially incorporates pressure change but the oscillation is mainly driven by the momentum injection through the control ports (momentum driven devices). In this type of oscillator, a portion of the fluid captured from the attached flow will be fed back to the control port and deflects the jet to the opposite side. The same process initiates on the newly attached side and a replication of this cycle generates the periodic oscillation.
The period of the oscillation in both types of the oscillators are governed by the pneumatic resistance and capacitance in the feedback loop as well as the interaction region geometry.

As these devices can generate pulsing jets in an absence of a mechanical moving part, they offer many advantages compared to other conventional PVGJs actuators. This would include greater mechanical and environmental condition robustness. Furthermore, they usually consist of simple interconnections hence require less compartment space and maintenance work. As a result, the cost of fluidic systems promises to be low in long term. Also, with microfabrication a frequency response up to a range of several kilohertz is attainable with these fluidic devices (Morris 1973; Raber and Shinn 1964), which is commonly required by a PVGJs based separation control device. An example of a microfabricated bistable vent-fed fluidic oscillator is shown in Figure 1.5.

![Figure 1.5 Microfabricated bistable vent-fed fluidic oscillator (Simões (2004))](image)

In recent years, there has been a lot of published work suggesting the use of fluidic devices for flow control applications. These research efforts range from studying the fundamental performance characteristics of innovatively designed fluidic devices (Tesar et al 2006; Yang et al 2007; Furlan et al 2006) to validation of a device in laboratory scale internal/external flow control applications (Culley et al 2003; Raman and Raghu 2004; Cerretelli et al 2009, 2010). Also, notable numbers of publications have been made on studying the feasibility of incorporating other known flow control actuators, such as piezoelectric transducer or plasma actuator, into a fluidic amplifier in order to gain control authority of the jet switching independent of supply flow level (Gregory et al 2005, 2007, 2009). However, most of research efforts are focused on fluidic device employing the momentum interaction as a principle of jet switching. Thereby the work shown in this thesis represents an early attempt of evaluation of a pressure driven fluidic oscillator as an alternative PVGJs actuator.

Fluidic devices hold many potential benefits making them an attractive PVGJs actuator mainly due to their geometrical simplicity and superior physical durability.
Nevertheless, our knowledge as how to design the fluidic devices which satisfy the requirement on jet velocity, pulsing frequency and optimisation of actuator geometry for flow control purposes is still lacking.

1.4 The objectives of this project

As the result of a collaborative research with Czech Academy of Sciences, the prototype of a bistable control loaded fluidic oscillator and the design of a fluidic circuit consists of an array of bistable fluidic amplifiers with a potential application for delaying flow separation on a 2D aerofoil section have become available.

The objectives of this work are;

1. To examine the jet characteristics of a single fluidic oscillator over a range of supply flow rates and control loop length;
2. To examine the jet characteristics of the fluidic circuit at a range of supply flow rates and actuation frequencies;
3. To propose the modification of the fluidic circuit so that it can deliver the required jet characteristics when it is embedded in an aerofoil section to be used in a flow separation control experiment.

All the experiments were conducted at the Goldstein Aeronautical Lab. The jet characteristics were measured using a single-normal hotwire probe.

1.5 The layout of this report

Chapter 2 describes the fundamentals of the boundary layer concept including the definition, the nature of turbulent boundary layers and the separation mechanism. A literature review of flow control and fluidics is also included describing their classification, principles of operation and previous work that has been undertaken as to demonstrate their capability.

Chapter 3 describes the experimental models and experiment methods used to accomplish the aforementioned objectives.

Chapter 4 discusses the experimental findings attained from the study of the single prototype fluidic oscillator. This includes its steady state performance parameters, oscillation characteristics and exit jet responses. Chapter 5 presents the experimental results gained from the study of an array of bistable fluidic
amplifiers driven by the prototype fluidic oscillator. Most of the section has been devoted to show their oscillation and jet response to a driving actuator.

Chapter 6 shows the author's perspective of the feasibility of the existing control loaded bistable fluidic oscillator as a separation control device based on his understanding gained through this project. Suggestions for future work are also given.
2 Literature review

2.1 Boundary layer and its separation

As the fluid moves over a solid surface, the fluid molecules in contact with the surface adhere to the surface owing to fluid viscosity, resulting in the no-slip condition. On the other hand, the effect of viscosity becomes negligible further away from the surface, allowing the fluid to follow the speed of the freestream velocity. Thereby a region exists in the vicinity of the surface where a steep velocity change occurs. This region is commonly known as a boundary layer or wall-bounded shear layer.

A detachment of the boundary layer from the surface can occur at some point (or line) along the wall and the phenomenon is known as a flow separation. Prandtl (1904) stated that the separation occurs owing to the development of an adverse pressure gradient along the surface in the direction of flow, which is present in the freestream flow. The presence of the adverse pressure gradient will resist the motion of fluid and if the fluid particles within the boundary layer no longer possess enough energy to overcome the resistance, the motion will be halted. Further downstream the flow starts to deflect laterally and eventually recirculation is generated causing the departure of the boundary layer from the surface and resulting in greatly increased pressure and form drag. The flow visualisation showing development at the time of boundary layer separation is shown Figure 2.1.

At this point, it would be worth reviewing the total drag components of an aerodynamic body moving in a flow and their origin to aid understanding of their interrelation with the boundary layer. The total drag components of an aerodynamic body can be broadly divided into a profile drag and a lift-induced drag. When a fluid flow moves over a finite length lifting surface/body the resulting pressure difference between the suction surface and the pressure surface creates a vortical structure at its tip known as a wing tip vortex and this is the major source of the lift-induced drag. The profile drag is usually subdivided further into a skin friction drag, a form drag and a pressure drag. The skin friction drag is governed by the fluid viscosity and a surface roughness of the body, hence determining the magnitude of the shear stress that develops between the body surface and fluid particles moving over its vicinity. On the other hand, as a boundary layer develops over a body moving in a
fluid flow this causes a physical profile change of the body shape making it blunter with a corresponding increase in drag. This additional drag owing to a change in an effective body shape is called a form drag. The origin of the pressure drag can be easily understood by visualising a fluid flow over a blunt body or an aerodynamic body at an angle. A boundary layer under this circumstance will usually separate from the body surface creating a low pressure region at downstream, which results in a considerable pressure difference between the forward and the aft of the body. This causes the body to experience a pressure force in the opposite direction to its motion and this force is known as a pressure drag.

Figure 2.1 Flow visualisation showing a development in time of the separation at the aft of an upper quadrature of a symmetric cylinder. a) The flow recirculation has just begun at the trailing edge shortly after the initiation of the boundary layer separation; b) The boundary layer has thickened and the start of the reversed motion moved forward with growing in its size; c) A large vortex has formed from the backflow; d) The vortex grows in its size; e) The vortex separates from the body and moves downstream changing the flow portrait at the rear of the body. Schlichting and Gersten (2000).

2.2 Boundary layer transition

Boundary layer can be either laminar or turbulent and a laminar boundary layer will always naturally develop into a turbulent boundary layer unless otherwise controlled. For a flat plate with a sharp leading edge at zero incidence, the laminar-turbulent transition in a normal air stream takes place at a point where the Reynolds number based on a distance from the leading edge exceeds \( \text{Re}_{x \text{ crit}} = 3.5 \times 10^5 \) to \( 10^6 \) (Schlichting and Gesten 2000). There are many other factors, apart from the Reynolds number, governing the onset of the turbulent boundary layer such as the pressure distribution and turbulence intensity of the outer flow and surface roughness (Schlichting and Gesten 2000).
The Tollmien-Schlichting (T-S) instability transition mechanism is proven by free flight experiment that this dominates in most of subsonic boundary layers, except for swept wing where cross-flow instability plays an important role (Schlichting and Gesten 2000). For two-dimensional zero pressure gradient boundary layer, the process can be summarised as follows; above the indifference Reynolds number (Re_{ind}) the laminar boundary layer becomes sensitive to small disturbances, leading to amplification of unstable two-dimensional linear T-S waves (primary instability). Once these primary T-S waves exceed a threshold value of 1% of the freestream velocity (King and Breuer 2002), they slowly become three-dimensional and form hairpin vortices (non-linear secondary instability). These then interact together and are intensified as they are stretched to form the turbulent spots. The turbulent spots grow as they propagate downstream and they eventually merge, leading to a fully turbulent flow. The sketch of the process is given in Figure 2.2 and some of the pictures of structures developed in the boundary layer are shown in Figure 2.3.

Tollemien (1929) and Schlichting (1933) applied the Orr-Summerfield equation to analysed the linear primary instability with the laminar Blasius boundary layer and estimated theoretical Re_{ind} of 520 (Schlichting and Gesten 2000) based on the boundary layer thickness using Eq. 2-1

$$\delta = 5.0 \sqrt{\nu x / U_\infty}$$  \hspace{1cm} Eq. 2-1

Most of their theoretical estimation of the essential points agreed well with experimental data (Schlichting and Gesten 2000) of Dryden (1946, 1948) and the existence of T-S waves in natural transition has also been experimentally confirmed by Arnal et al (1977).

There is one other important transition mechanism, first verified by Morkovin (1988), in the flow control point of view known as the bypassed transition. In this scheme the aforementioned T-S instability is bypassed and the subcritical perturbation introduced in the initial condition, such as turbulence intensity and/or surface roughness, etc., directly amplifies the secondary instability or even turbulent spots, leading to advancement in transition. An example of bypass transition is shown in Figure 2.4 where transition is triggered by a tripwire placed around a sphere.
1) Stable laminar flow
2) Growth of unstable 2D T-S waves
3) T-S waves turn 3D and form hairpin vortices
4) Hairpin vortices interact and intensify by undergoing vortex stretching
5) Cascading into smaller eddies takes place and forms turbulent spot
6) Spots propagate and grow and merges forming the fully turbulent flow

Figure 2.2 Laminar to turbulent transition process in a boundary layer
a) Sketch of the process; b) View of transition
Schlichting and Gerten (2000)
Original picture after Brown, F. N. M. (1957)
Figure 2.3 Hairpin vortices and turbulent spot
a) Hairpin vortices created by boundary layer trip wire;
a’) cross-sectional view of hairpin vortices; b) A turbulent spot after Lim, T. T. http://serve.me.nus.edu.sg/limtt

Figure 2.4 Flow past a sphere. a) Subcritical flow in subcritical Re regime; b) Supercritical flow in subcritical Re regime with a trip wire. Schlichting and Gesten (2000). Original Picture after Wieselsberger, C. (1914)
2.3 Turbulent boundary layer

Turbulence in a flow can be characterised by continuous generation of the large spatially coherent structures (eddies), which have size comparable to the boundary thickness in the case of a turbulent boundary layer. The turbulence kinetic energy in these is then cascaded into various sizes of smaller-scale eddies as they are convected downstream and stretched and diffused until the eddy size scale reaches the Kolmogorov length scale, where viscous dissipation starts to dominate and convert the energy into heat energy. Note that the breakdown of large-scale eddies can be realised in a hierarchical manner and/or by simultaneously evolving multiple eddies with various sizes and in certain cases, such as in 2D flow, energy can be transported from smaller eddies to larger eddies (Oertel et al 2004).

These wide spectra of eddies effectively impose irregular fluctuations on the time-mean flow properties, so the velocity, pressure and temperature measured at a fixed point in space do not remain constant in time. As a result of these fluctuations macroscopic and microscopic mixing takes place over several layers of flow while only a microscopic scale mixing is realised between the adjacent layers in a laminar flow, hence greatly enhancing the mass, momentum and energy exchange as schematically represented in Figure 2.5.

![Figure 2.5 Schematic representation of laminar and turbulent flow](http://ocw.mit.edu/NR/rdonlyres/Civil-and-Environmental-Engineering/1-061Fall-2004/FE208D33-7F5B-40DA-9E43-92DE1B4264AE/0/turbulent.pdf)

As a result of enhanced mixing, a steeper velocity gradient as schematically represented in Figure 2.6 develops in a turbulent boundary layer. This effectively indicates a higher shear stress at the wall, producing skin friction drag as much as one order of magnitude higher than that of the laminar boundary layer, as can be seen in Figure 2.7. Also, an increased boundary layer thickness would be another
characteristic feature of the turbulent boundary layer. The increased shear stress also leads to a greater tendency to remain attached to the surface under the influence of an adverse pressure gradient. Consequently the flow separation can be delayed and/or prevented by provoking a turbulent boundary layer, as shown in Figure 2.8.

Thereby the turbulent boundary layer is favourable, if there is a danger of increasing pressure region, because the drag increase owing to the flow separation (pressure drag) is far greater than the increase in skin friction drag by promoting the turbulent boundary layer. The associated advantages of postponing separation would include enhanced lift and/or stall characteristic, improved flight safety from buffeting, greater manoeuvrability, better control surface effectiveness, simplified and/or safer use of high lift system, and therefore lower take-off gross weight.

![Figure 2.6 Schematic representation of velocity profile in laminar and turbulent boundary layer](image1)

![Figure 2.7 Skin friction coefficient for a smooth flat plate at zero incidence](image2)

Gad-el-Hak (2000)
2.4 Flow control

2.4.1 Definition and its classification

After the publication by Ludwig Prandtl in 1904 of seminal research which presented the concept of the boundary layer and the possibility of its forced manipulation by delaying the flow separation by the use of slit suction on a cylinder, the study of new technologies for flow control took off. The majority of researchers in the realm of aerospace industry believed that this unique technology could realise development of an aircraft with improved overall performance beyond the level achievable by traditional geometrical optimisation of the body and named this field of study ‘Flow Control’, which can be defined as follows:

The controlled modification of the local boundary layer flow conditions of wall-bound and/or free shear flow fields in order to achieve a desired global modification in the dynamics of their flow characteristics.

The desired global changes usually include suppressed/enhanced turbulence, accelerated/delayed transition and prevented/provoked separation. The potential end results of these consist of reduced drag, enhanced lift, suppressed flow-induced noise, improved manoeuvrability, augmented mass/momentum/energy mixing and
removal/simplification of control surfaces. One should, however, recognise the interrelation between one control goal and another, as in Figure 2.9 in determining the net performance gains.

For the past one century many different flow control strategies have been developed and demonstrated their capabilities ranging from as simple as controlled surface roughness to as advanced as feedback closed control loop MEMS (Micro-Electro-Mechanical Systems) based actuators. These various flow control devices are commonly categorised as shown in Figure 2.10

A technique that requires any form of energy expenditure in driving the actuator would be classified as an active method while the term passive is used to describe those which do not. Active control methods are subdivided further depending on the capability of producing a dynamic response to changes of flow state, into predetermined and reactive methods. The reactive system can be distinguished by the use of sensors to make pre-specified flow measurements to adjust the control input to vary accordingly, and the system can either be open or closed loop.

2.4.2 Classic methods and principles of operation

In this section some of the historical developments of the classical and recent flow control methods of the last century and their principles of operations are discussed.

2.4.2.1 Stability modifiers

Stability modifiers are those of flow control actuators/methods intended to suppress the initiation of the boundary layer transition to benefit from the customary lower skin friction in laminar boundary layer by controlling the primary instability growth in the boundary layer. Although the ultimate goal of this method is to reduce the skin friction, separation delay is another beneficial by-product. This scheme is generally applied to a flow with a Reynolds number lower than $3 \times 10^7$ and various techniques are available, such as moving wall, wall heating/cooling, aerodynamic shaping, wave cancellation, etc.
Figure 2.9 Interrelation between flow control goals
Gad-el-Hak (2000)

Figure 2.10 Classification of flow control strategies
Gad-el-Hak (2000)
For the completeness of the report the suction method (Laminar Flow Control), which has been extensively studied and is still receiving considerable attention from the industries, is reviewed in terms of its development and principles of operation.

The laminar flow control (LFC) by means of active suction has been studied extensively and yielded many promising flight test results. Its ultimate goal was to reduce the skin friction drag during level flight in cruise condition, given that this drag accounts for 50% of the total drag of conventional subsonic transport aircraft (Joslin 1998), mostly owing to the presence of a turbulent boundary layer (TBL), which has 90% higher skin friction coefficient than that of a laminar boundary layer (LBL).

The principle of the method is postponement of the laminar-turbulent transition over the surfaces such as the wings, tailplane, fuselage nose and engine nacelle by removing the low energy near wall flow in the boundary layer. Suction can be applied through permeable surfaces (porous or slots or perforated) or by tangential slit suction.

When a perforated surface is used, this will cause a thinning effect on the existing LBL, hence giving a more favourable velocity profile curvature leading to greater shear stress, but much smaller than the ones in TBL, that can resist increasing pressure more effectively. This method, however, imposes one extra drag source called a sink drag which results from the flow withdrawn just downstream of the point of application. In the case of slit tangential suction, existing retarded flow will be effectively removed, followed by the development of a newly-initiated LBL that can overcome higher pressure increase than that achievable by the former boundary layer.

The first flight test of the method was in 1941 with a Douglas B-18 via nine slots placed between 20 to 60% chord, as in Figure 2.11. They were able to maintain laminar flow up to 45% chord over the range of the Reynolds number when suction was applied decreasingly from the leading edge (Joslin 1998). Braslow et al (1951) used a porous surface in a low turbulence wind tunnel and they were able to achieve full chord laminar flow up to Re of $24 \times 10^6$ and drag reduction by approximately 60% as compared with a no-control case. This work was followed by Head et al (1955), which studied the effectiveness of the continuous distributed porous suction and the capability of its system design methodology by using
Vampire III fitted with wing glove as in Figure 2.12. They achieved full-chord laminar flow at a Mach number of 0.7 giving Re of $26 \times 10^6$ and overall profile drag reduction of 70 to 80 percent at lower Re. In addition, the inability of the theoretical prediction for optimal size and spacing and minimum suction requirement were verified by disparity between measured and estimated velocity profile. (Joslin 1998)

![Figure 2.11 Slot suction wing glove on B-18 Chambers (2005)](image1)

![Figure 2.12 Vampire III Wing glove with distributed porous suction Braslow (1999)](image2)
Although there are many experimental results showing promising potential for delaying the transition, the active LFC method raises many issues, especially for the transport aircraft application, such as the manufacturing difficulties and cost, embedment complexity, accumulation of ice, insects and atmospheric particles that can trip the TBL and maintenance cost of the system.

In an effort to overcome these problems, except the accumulation disturbance, the Hybrid Laminar Flow Control (HLFC) idea was introduced, suggesting the use of active LFC in conjunction with Natural Laminar Flow aerofoil (NACA 6 series). On its own the latter also suffered practical difficulties such as susceptibility to a cross flow instability at high sweep angle, poor high lift coefficient (AoA) performance owing to small leading edge radius and necessity for very smooth surface conditions. It was believed that the HLFC method could benefit from the compromised overall system simplification by NFL and enhanced high AoA and sweep angle performance from LFC, as shown in Figure 2.13

![Figure 2.13 Predicted benefits of NLF, LFC and HLFC](image)

Joslin (1998)
The capability of the HLFC system was well demonstrated by the flight test performed during 1990 and 1991 on the Boeing 757, employing the configuration shown in Figure 2.14 with perforated surface suction to original 757 wing surface and contour. The results showed existence of the laminar flow up to 65% chord at Mach number 0.82, as shown in Figure 2.15 and drag reduction of 25 percent giving approximately 6% overall drag reduction for the aircraft, (Maddalon 1991; Shifrin 1991; Collier 1993).

\[ \text{Figure 2.14 HLFC in practice} \]
\[ \text{a) conceptual configuration of HLFC (Joslin 1998); b) HLFC wing glove on B757 wing Chambers (2005)} \]
While the concept of the HLFC greatly enhances the feasibility of the practical application of laminar flow control and is believed to provide potential reduction in direct operating costs, for example, a 14% drag reduction from the wings, tailplane and engine nacelle of the subsonic transport aircraft A340 (Robert 1992) which consumes 80% of total fuel during cruise, the technology requires further evaluation before it can be considered as a ready technology for application. Areas requiring further study include long-term structural robustness of the system, maintenance requirements, high sweep angle performance for future application in high-speed civil transport configuration and military use, theoretical design methodology and possible optimisation of extra system requirements, such as anti-accumulation and/or anti-attachment line turbulence contamination devices.
2.4.2.2 Turbulators

This term generally describes methods involving turbulence enhancement in the boundary layer, usually to take advantage of increased mixing and tendency to postpone the boundary layer separation. These devices are best applied within the region where the transitional range of Reynolds number occurs, as they are prone to disturbances.

In the realm of aerospace industry, passive vortex generators (VGs), such as vane VGs, Leading Edge eXtension (LEX) and etc., are the most widely known and routinely used example of the turbulators. Their application is relatively simple and yet gives an acceptable performance when they are carefully oriented and located.

Since the first demonstration by Taylor (1948) in the late 1940s, who used a row of small plates projected normal to the surface and placed at an angle to the local flow, various shapes of conventional vane vortex generators have been developed and are broadly used in delaying boundary layer separation (Schubauer and Spangenber 1960), to enhance lift (Pearcey 1961 and Bragg and Gregorek 1987), and to improve wing buffet characteristics (Pearcey 1961).

These devices, usually with height compatible with the local boundary layer thickness, generate tip vortices travelling in streamwise direction and the retarded near wall flow is energised by these vortices owing to the entrained high energy freestream flow. Their performance is greatly governed by planform shape, section profile and camber, yaw angle, aspect ratio and height in respect of boundary layer thickness (Gad-el-Hak 2000). In addition, the orientation of devices relative to others determines the directional nature of the trailing vortices, producing either co-rotating or counter-rotating vortices as in Figure 2.16 and magnitude of vorticity by their interactions with each other. It is generally known that counter-rotating VGs tend to function more effectively in controlling 2D types of separation while co-rotating provides better performance in 3D separation cases (Lin 2002).

Although they are effective in delaying the separation, these devices produce considerable parasite drag during cruise condition unless they are retracted or stowed. Recent work by Lin and Howard (1989) produced substantial reduction on the size of conventional VVGs enabling the use of miniaturised VVGs with a height of only 20% of the local boundary layer but giving comparable effect, as shown in Figure 2.17 and these are known as Micro VVGs (MVVGs), as in Figure 2.18.
Figure 2.16 Spatial relation of VVGs and sketch of generated vortices pattern.
   a) Co-rotating; b) Counter rotating
   (ESDU 93024)

Figure 2.17 Oil-flow visualisation showing the separation delay of 3D flow over a
   back ward facing ramp by co-rotating MVVGs. a) normal flow condition; b) with
   MVVGs having the height of 20% of BL thickness. Lin and Howard (1989)

Figure 2.18 MVVGs on leading edge of trailing edge flap of Piper Malibu
   Chambers (2005)
Unlike the macro VVGs, however, there is a limitation on the regions where they can be used, as these devices can only be applied in situations where flow separation point/line are reasonably fixed and need to be placed quite close to the upstream of the baseline separation, usually 5 ~ 30 times the height of MVVGs, (Lin 2002).

Nevertheless, although this method greatly reduced habitual parasite drag associated with VGs to some extent, it did not satisfy the demand for closed-loop control over its actuation to give optimised performance over a wide range of flight conditions and room still existed for further improvements in its profile drag. The idea of Vortex Generating Jets triggered by Wallis (1952) provided a possibility in overcoming these drawbacks of the VGs.

When continuous air jets are blown through an orifice in a solid surface, that is skewed and pitched at an angle to local onset flow, as schematically shown in Figure 2.19, and this generates persistent longitudinal vortices which then interact with boundary layer flow as they are convected in streamwise direction producing a literally analogous effect like that of conventional passive VGs. Compton and Johnston (1992) found the optimum pitch and skew angle of 45° and 90° in producing strong streamwise vortices.

![Figure 2.19 Definition of pitch and skew of an vortex jets orientation](image)

*Figure 2.19 Definition of pitch and skew of an vortex jets orientation*  
Johari and Rixon (2003)
Selby et al (1992) successfully demonstrated the effectiveness of the VGJs in suppressing the 2D flow separation associated with low speed turbulent flow over a rearward facing ramp using an array of ten actuators as in Figure 2.20 and studied its performance parameters including velocity ratio, jet orientation and location of application.

![Figure 2.20 Oil flow visualisation of steady PVGJs effectin delaying flow separation. a) Baseline case; b) Steady jets in action with \( V_R = 6.8 \), \( \alpha = 45^\circ \) and \( \beta = 90^\circ \). Selby et al (1992)](image)

They have verified generally improved performance of VGJs with the increasing velocity ratio and skew angle of orifice values of 60^\circ to 90^\circ. Also, since they studied the 2D separation it was found orifice orientation producing co-rotating vortices more effective than that of generating counter-rotating vortices.

The overall performance efficiency of the steady VGJs was further improved when the use of pulsing jets was conceived instead of continuous jets (Wallis and Stuart 1958; Pearcey 1961), commonly known as Pulsed Vortex Generating Jets (PVGJ), as they essentially require less flow supply. Seifert et al (1993) used oscillatory tangential blowing over the hinge of the flap on an aerofoil and demonstrated the capability of the PVGJs in delaying flow separation by achieving up to 30% increase in lift. The potential benefit of smaller flow requirements of pulsed jets has been confirmed by McManus et al (1995) through flow separation control over a flap. They verified that for a given flow rate, PVGJs performed more effectively than the steady jets in delaying flow separation and can control a significantly larger surface area on an aerofoil section. Furthermore, enhanced entrainments as well as a deeper streamwise vortex penetration, that is to say, approximately 50% further into the boundary layer (Johari and Rixon 2003), compared with analogous steady VGJs have been demonstrated (Grow and Champagne1971; Bremhorst and Hollis 1990; Johari and McManus 1997). Also maximum circulation and peak vorticity during the
pulsation were found to be 30% greater than the mean values of the steady jets. (Johari and Rixon 2003)

This superior performance of PVGJ is believed arise from the structural dissimilarity in the free jet which is characterised by the formation of a starting vortex at the onset of each pulsing cycle accompanied by the usual pairs of counter-rotating vortices as shown in Figure 2.21 (Johari and McManus 1997).

![Figure 2.21 Starting vortex ring of the pulsed jet. Johari and Rixon (2003)](image)

The performance of both the steady and the pulsed VGJs is influenced by the number of parameters. Primary parameters include jet geometry (orifice shape, size and spacing), jet orientation (pitch and skew angle and their correlation with neighbouring jets; co-rotating or counter-rotating), jet velocity and additionally the pulse duty cycle and frequency of the pulsed jet case.

Many research efforts have been directed towards optimising the aforementioned parameters and in general a round jet orifice is found to be more efficient than a slot one (Godard and Stanislas 2006). Godard and Stanislas also concluded that the counter-rotating vortices can give comparable performance to the co-rotating condition, but requires extra care in their spatial and magnitude optimisation. In terms of other parameters Warsop et al (2007) provide optimal values for practical PVGJs application. They stated that the diameter of an orifice should be 5 to 15% of the local boundary layer thickness with skew and pitch angles of 90° and 45° respectively while the jet frequency response should attainable up to 500Hz with pulse duty cycle of 50% or less. Also the jet velocity should be close to the same order of magnitude as the local freestream velocity.
Traditionally, solenoid valves are used in generating pulsing jets but recent notable advances have been achieved in pulsed jet actuators by the use of synthetic jets, which are otherwise known as massless jets. This method requires only an electrical energy in the generation of pulsed jets, unlike others requiring a supply of air, and has the characteristic zero time averaged mass flux through the orifice yet it gives non-zero momentum flux. Its effectiveness in reducing the boundary layer susceptibility to separation is well established in laboratory experiments by Amitay et al (1998, 2000) and Smith et al (1998).

Although this innovative method enables simplification in interconnection and reduced power requirements, there are several practical issues, which include difficulty in controlling actuation frequency, speed of response and possible ingestion of foreign particles in the air during suction cycle, (Lockerby et al 2002). Also the diaphragm can be excited during the inactive state by Helmholtz resonance (Lockerby and Carpenter 2004). Furthermore, in order to achieve the optimal performance the diaphragm must be operated at the appropriate structural resonance frequency of a diaphragm, which can lead to a fatigue failure resulting a significant reduction in its lift cycle.

Developing a performance-optimised PVGJs actuator for incorporation in a MEMS based reactive closed-loop control still presents a difficult question and there is much room for improvements. The fluidic oscillator as an alternative possibility holds many potential, such as zero electrical power requirement in giving periodic oscillation to output jets, relatively simple connection, physical robustness giving a long life cycle, low flow requirement in operation, ease of manufacture and microfabrication.

2.5 Fluidics

In this section literature reviews on the early history and the basic operation concepts of fluidics are presented. Also principal devices to demonstrate the application of the fundamentals of fluid dynamics as the basis for the fluidic elements are described.

2.5.1 Definition and general principles of operation

Fluidics (Fluid Logic) is the term given to the study of circuitries or elements, which do not involve any mechanical moving parts and use flow or pressure of fluid
as a power supply medium to achieve analogous operation of electronic circuits or devices. The first patent of a fluidic device was issued in 1916 to Nikola Tesla (1916) for the ‘Valvular Valve’, which functioned as a check valve by making a returned flow experience a high resistance owing to the vortical motion it needed to undergo, as shown in Figure 2.22. Nevertheless, the field of fluidics started to evolve as a new technology only after the demonstration at the Harry Diamond Laboratories (HDL) in 1959 of the concepts of fluidic amplifiers utilising an analogue beam deflection amplifier and a digital wall attachment amplifier (Horton 1960). In addition to this, owing to the progress of the fundamental study on fluidic amplifiers at MIT, fluidics started to be considered as a possible alternative for a reliable system. Extensive research was then initiated actively in consideration of the robustness, superior environmental tolerance, simplicity and cost-effectiveness of the devices (Greenwood 1960; Ezekiel and Greenwood 1961; Brown 1962).

![Flow direction](image)

**Figure 2.22 Tesla valve and its principle of operation**

Tesla (1916)

Primary fluidic elements involve an amplifier, diode, modulator, sensor and switch with either analogue or digital output, which can then be cascaded into a circuit to perform complex functions such as shift register and counter. Each one of these elements generally employs one or more of four different fluid phenomena as their fundamental principle of operation and they can be summarised as:-

- Wall attachment (Coanda Effect) – tendency of turbulent jet issued from a nozzle to attach itself to the wall and flow along it.
- Momentum interaction – exchange of momentum when two or more unbounded jets impinge upon one another.
- Jet turbulence – laminar flow can be destroyed when the flow is disturbed by a large radial component of velocity.
- Vortex effect – the momentum of a stream of fluid remains constant unless external forces are introduced.
In general, fluidic elements can be categorised as analogue devices or digital devices according to the nature of their output. The analogue devices generate a smoothly varying output signal upon the receipt of control signal and the output is proportional to the input signal. But this linearity starts to break after the input signal exceeds the predefined value and further increase in input signal will result as an output signal saturation giving a constant value. On the other hand, a digital device can produce only two different states of output, i.e., on and off (high or low). The other characteristic which distinguishes these devices from analogue devices is that a change in output state from one to the other occurs rapidly through the application of the control signal. A further increase in control signal after the required state change has been accomplished has little effect on the output signal.

General examples of the analogue and the digital devices are presented in the following sections with brief descriptions of how they incorporate the aforementioned fluid phenomena in achieving their functions and a summary of the operation mechanism.

2.5.2 Analogue devices

*Beam deflection proportional amplifier:* – This type of fluidic device best represents the use of momentum exchange as an operation principle. In Figure 2.23 the general geometry of the device is presented. As the turbulent power jet issues from the supply nozzle, the flow is evenly divided between output A and B in the absence of the control flow. With the application of a control signal from the control port C₁ the power jet starts to bend towards output port A as the control jet impinges onto a power jet stream, hence increasing the output pressure at port A, while output from port B decreases. The deflection of the power jet stream is proportional to the momentum differential of the control jets while the output is proportional to this deflection of the main jet.
**Turbulence amplifier:** – The operating principle of this amplifier is based on the use of laminar jet transition to turbulence by application of a radial control flow that is much lower than the supply flow, as in Figure 2.24. The Reynolds number of the supply jet is kept low within the range of 1000 to 2000, at which the jet is laminar but very sensitive to disturbances (transitional regime). This jet then stays laminar for an appreciable distance after leaving the orifice until the control flow is introduced.

The geometry of the device presented in Figure 2.24 is usually called an Axial Turbulence Amplifier owing to its inline arrangement of the input and output nozzle. Consequently, most of the supply jet will enter into the output port in the absence of the control flow. This high output signal state can then be gradually changed by the injection of the control flow, which increases the main jet spread and consequently reduces the output flow (pressure) practically to zero. The flow from the main jet is then dumped through a vent.

**Figure 2.23 typical geometry of beam deflection proportional amplifier**
Morris (1973)

![Diagram](image)

**Figure 2.24 Axial turbulence amplifier**
a) Power jet path without control; b) Perturbed power jet with control flow
Raber and Shinn (1964)
2.5.3 Digital devices

Wall attachment bistable amplifier: – Owing to the Coanda effect, the presence of attachment walls in the vicinity of the supply nozzle will force the turbulent power jet to adhere to one of the attachment walls, depending on the instantaneous pressure gradient developed across the stream as it leaves the supply nozzle. Therefore this process is unstable in bistable devices making it hard to predict on which side the main jet attachment will be initiated. One can also see that the attachment walls are placed much closer to the supply nozzle with a smaller angle when compared with that of the beam deflection proportional amplifier in Figure 2.23 which is a major difference in their principle of operation. The attached jet forming a separation bubble on that side of the attachment wall will maintain its attachment until an appreciable level of flow or pressure, which is usually 5 to 15% of the supply flow level, is admitted to this separation bubble.

The usual geometrical configuration is defined in Figure 2.25. Imagine the main jet is attached to the Set (S) side of the attachment wall producing the high output signal from the O₆. Admission of the steady or impulsive appreciable control signal from the C₆ will detach the jet from the S and attach it to the Re-Set (R) side of the wall. Now the high state output signal is generated from the O₉ while O₆ gives a low state signal.

![Figure 2.25 Typical wall attachment geometry](image)

(Morris 1973)

Vortex amplifier: – The principle of this device is most easily explained with the aid of Figure 2.26. The power jet will continue its original path towards the centre and escape into the output port during normal operation. When the control flow is applied the resulting power jet is forced into spiral motion towards the centre. This then effectively increases the flow path length and accelerates the flow as they
spiral towards the output port owing to the centrifugal effect and both of these resulting effects lead to pressure drop in the main jet.

![Vortex Amplifier Diagram](image)

**Figure 2.26 Typical geometry of vortex amplifier.**
(Morris 1973)

Another crucial assumption in the principle of operation is that when the radius of the output port is less than that of the limit of a circle an inviscid fluid whirls within the vortex chamber without ever leaving the output port. However, this is not achievable in real application.

### 2.6 Bistable wall attachment fluidic oscillator

The basic concepts of the operation of fluidic devices have been introduced earlier. In this section of the report an extensive study of the wall attachment fluidic amplifier is presented, since the actuator employed for the flow control application falls within this category of the fluidic elements. The study includes a literature review on their concepts, previous analytical and experimental work and geometrical parametric considerations. The section will end with a description of the conceptual design developing methodology of the unit.

#### 2.6.1 Early history

Ever since the introduction of fluidics, the bistable wall attachment fluidic amplifiers received by far the most attention both from industries and universities. But its discovery was rather accidental. After Horton (1960) conceived the possibility of deflecting a fluid jet with another jet of much smaller flow, many experiments with
round jets have been performed. It was soon found, however, that the efficiency of this method was too low.

Raymond Warren and Ronald Bowles made use of cover plates, which effectively changed the power nozzle to a slit nozzle, as an effort to improve the gain of the proportional beam deflection amplifier suggested by Horton (Kirshner and Katz 1975). This provided an increased gain owing to a pressure gradient which developed across the power jet as the result of an entrainment in the vicinity of the power jet nozzle. Nevertheless, the pressure effect became momentous, causing the power jet to attach to the wall and giving the amplifier a bistability rather than the desired monostability. Figure 2.27 shows the typical components and corresponding symbols forming the geometry of wall attachment devices.

This rather fortuitous discovery later gained wide recognition and formed the basis for the development of a special type of wall attachment device that produced a self-sustained periodic oscillation in its output, later known as a bistable fluidic oscillator. Various designs of fluidic oscillators have been suggested and patented since then, though they are all based on the fundamental principle of operation associated with the two classic designs shown in Figure 2.28. Henceforth, for ease of reference Figure 2.28 a) will be known as a control loaded fluidic oscillator while the term vent-fed oscillator is reserved for Figure 2.28 b).

**Figure 2.27** Typical wall attachment device geometrical components and symbols.
2.6.2 The Coanda effect in bistable wall attachment devices

The ground rule of operation of wall attachment devices is based on the well-known turbulent re-attachment principle (Coanda Effect). The coanda effect states that when a turbulent jet issues from a nozzle it will bend towards any nearby surface or wall and attach itself to it further downstream, unless the surface is displaced too far from the orifice and/or is sufficiently short and/or inclined at a large angle to the centreline of the jet.

Figure 2.28 Classic bistable fluidic oscillator designs
a) Control loaded oscillator; b) Vent-fed oscillator
Morris (1973)

Figure 2.29 Jet attachment in a bistable wall attachment device with schematic presentation of a separation bubble. Kirshner and Katz (1975).
The Coanda effect was first noticed by Thomas Young as early as 1800 (Young 1800) and rediscovered by Chilowski and Coanda in 1930. In 1936, Henri Coanda acquired a patent for a ‘Device for Deflecting a Stream of Elastic Fluid Projected into an Elastic Fluid’ (Coanda 1936) and later the phenomenon was entitled the ‘Coanda Effect’ by Metral and Zerner in 1948.

In bistable wall attachment devices, as the turbulent jet issues from the power nozzle it entrains fluid from its surrounding which is confined by the presence of the top and bottom plates and attachment walls. This enables low pressure regions to develop between the jet and attachment walls, owing to the entrained fluids, causing a counterflow to issue from the downstream of the wall towards the low pressure region, as schematically shown in Figure 2.29 a).

Then a momentary disturbance, usually the turbulence in the jet, causes the jet to bend towards one side of the wall, decreasing the recirculating flow and hence the pressure on that side of the jet. As a result of this increased differential pressure across the stream of the jet, it will bend towards that side of the wall further and attaching itself on to the wall, as shown in Figure 2.29 b).

The jet attachment will lead to a formation of trapped pocket of fluid (henceforth a separation bubble). The state of equilibrium is achieved at the point of attachment when the mass flow in the bubble is constant, i.e. when the mass flow fed back into the bubble from the jet owing to the vortex flow inside the bubble is equal to the entrained flow by the power jet from the bubble.

For a symmetric bistable oscillator the side of the jet attachment is totally governed by the naturally introduced momentary disturbances, hence making it an indeterminable process. On the other hand, in an asymmetric monostable oscillator the power jet will always attach to a predetermined side where the entrainment is more restricted.

Other work forming the basis of a fundamental study of these devices was published by Kline (1958). Effects of the diffuser angles on the Coanda effect were presented. He showed how the flow can separate from one side and flow along the other at an appreciable diffuser angle and that at wider diffuser angles the flow will separate from both walls. Bourque and Newman (1960) concluded from their
experimental results that the maximum allowable attachment wall angle was $67^\circ$ for a bistable wall attachment device with zero offset, as shown in Figure 2.35. They stated that at an angle greater than $67^\circ$, the state of equilibrium condition of the separation bubble will begin to break down owing to the returned flow being higher than the entrained flow and making the jet detach from the wall.

2.6.3 Jet switching method and mechanism

Power jet switching in fluidic devices is commonly achieved by the application of positive/negative pressure and/or jet momentum control signal through the control ports. Further admission or removal of the control signal after the completion of the jet switching will not cause any changes in its status, but if an excessively strong signal is introduced this can overdrive the device which forces a the portion of the power jet out to the inactive output port and/or vent. Morris (1973) states the typical ratio of the control signal as 5 to 15% of the supply flow level.

Although any fluidic oscillators can utilise both types of control signals, there is a marginal difference in the geometry of interaction region between pressure-driven devices and momentum-driven devices. In general, momentum devices have larger clearance, D, between the edges of power nozzle and control nozzles, as shown in Figure 2.30

![Diagram of jet switching method](image)

**Figure 2.30 Definition of clearance**
Kirshner and Katz (1975)

An instantaneous sealing of a control port on the inactive side or increase in the load on the output port of the active side can be regarded as another form of a control signal, which the sealing method known as vacuum switching.
Warren has established three different possible switching mechanisms in wall attachment devices in a flow visualisation study and his classifications are described below;

*Terminated wall or bleed type switching* – If vents or bleeds are introduced upstream of a splitter and considerable offset is given, a gradual addition of control flow into separation bubble will enlarge it hence delaying the attachment point. Once the attachment point moves beyond the point where the vent intercepts the provided attachment wall, flow through the vent is fed into the bubble, making the switching process continue even if the control flow has ceased.

*Contacting-both-wall switching* – This mode of switching can be observed when an amplifier has small offset, which makes the power jet to attach shortly after leaving the power jet nozzle and hence the usual separation bubble does not form and is replaced by a small recirculation bubble formed within a control port. In this situation slight application of a control signal will initiate development of a separation bubble and its expansion from the originally attached side, causing a proportion of the opposite side of the power jet to adhere to that side of the wall. Different from terminated wall switching process, removal of the control signal at an early stage of the jet switching will make the jet re-stabilise back to its initial state. A schematic sketch of this switching process with descriptions can be found in Figure 2.31.

*Splitter switching* – When vents or bleeds are placed considerably further downstream of the splitter, the separation bubble grows and makes contact with a splitter. This then separates a proportion of the flow from the main jet and the bubble starts to open up. If the control flow is withdrawn at this point flow from upstream starts to fill the gap created as the bubble opens up and hence the switching continues. A schematic sketch of this switching process with descriptions can be found in Figure 2.32.

After the jet attachment in a control loaded oscillator, the separation bubble formed on the attached side wall further increases the pressure difference, generating a rarefaction wave (compression) on that side of the control port while a pressure wave (expansion) grows on the opposite side. These waves travel in an opposite direction along the feedback loop connecting the two control ports, and on its completion, the power jet will detach from the initially attached side wall and attach onto the other side. This detachment and attachment process continues, giving the
periodic oscillation. The involvement of the pressure waves in their oscillation mechanism allows them to be regarded as pressure-driven devices.

a) Initially the power jet is attached to the left side attachment wall. The usual formation of a separation bubble on the attached side wall is replaced by a vortex (recirculation bubble) formed at the control nozzle owing to a small or zero offset. A standing vortex formed further downstream can also be seen.

b) As the control flow is applied the recirculation bubble transforms into a separation bubble and expands it. This causes the right side of the power jet to make a contact with an off side wall. If the control signal ceases at this point the power jet will re-stabilise itself back to its initial state.

c) The separation bubble expands further and moves downstream. The standing vortex is suppressed in its size and its core starts to shift towards the left side output. A recirculation bubble starts to form at the right control.

d) The power jet has now attached to the right wall. The separation bubble on the left side of the jet enlarges further and reaches the vicinity of the splitter while the standing vortex is pushed out to the left output.

e) The jet switching is completed and the power jet is now attached to the right side wall. The separation bubble now forms a standing vortex.

Figure 2.31 Sequential illustration of a contacting-both-wall switching process.
a) Initially the power jet is attached to the right side of the nozzle and given a considerable offset a separation bubble forms on that side.
b) As the control flow is introduced to the separation bubble it increases in size and a portion of the power jet is piled by the splitter, forming a standing vortex.
c) The separation bubble further increases its size and most of the power jet flow is captured by the splitter, forming a larger standing vortex, while the previous separation bubble is now opened (the jet is now permanently detached from the initially attached side). As the standing vortex makes contact with the left side wall, pressure difference increases between the right and the left side of the power jet causing the jet to bend further towards the left side. Power jet flow exits through both of the output conduit.
d) Power jet switching is completed and a separation bubble is formed on the left side wall.

**Figure 2.32 Sequential illustration of a splitter switching process**
The oscillation mechanism in a vent-fed oscillator partially incorporates pressure change but is mainly driven by a momentum injection through the control ports (momentum-driven devices). Following the initial attachment, part of the attached flow will be captured by the vent port located on the corresponding side of the exit conduit. This will then be fed to that side of the control port through the feedback connection and once sufficient momentum has been injected the power jet will detach to the opposite side. The same process takes place again and periodic oscillation is generated.

In general, all three switching mechanisms described above participate to some extent in power jet switching of most fluidic devices. The specific designs of fluidic oscillators given in Figure 2.28 and the current test model, however, which have small or zero offset and are without a vent orientation, will incorporate dominantly the contacting-both-wall switching and the splitter switching mechanisms during their oscillation process. Also their period of oscillation, $T$, can be approximated by the Eq. 2-2 on the assumption that the propagation time, $\tau_p$, is the function of the length of the feedback loop, $L$, and the local speed of sound, $a$, as in Eq. 2-3. The $\tau_s$ in Eq. 2-2 is the switching time of the device which is governed by the input and output characteristics and the device and the feedback loop geometry. Further details of the propagation time and the switching time can be found in Chapter 4.

$$ T = 2 \cdot \tau_s + 2 \cdot \tau_p $$ \hspace{1cm} \text{Eq. 2-2} \\
$$ \tau_p = \frac{L}{a} $$ \hspace{1cm} \text{Eq. 2-3}

### 2.6.4 Static characteristic

The first attempt to study the Coanda effect theoretically is known to be that of Dodds (1960). His analysis is based on a principal assumption that the momentum flux is conserved in a volume surrounding the attachment point, $a_5$.

As shown in Figure 2.33, the input momentum $J$ is divided into downstream and upstream (recirculating) components of $J_1$ and $J_2$, giving the reduced equation as Eq. 2-4

$$ J \cos \theta = J_1 - J_2 $$ \hspace{1cm} \text{Eq. 2-4}

and this approach is known as the attachment point model.
The angle $\theta$ at which the jet strikes the wall is then found with the further assumptions that the jet centreline is a circular arc of radius $R$ and that the velocity distribution is the same as the free jet, employing Görtler's analysis, up to the point of reattachment.

In addition, the following assumptions are generally made for the development of the analytical description of the steady state characteristic, (Levin and Manion 1962):

- The jet flow is turbulent, incompressible and two-dimensional.
- The jet velocity is uniform at the nozzle exit.
- The jet entrains the same mass flow from each boundary up to the point of reattachment.
- The jet velocity is independent of the reduced pressure in the separation bubble.
- The pressure within the separation bubble is uniform.
- The shear force owing to the wall is neglected.
- The nozzle width is small compared with the arc radius $r$; and the attachment wall length is long compared with the nozzle width.
- Changes in the jet structure owing to the centrifugal force of curvature are negligible.

Based on these assumptions, extensive analyses have been performed by many researchers and notably by Bourque and Newman (1960) and Sawyer (1960, 1963).
Bourque and Newman (1960) first studied the case of parallel wall with an offset by approximating the $J_1$ and $J_2$ as in Eq. 2-5 and Eq. 2-6 using the corresponding free jet values and the jet stream velocity approximated using a Görtler’s profile at a distance $S = c_i c_6$ along the jet centre line and over the distance $y_a = c_a c_6$ normal to the jet centre line as shown in Figure 2.33. The $y$ in Eq. 2-5 and Eq. 2-6 is the distance from the jet centre line to an arbitrary point on a line normal to the centre line, i.e., the $y_a$. For a detailed description of the Görtler’s profile, please refer to Kirshner and Katz (1975).

\[
J_1 = \int_{-\infty}^{y_a} \rho u^2(s, y) \, dy \quad \text{Eq. 2-5}
\]

\[
J_2 = \int_{y_a}^{y_c} \rho u^2(s, y) \, dy \quad \text{Eq. 2-6}
\]

The resulting theory achieved good agreement with the experimental data when choosing the jet spreading factor $\sigma$ as an empirical constant of 12, as in Figure 2.34 and both experimentally and analytically demonstrated the functional relationship between the attachment distance, $x_R$, and the offset, $B$, as in Eq. 2-7.

\[
x_R / b_3 = f(B/b_3) \quad \text{Eq. 2-7}
\]

Figure 2.35 also shows that the maximum allowable inclination of the wall is around 67° and beyond this value the attachment length becomes infinite.
Figure 2.34 Effect of offset in jet attachment length
Bourque and Newman (1960)

Figure 2.35 Effect of attachment wall angle on the jet attachment length for various offset values
Broken line – Sawyer’s (1963) analytical result
Solid line – Bourque and Newman’s (1960) theoretical results and Bourque and Newman’s (1960) experimental results
Sawyer (1963)
They also attempted a different approach to the problem, namely the control volume model. The major dissimilarity between the control volume model and aforementioned approach is the negligence of the recirculating momentum flux and introduction of the separation bubble pressure, \( P_x - P_B = \Delta P \), giving the resulting expression for \( \theta \) as in Eq. 2-8, on the assumption that the momentum flux returned to the low-pressure region, \( P_B \), balances the pressure difference times the area normal to the wall.

\[
J \cos \theta = J_1 \quad \text{Eq. 2-8}
\]

The results of this model, however, were not as good as those of the attachment point model and furthermore none of these models gave good representation of the experimental data for the cases of inclined wall with/without offset.

Levin and Manion (1962) extended both models of Bourque and Newman to the study of inclined offset wall cases, which led them to the following expression of momentum relation for the cases of attachment point and control volume theory respectively, as in Eq. 2-9 and Eq. 2-10.

\[
J \cos (\theta + \alpha) = J_1 - J_2 \quad \text{Eq. 2-9}
\]

\[
J \cos \alpha - J_3 = J_1 \quad \text{Eq. 2-10}
\]

They compared their theory with the experimental data obtained from a test conducted with a 0.5 Mach air jet, varying the offset distances of 0, 2, 4 and 10 nozzle widths and the wall angles from 0° to 55°. The attachment point theory achieved satisfying agreement with the experimental data for most of the offset values except the value of 10 at angles up to 35 degrees, when \( \sigma \) chosen as 15 and/or 20 as in Figure 2.36 Their modified control volume theory only agreed with experimental data when the spread parameter was assumed as a functional variable of offset and wall inclination. Their results fro the control volume method are shown in Figure 2.37 and the deduced representative values of \( \sigma \) for various combinations of offset and attachment wall angle in Figure 2.38.

Sawyer (1963) separately worked on developing the jet attachment model to describe both the cases of offset parallel wall and inclined wall without offset based on the work of Dodd. Major divergences from the work of Bourque and Newman involve the appreciation of difference in the entrainment depending on the jet curvature i.e. entrainment for the jet curving over a convex surface is greater than
that of the free jet and lower for the jet curving within a concave surface and the inclusion of pressure force in the vicinity of the attachment point based on the assumption that the pressure distribution is similar to that of the forced vortex.

His result matches well with experimental data for both the offset parallel wall and inclined wall, as in Figure 2.35, without the necessity of assuming spreading factor to be an empirical constant but using that of the free jet value of ~7.7.

![Figure 2.36](image)

Figure 2.36 Comparison of experimentally measured jet attachment length with theoretical model of Levin and Manion based on attachment point method.

a) $\sigma = 15$; b) $\sigma = 20$

Levin and Manion (1962)
Figure 2.37 Comparison of experimentally measured jet attachment length with theoretical model of Levin and Manion based on control volume method for \( \sigma = 8 \). Levin and Manion (1962)

Figure 2.38 Approximate value of \( \sigma \) for combinations of attachment wall angle and offset ratio for Levin and Manion’s control volume method

Levin and Manion (1962)
Boucher (1968) improved and simplified Sawyer's model and analysed offset parallel and offset with inclined wall cases and his theory provides the best agreement with the experimental data for the offset parallel wall case, as it can be seen from Figure 2.39.

Figure 2.39 Theoretical results of jet attachment length for the cases of various offset parallel wall by Boucher's compared with other experimental results

Boucher (1968)

In his analysis he substituted Simpson's jet profile for the modified Görtler's profile used by Sawyer, and further simplification was achieved by disregarding the discrepancy in entrainment between both sides of the jet. In addition to this the model allows variation of the jet width by defining the inner edge line of the jet along the point where \( n = u/u_e \) equals the empirically determined constant. When \( n=0.05 \) the theory best represents the data.

From successful representation of the offset parallel cases it was verified that the effect of jet curvature on the entrainment rate is negligible, and Boucher re-emphasised the importance of the appreciation of the pressure force by showing good agreement with experimental data.
Boucher’s analysis reasonably matches well with data for offset inclined wall cases up to 30 degrees except for the zero offset case as shown in Figure 2.40. He states that this deficiency is because of the development of the much smaller separation bubble associated with zero offset inclined wall leading to the establishment of much faster internal flow movement within the bubble; therefore the developed pressure gradient will divert the jet centre line from being the circular arc.

**Figure 2.40** Boucher’s theoretical results of jet attachment length for the cases of various attachment wall angle with offset wall compared with other experimental results. a) Zero offset case; b) for offset ratio of 2, 4 and 10.

Boucher (1968)

Following these works, researchers have now started to verify the effects of other fundamental parameters on the performance characteristics of wall attachment devices.
Comparin et al. (1962) investigated the effects of various geometrical parameters, which includes power nozzle aspect ratio (AR), splitter distance and attachment wall angle and length (Raber and Shinn 1964), on the minimum Reynolds number for the jet attachment (critical Reynolds number) and it was found that the AR had a dominating effect while the rest of the parameters were not significant influences.

His result, as shown in Figure 2.41 indicates that for power nozzle aspect ratios up to 2, the critical Reynolds number does not change greatly, but then starts to increase with further reduction in AR. Even though this may well represent the trends in the variation of critical Reynolds numbers with AR, one should recognise that the critical Reynolds number depends on many interrelated parameters that form the configuration of the device under consideration.

![Figure 2.41 Variation of the minimum Reynolds number based on the power nozzle width for the initiation of jet attachment with AR. Comparin et al (1962)](image)

Sarpkaya and Kirshner (1968) studied the effects of attachment wall profile on the performance characteristics of a vented and unvented wall attachment bistable amplifier with a cusped splitter. Their designs for amplifiers with geometrical parameters and splitter shape can be found in Figure 2.42
Figure 2.42 General amplifier geometry and profiles of splitter and convex and concave attachment wall studied by Sparkaya and Kirshner. Sparkay and Kirchner (1968)

They argued that the pressure and power recovery of unvented amplifiers are an extension of the corresponding values of the vented amplifiers sharing the same geometry. Also, they have shown that the use of a convex attachment wall can give improved performance compared with straight and concave walls and their results are shown in Figure 2.43. As they expected, a concave wall presented a similar
effect of increasing the offset of the wall which results as increased separation bubble size and hence increased pressure loss.

Figure 2.43 Pressure recovery with active port flow
a) Straight attachment wall for closed control ports operation
b) Concave attachment wall for closed control ports operation
c) Convex attachment wall for closed and open control ports operation
Sarpkaya and Kirshner (1968)
2.6.5 Dynamic characteristic

Johnston (1963) studied the switching mechanism of a single-sided wall fluid amplifier without a splitter and a receiver. In his analysis he assumed that jet detachment occurs when the separation bubble has sufficiently enlarged. Johnston and later Katz and Dockery (1964) arrived at the same conclusion, namely that the switching time is a function of the control flow rate and it decreases with increasing control flow level. He defined the switching time of the amplifier as in Eq. 2-11, where $\Delta V$ is the volume change in the separation bubble, $Q_{cav}$ is the average entrainment of the control flow into the power jet during the switching and $Q_c$ is the control flow level. He showed that $Q_{cav}$ increases with the Reynolds number and $\Delta V$ is independent of the Reynolds number and $Q_c$. Also he demonstrated that $Q_{cav}$ is independent of $Q_c$. These results are shown in Figure 2.44 to Figure 2.46 and the axis labelled as a separation time in the Figure 2.44 is equivalent to the switching time in this report.

$$t_s = \frac{\Delta V}{Q_c - Q_{cav}}$$  \hspace{1cm}  \text{Eq. 2-11}

![Figure 2.44 Separation time variation with control flow level](Johnston (1963))
Figure 2.45 Dynamic separation bubble area change with power jet Reynolds number. Johnston (1963)

Figure 2.46 Measured control flow level and calculated average entrained flow. Johnston (1963)
Savkar (1966) carried out an extensive experimental study on the effects of various geometrical parameters on switching time of bistable fluid amplifiers. This included the variation of control to power nozzle width ratio of $1 \sim 0.25$, setback between $0 \sim 4b_S$, offset and splitter location from $6 \sim 39b_S$.

He described the switching time as the sum of two components, as in Eq. 2-12 where $\tau_i$ is the minimum pulse length of the control signal to initiate the switching and $\tau_b$ is the additional time required to receive the output signal down the inactive conduit. The beginning of their switching time was defined as the instance of the control signal application and its termination when the power jet reached a predefined point within the inactive conduit.

$$\tau_a = \tau_i + \tau_b$$  \hspace{1cm} \textbf{Eq. 2-12}

His results show the reduction in the switching time with the offset as this required shorter control pulse and a significantly increased dynamic flow gain has been achieved as in Figure 2.47 and Figure 2.48. He also concluded that the momentum ratio of the control jet to the power jet plays an equally important role to that of the flow rate ratio in determining the switching time. This was derived by recognising the decrease in actual switching time as the width of the control nozzle reduced for a given control flow rate and demonstrated the minimum momentum ratio of 0.2 for a bistable amplifier without a splitter. Unlike other parameters studied in his work, the setback presented a rather negligible effect on the switching time.

While none of the aforementioned parameters had direct effects on the power jet attachment, the splitter location had considerable influence to a certain extent. When the splitter is closer than 16 throat width from the plane of the control jet, the power jet develops an increasing tendency to oscillate about the splitter before its attachment. Moreover the jet attachment will completely cease if the splitter moves further upstream than 2 throat width downstream of the centreline of the control jet, as can be seen in Figure 2.49

Savkar also demonstrated increase in the switching time as the pointed splitter moved upstream and his results can be found in Figure 2.50. He stated that this is because of the increased adverse interference from the recirculation vortex forms near the splitter, which suppresses the growth of the separation bubble under the application of the control signal and this is demonstrated in Figure 2.51.
Much effort (Lush 1967, 1968; Ogzu and Stenning 1972; Goto and Drzewiecki 1973; Drzewiecki 1973) have been expended in developing a theoretical modelling of the switching in a bistable fluid amplifier. Goto and Drzewiecki’s (1973) results agreed well with experimental data without the necessity of employing empirical constants. Their model can compromise the effects of vents, splitter, and opposite wall, flow status of the power jet (laminar and turbulent) and input and output impedances. Drzewiecki (1973) further improved the model to allow the analysis of switching by the application of suction to the inactive side control port.

![Figure 2.47 Control pulse length variation with offset](Savkar (1966))
Figure 2.48 Effect of offset on dynamic flow gain

Savkar (1966)
Figure 2.49 Flow visualisation showing jet attachment to a splitter when placed near to the power jet nozzle; Splitter located two nozzle widths downstream of the plane of the control nozzle. Savkar (1966)

Figure 2.50 Effect of splitter location on the switching time
Savkar (1966)
Figure 2.51 Flow visualisation showing the standing vortex formed near the splitter and their influence on the separation bubble
a) 22 power jet nozzle width away from the power jet nozzle; b) 14 power jet nozzle width away from the power jet nozzle. Savkar (1966)
Early research works and their results verifying the effects of various geometrical parameters associated with the designs of wall attachment fluidic oscillators have been described so far. Although the effects of these parameters on the performance of a device are strongly interrelated with each other and can only be optimised empirically, in the following paragraphs a general overview of their effects is provided and commonly used values are summarised for reference in the following subsection 2.6.6.

The primary geometrical parameter is the power nozzle width and the rest of the commonly employed values will be expressed in terms of this. Typical values of the power nozzle width range from 0.25mm to 0.5mm with AR of 1 to 2, while the power nozzle aspect ratio of two gives the maximum pressure gain. A very high AR (O(10)) can be used but a device employing high AR tends not to function very well, possibly owing to its susceptibility to promoting a swirl flow in the power jet. At low power nozzle aspect ratio, boundary effect at the power nozzle becomes important and increases the critical Reynolds number for jet attachment to occur: this is believed to be because of the turbulence attenuation caused by the boundary layer and consequent reduction of the jet entrainment rate.

Offset and wall angle introduce similar effects to a device. At large offset and angle the jet attachment may not occur at low supply flow level and at small values the jet may attach to both of the attachment walls or can result in loss in steady-state stability. The jet attachment length reduces with both of the parameters as less entrainment is allowed and hence forms a smaller separation which results as reduced pressure loss and increased control flow requirement. Commonly used offset to power nozzle width ratio falls within zero to one and attachment wall angle ranges from 12 to 15°. These values should be kept to a minimum allowed value if one requires to operate the device at low supply flow level but at the expense of dynamic stability.

Usual splitter distance ratio lies between eight and fifteen. The effect of this value is dependent, to some extent, on the offset and its shape. In general, at low splitter distance higher level of control flow is required for power jet switching owing to the increased adverse interference from the standing vortex formed near the splitter and the strength of this vortex becomes stronger as its shape becomes blunter. General splitter shapes are pointed, round, square or cusped. In addition, at low values edgetone may occur and further reduction will cause the jet to attach to the splitter.
For preliminary design work, control nozzle width usually equals the size of the power nozzle width with a zero setback. Setback has negligible effect both on the jet attachment and switching, but care must be taken to avoid placing the control port downstream of the jet attachment point, resulting in part or most of the power jet exiting through the control port. For a given flow level of control and supply, reduction in the control nozzle width can lead to earlier switching as a result of increased momentum addition. Increase in attachment length can result from the increased control nozzle size owing to greater entrainment being allowed.

A vent can make a device become output load insensitive as the reversing flow or excessive flow can be evacuated. This greatly reduces performance efficiency, however, as one would expect.

The further significance of these parameters on wall attachment devices is shown in Figure 2.52, which has been qualitatively summarised by Warren (1963).

![Diagram](image)

**Figure 2.52 Summary of geometrical influence on the performance of bistable wall attachment device.** (Warren 1963)
2.6.6 Steady-state design methodology

The steady-state modelling procedures presented here are according to Kirshner (1975). They offer one the chance to develop a conceptual design of a bistable amplifier, which can easily be manipulated for a fluidic oscillator, based on the analytically determined and commonly used principal geometrical parameters forming the interaction region of the device. Then these chosen values can be used in determination of the steady-state input and output characteristics.

One can start developing a device by specifying these fundamental values within the allowed ranges with consideration of their general effect as discussed earlier to suit the functional purpose of the unit. This then allows an approximation of the attachment wall length based on the approximated power jet attachment distance, \( x_R \), using the Figure 2.53, which should be four times greater than the value of the \( x_R \). This factor of four is multiplied to account for the fact that when controls open to the atmosphere the attachment length is twice as long as that of the closed control case and to ensure stability.

Figure 2.53 Attachment length with attachment wall angle for various offset
Kirshner and Katz (1975)
The approximation of the input characteristic curve for the particular geometry developed would be the next step. Using Figure 2.54, the blocked control pressure, $P_g$, can be estimated allowing for the approximation of point A in the Figure 2.55 which is the blocked control condition, as a starting point of the characteristic curve.

![Figure 2.54 Blocked bubble pressure with offset (Kirshner 1975)](image)

Figure 2.54 Blocked bubble pressure with offset (Kirshner 1975)

The control flow required on the active side to switch the jet to the inactive side can be estimated with Figure 2.56 that is deduced from the experimental data of Lush
(1967 and 1968), Moses and McRee (1969) and Kimura (1972) for attachment wall angle of 15° or can be approximated using the empirically derived Eq. 2-13

\[ \frac{q_{cs}}{q_s} = 0.3 - 0.16 \left( \frac{B}{b_s} - 1 \right)^2 \]  

\[ \text{Eq. 2-13} \]

At the switching point Drzewiecki noticed a sudden drop in control flow on the active side by 30% of \( q_{cs} \) (Kirshner 1975), which can be manipulated to determine the flow level of the inactive side control port as 0.7 \( q_{cs} \).

![Figure 2.56 Control flow to switch with offset (Kirshner 1975)](image)

Then the control pressure at the switching point, \( P_{cs} \), can be calculated with Eq. 2-14 by assuming that the total pressure is completely conserved in the dynamic pressure of the control flow with an associated discharge coefficient, \( C_{dc} \).

\[ P_{cs} = \frac{D}{2} \left( \frac{0.7q_{cs}}{C_{dc} \cdot A_c} \right)^2 \]  

\[ \text{Eq. 2-14} \]

This then completes the input characteristic curve approximation by providing the coordinates of the points S and R in Figure 2.55.

The final stage of the steady-state design is the approximation of the output characteristic curve. The blocked output pressure recovery, \( P_{ob} \), as indicated by
point B in Figure 2.57 can be deduced from Figure 2.58 as predicted by the theoretical model of Goto and Drzewiecki (1973) for $l_s/b_s$ between 8 and 16.

Figure 2.57 Input and output characteristic curve for bistable device. (Kirshner 1975)

Figure 2.58 Blocked output parameter with offset for various attachment wall angle. (Kirshner 1975)
This then allows the estimation of the output flow at the switching point and consequently the fan out ratio of the unit using Eq. 2-15 together with the fact that \( P_o = P_{cs} \) at the switching point with a square law impedance assumption for the output.

\[
P_o = P_{oB} - \rho \left( \frac{q_{os}}{A_o} \right)^2
\]

Eq. 2-15

The output width, \( b_o \), in calculating the \( A_o \) can be reduced by Eq. 2-16

\[
b_o = \left( \frac{l_s^2 + \left[ B + b_s/2 \right]^2}{2} \right)^{1/2} \sin\left( \alpha + \tan^{-1}\frac{B + b/2}{l_s} \right)
\]

Eq. 2-16

Then the fanout ratio, \( n_F \), can be found with Eq. 2-17

\[
n_F = \frac{q_{os}}{q_{cs}}
\]

Eq. 2-17

## 2.7 Fluidic devices in flow control application

In recent years, there has been a lot of published work suggesting the use of fluidic devices for flow control applications. These research efforts range from studying the fundamental performance characteristics of innovatively designed fluidic devices to validation of a device in laboratory scale internal/external flow control applications. Also, notable numbers of publications have been made on studying the feasibility of incorporating other known flow control actuators, such as piezoelectric transducer or plasma actuator, into a fluidic amplifier in order to gain control authority of the jet switching independent of supply flow level.

Yang et al (2007) analysed the geometrical influences on the oscillation stabilities and pressure loss of a single output beam deflection fluidic oscillator. For this study, a single output vent-fed bistable fluidic oscillator is used, but with a novel design step-shaped attachment wall and acute-angle splitters; its performance is compared with a standard plane-attachment wall device. The schematic drawings of the fluidic device are shown in Figure 2.59, while Figure 2.60 shows the general internal flow pattern inside the fluidic oscillator.
Prior to the introduction of a step-shaped attachment wall, Yang et al. (2007) performed geometrical optimisation work on the plane-attachment wall model by varying the span angle (from 5° to 45°) and splitter angle (from 70° to 120°) as shown in Figure 2.61. Yang et al (2007) state that a span angle greater than 40° would cause the formation of a large standing vortex that results in a ceasing of the oscillatory motion over the tested flow rates (0 ~ 20 L/min). Also, they conclude that a 10° span angle does not provide enough space for stable oscillation. They further suggest that the optimum splitter angle should be between 80° and 90°, to prevent it from developing a pressure drag exerted on the main flow.
For further analysis, Yang et al. (2007) chose the span angle of $20^\circ$ and a splitter angle of $80^\circ$. In an effort to remove the development of an unstable oscillation in the plane-wall model, at flow rate higher than $15\text{L/min}$, Yang et al. (2007) introduced an extra step to the attachment wall. They state that this instability arises due to the control flow not overcoming the latching stability of the standing vortex at high flow rates. The spectral measurements and the internal flow pattern of the two fluidic models are shown in Figure 2.62 and Figure 2.63.

**Figure 2.62 Spectra cascade.** a) plane-wall oscillator, and b) step-wall oscillator. Yang et al (2007)

**Figure 2.63 Visualisation of the internal flow for comparison of flow structure.** a) plane-wall oscillator, b) step-wall oscillator. Yang et al (2007)

As can be seen from the above results, stable oscillation (with clear harmonics) is produced and a recognisable reduction in the standing vortex strength has been achieved as expected by Yang et al (2007) when using a step-wall oscillator. This
effectively widened the operating range of the model giving a range of 5L/min to 65L/min. However, an increase in pressure loss has also been noted with a step-wall oscillator, as Figure 2.64 shows.

![Graph showing pressure drop vs. flow rate](image)

**Figure 2.64 Total pressure drop as a function of flow rate.** Yang et al (2007)

The results for Yang et al (2007) demonstrate an improvement in oscillation characteristics with an introduction of a step to attachment walls, nonetheless, the author believes the effect is similar to that of an offset based on the gain of dynamic stability with an increase in pressure loss. Also, as the introduction of a step-wall effectively altered the cross sectional area of the vent connected to the feedback channel, possible change in the control flow momentum must have been considered.

Furlan et al (2006) visualised the internal flow pattern of a meso-scale vent-fed fluidic oscillator for Reynolds number (based on hydraulic diameter) range of 1000 < Re < 10000 (equivalent to supply flow rate range of 60ml/min to 600ml/min) with a liquid dye method. The geometrical specification of the model can be seen in Figure 2.65. Throughout the evaluation a step reduction in the width of the control port has been performed from ≈1.3mm to ≈0.4mm, however Furlan et al (2006) only presents the results of the two configuration shown in Figure 2.65 (b) and (c).
Figure 2.65 Geometrical specification of the model (dimensions in mm). a) global dimension, b) model with large control port width, and c) model with small control port width. Furlan et al (2006)

Figure 2.66 visualises successful jet switching of the meso-scale fluidic oscillator with a relatively large control port width (≈0.9mm) at both laminar and turbulent flow regimes, showing a sequentially guided main jet flow into both output plenums. However, close examination of the results shown in Figure 2.66 iii), obtained with a narrow control port (≈0.4mm) model, reveals an absence of the jet switching in the main jet flow. This shows rather an even distribution of the main jet flow into both of the output conduits and almost a stagnating flow occurs in the feedback arms. Although, the model with a relatively narrow control port width found not to produce intended jet oscillation, Furlan et al (2006) recommend possible use of this for the retention of emulation and/or particles.

Culley et al (2003) assessed the use of a bistable fluidic oscillator in separation control of a stator vane in a low speed axial compressor. The vane design employs three embedded bistable fluidic oscillators with each output port of the oscillator connected to two separate plenums. Each output plenum covers 20% of span and two plenums are separated in a streamwise direction by a 4% chord length. To achieve a spanwise uniformity of the exit jet flow, a row of holes on the vane skin (0.73mm in diameter with 30° pitch angle and at a 35% chord length from the leading edge on the suction side) is used over the each output plenum, as shown in Figure 2.67. The phase differences, however, between the three fluidic oscillators could not be controlled due to the bistability of the devices.
Figure 2.66 Internal flow visualisation with liquid dye. i) operation of oscillator with control port width of 0.87 at Re=1038, ii) operation of oscillator with control port width of 0.87 at Re=9737, and iii) operation of oscillator with control port width of 0.4 at Re=1168. Furlan et al (2006)
The preliminary hotwire measurements of the output jet from the fluidic device, during the calibration stage, have been visualised in Figure 2.68. The jet response shows similar oscillatory characteristics to that of the modulating flow coupled with a continuous jet flow. Culley _et al_ (2003) state that the absence of a complete shut off state (zero jet velocity) of the modulation is due to the fact that the flow, within the output plenum, does not empty in the time it takes the device to switch. The fluidic device has demonstrated capability of generating a frequency range of $1.8\text{kHz} \leq f \leq 2.1\text{kHz}$, which gives a reduced frequency number (based on injection location) range of $3.3 \leq F^* \leq 3.9$.

Culley _et al_ (2003) used total pressure loss coefficient as a figure of merit to quantify the performance of the embedded-fluidic-vane and compared it with other actuation methods. The experimental results demonstrated that the harmonically oscillating control jet is advantageous in a sense of its energy expenditure over that of a steady jet actuation. The steady jet control required a 56% increase in injected mass flow in order to achieve the same reduction gained by harmonically oscillating the jet. The performance of the embedded-fluidic-vane in separation control, at a
vane restagger angle of 5°, can be seen in Figure 2.69; this shows a change in the reduction of total exit pressure loss relative to the non-injection case.

Figure 2.69 Comparison of loss reduction for the slot-vane, embedded-fluidic-vane and hole-vane at 5 deg restagger angle, steady injection, 56% span, \( \phi = 0.36 \). Culley et al (2003)

Figure 2.69 shows that the control attempts made with injections at very low flow rates present rather an adverse effect, and loss reduction starts to reduce at an injection momentum coefficient higher than approximately 0.03. Culley et al (2003) state that actuation with a low injection flow rate simply adds more low momentum fluid to a region that is already at (or near) separation, which leads to enhanced viscous loss. Whereas, an excessive injection flow rate increases the mixing loss arising from an interaction between the injected jet and the freestream flow.

The embedded-fluidic-vane model successfully demonstrates its capability of reduction in total exit pressure loss of a low speed axial compressor. It realised an appreciable 20% reduction in loss compared to a non-control case with an injection flow rate of approximately 1% of the compressor through flow (velocity ratio of \(~ 1.2\)).

The investigated fluidic model, however, was found to be less effective in separation control when compared to other actuation methods examined during the study (as shown in Figure 2.69). Culley et al (2003) conclude that this is due to an existence of attenuation in the magnitude of harmonic oscillation generated by the fluidic device, which arises from the small exit holes in the vane surface. However, they further suggest possibility of an improved performance of the fluidic device
compared to the siren valve, when its output load matching is considered during the design stage of the model.

Raman and Raghu (2004) introduced a single output vent-fed bistable fluidic oscillator as a tool to suppress the cavity resonance tone arising from the freestream jet flow and cavity interaction. A schematic drawing of the miniature fluidic device used by Raman and Raghu (2004) is shown in Figure 2.70, together with a photo showing the instantaneous output jet oscillation visualised by using water as a working medium. One can note that the model generates a timely and spatially oscillating jet with a square waveform. Raman and Raghu (2004) state that the fluidic model is also capable of generating an oscillating jet with either a sawtooth or sine waveform, when the exit nozzle design is altered.

![Figure 2.70 Schematic showing design and operation of miniature fluidic device. a) schematic drawing of the model, and flow visualisation using water injection depicting square-wave behaviour of oscillatory jet exiting the nozzle, where supply pressure = 4psig. Raman and Raghu (2004)](image)

The fluidic oscillator produced an unsteady oscillatory jet in the frequency range of 592Hz to 2760Hz, with a supply pressure variation between 0.4psig and 40psig. An inherent frequency saturation initiated beyond the supply pressure of 15psig (flow rate of $0.75\times10^{-3}$kg/s).

The performance of the fluidic oscillator in the cavity resonance tone suppression has been examined by locating the exit nozzle (1.693mm X 0.954mm in dimensions) on the upstream end of the cavity base (depth of 1.27cm, width of 4.45cm and length of 7.62cm). This points perpendicular to the main flow over the cavity, and two different freestream Mach number cases of M=0.485 and M=0.69 are considered in the experiment.
The fluidic oscillator achieved noticeable resonance tone reduction in both freestream Mach number cases. The measurements of the cavity tone amplitude is shown in Figure 2.71, with the fluidic oscillator actuation levels at the main jet flow of \( M=0.69 \). Figure 2.71 clearly shows tone reduction of as much as 10dB with the supply mass flow rate of \( 1.15\times10^{-3}\text{kg/s} \), which is equivalent to 0.12% of the main jet flow. Raman and Raghu (2004) remark that the mass addition is a more dominant factor than the actuation frequency in jet-cavity tone suppression, noting the initiation of frequency saturation beyond the supply pressure of 15psi. Also, Raman and Raghu (2004) stress an efficiency of an oscillatory mass addition method, by stating that 1dB tone reduction is achieved when using steady jet addition at a flow rate of 0.12% of the main jet flow.

![Figure 2.71 Fluidic oscillator supply pressure and mass flow requirements for jet-cavity tone suppression.](image)

Cerretelli and Kirtley (2009) examined the implementation of a vent-fed bistable wall attachment fluidic oscillator as an actuator to control flow separation and then compared its efficiency with the steady blowing control method.

In addition, the effectiveness of the reduced frequency number (\( F^+ \)) has also been studied and, for this reason, two different fluidic oscillator designs are considered, namely: insert (A) and insert (B). Both share an identical basic planform of fluidic amplifier, but their feedback path geometry differs. This gives insert (A) a constant output jet frequency characteristic while the output frequency of insert (B) is dependent on the supply flow pressure. The general output jet responses and frequency responses of those fluidic oscillators can be seen in Figure 2.72 and Figure 2.73. The exit jet profile demonstrates a strong modulation with rms values ranging from 35 to 60% of the jet mean velocity. Insert (A) is designed to operate at
a constant reduced frequency number of ≈ 0.7, while insert (B) covers the range from \( F^* = 3 \) to \( F^* = 6 \). In addition, Cerretelli and Kirtley (2009) remark that due to an inherent leakage existing with any fluidic switch, a complete shut-off state (zero velocity) during the jet modulation can never be attained (Kirshner and Katz, 1975).

**Figure 2.72 Feedback fluidic oscillator in insert (A).** a) velocity wave form and b) frequency response as a function of supply pressure. Cerretelli and Kirtley (2009)

**Figure 2.73 Feedback fluidic oscillator in insert (B).** a) velocity wave form and b) frequency response as a function of supply pressure. Cerretelli and Kirtley (2009)

The fluidic oscillator models developed by Cerretelli and Kirtley (2009) are then used in boundary layer separation control over a hump diffuser model, as schematically shown in Figure 2.74 for \( U_{fs} = 25.9 m/s \) (\( Re = 3 \times 10^5 \), trip wire of \( D = 0.25 \text{mm} \) used upstream of the test section to ensure turbulent transition) and \( \alpha = 0^\circ \). The flow control inserts are situated between 50 and 62% chord lengths from the leading edge.
Cerretelli and Kirtley (2009) examined the overall pressure recovery coefficient, $C_p$, as a metric measure of the separation control performance and also presented PIV measurements of the jet interaction flow field. These results are illustrated in Figure 2.75 and Figure 2.76. A trend of change in $C_P$ with an increasing injection momentum coefficient, $C_{\mu}$, for all three control methods, show an initial drop in $C_P$ up to a value $C_{\mu} \approx 0.9$; then, it starts to increase steeply giving a relatively constant value of $C_P \approx 0.75$. The initial drops at low injection levels are supported by the presence of a larger re-circulating flow region in Figure 2.76 (b), when compared to Figure 2.76 (a). Cerretelli and Kirtley (2009) state this is due to an adverse perturbation introduced to the main flow by the low injection level control flow.

**Figure 2.74 Schematic drawing of the hump diffuser cross section.** Cerretelli and Kirtley (2009)

**Figure 2.75 Diffuser pressure recovery curves for steady and unsteady blowing as a function of $C_{\mu}$ for $\alpha=0$ deg.** Cerretelli and Kirtley (2009)
Also, Cerretelli and Kirtley (2009) demonstrated a superior performance of insert (A) over insert (B), and possible development of large-scale global instability triggered by coupling between unsteady actuation at low frequency ($F^+ \leq 1$) and the boundary layer has not been observed as in Figure 2.76.

To compare the efficiency between the unsteady fluidic oscillator and the steady blowing control methods, Cerretelli and Kirtley (2009) presented graphs (as in Figure 2.77) where the y-axis shows the relative reductions gained by the fluidic oscillator and the x-axis shows the pressure recovery coefficients. A possible reduction of 60% in the required momentum and a corresponding 30% reduction in required mass flow are achievable in attaining an overall pressure recovery coefficient of 0.75.
Complementary to Cerretelli and Kirtley (2009), Cerretelli et al (2010) further validated the performance of the fluidic oscillator, which has a similar output characteristic as that used for insert (A) in Cerretelli and Kirtley (2009) in flow separation control over a DU-96 aerofoil model (span of 0.73m and chord length of 0.8m). An array of fluidic oscillators with exit nozzle diameter of 3mm and interspacing of 5mm between the exits were used. This is then installed at 60% chord from the leading edge on the suction surface of the model with exit nozzle pitch and yaw angles of 30°.

The fluidic oscillator employed in the study provided a constant frequency response independent of the driving pressure. For the completeness of the review, the general jet response and frequency response are shown in Figure 2.78.

The validation work of the fluidic oscillator in separation control has been performed covering the Reynolds number (based on chord length) range of $2 \cdot 10^6 \leq Re \leq 4.8 \cdot 10^6$. Measurements of lift and drag have been analysed for two actuation levels.
based on total mass flow rate driving an array of fluidic oscillators, namely high refers to $\dot{m}=69.9$g/s and low refers to $\dot{m}=51.4$g/s.

Cerretelli et al (2010) successfully demonstrated improvements in lift and drag, especially in post stall applications for all Reynolds numbers and actuation levels studied. A considerable increase of 10 to 60% in lift, depending upon the operating conditions of the fluidic actuator, can be seen in Figure 2.79. Also, Cerretelli et al (2010) repeated a similar experiment but with an artificial roughness (7 bump tapes with 10mm spacing) applied to the leading edge of the previous aerofoil model. As Figure 2.79 shows, actuation of the fluidic oscillator with a rough surface model generally resulted in further enhancement of the lift. However, this is possibly due to an early transition of the boundary layer.

![Figure 2.79 Improvements in max CL as a function of Re and velocity ratio for all tested configurations.](image)

Figure 2.79 Improvements in max CL as a function of Re and velocity ratio for all tested configurations. High level of actuation refers to a mass flow of 69.9g/s, and low level refers to a mass flow of 51.4g/s. Cerretelli et al (2010)

Readers would have recognised the habitual dependency of the output jet frequency and velocity characteristics of fluidic devices on the supply flow conditions. This intrinsic characteristic can be disadvantageous as a potential separation control actuator due to the loss of controllability of actuation reduced frequency number. In this point of perspective, some research work have been made in combining the use of other flow control actuators, generating an impulsive output flow, with fluidic amplifier; this is based on the fact that fluidic amplifier can provide relatively large amplification of a minute level of flow.
Gregory et al (2009) have evaluated the piezo-fluidic oscillator in an effort to decouple the frequency response of the fluidic device from its input characteristic and to enhance the output momentum of the usual actuators driven by piezoelectric transducers. In this study, the oscillatory jet response characteristics to the piezoelectric bender oscillation have been evaluated at various actuation frequencies and supply pressures.

The piezo-fluidic oscillator consists of a simple control free bistable wall attachment fluidic device, but the usual splitter replaced by a piezoelectric bender which can deflect under an electric signal. The fundamental performance characteristics of the fluidic oscillator with a piezoelectric-bender idea have been evaluated previously in Gregory et al (2005). In the work of Gregory et al (2005), two different designs of piezo-fluidic oscillator have been evaluated, which includes one with a bender tip pointing the upstream of the power jet while the other bender points downstream. Through this this, upstream-pointing-bender found to provide superior modulation characteristics, including higher modulation index and frequency roll-off (Gregory et al, 2005).

The design of the piezo-fluidic oscillator with an upstream pointing piezoelectric bender model, in Gregory et al (2009), is shown in Figure 2.80, while Figure 2.81 shows hot-film measurements of the output jet at various actuation conditions.

![Diagram of the piezo-fluidic actuator design](image)

**Figure 2.80 Diagram of the piezo-fluidic actuator design.** The piezoelectric bender is positioned in the diffuser, pointing upstream into the flow. Gregory et al (2009)
Figure 2.81 Time history of the oscillator outputs simultaneously measured by hot-film probes. a) at pressure ratio of 1.69 and 10Hz, b) at pressure ratio of 1.69 and 200Hz, c) at pressure ratio of 1.14 and 1.0kHz, and d) at pressure ratio of 2.15 and 5Hz. Gregory et al (2009)

It is observed through increasing the driving frequency that a change in the shape of the output jet waveform occurs. Yet, an increase in driving frequency results in an increase in the phase delay from the control signal, with an increase in rise and decay times. At a low driving frequency range (0 ~ 250Hz), an oscillatory jet with a square waveform is generated with a modulation level higher than 80% (modulation index of 0.8); whereas, a high actuation frequency bounds, a waveform similar to a sawtooth signal is generated as in Figure 2.81 c). In addition, Figure 2.81 d) demonstrates the capability of the piezoelectric bender in controlling the power jet flow at sonic nozzle condition (P/Patm=2.15).

Gregory et al (2009) also examined the effect of the actuation pressure level on the flow signal response time to the step inputs sent to the piezoelectric bender. These results are shown in Figure 2.82 and one can note a decreasing response time with an increasing pressure ratio, i.e., a faster power jet flow. Gregory et al (2009) state that this is solely due to an increase in convection time within the output plenum.
Gregory et al (2009) further evaluated a bandwidth of the unsteady output jet for a chosen pressure ratio of 1.14; the results are illustrated in Figure 2.83. The output jet frequency shows one to one linear relation to the examined actuation frequency range of 0 to 1.2kHz. However, Gregory et al (2009) note a region between 250Hz and 500Hz, where the piezo bender appears unable to actuate the jet oscillation for the examined input flow condition. They state it is a possibility for the piezo bender to oscillate at its resonant frequency within this driving frequency range, after noting an energy shift in the power spectrum measurements to 121Hz as shown in Figure 2.83, which is a pre-defined resonant frequency of the piezoelectric bender.

In addition to the piezo-fluidic oscillator, Gregory et al (2007) introduced a single output plasma-fluidic oscillator by integrating a plasma actuator within a control port of a usual bistable wall attachment fluidic amplifier. The conceptual drawing of the model is shown in Figure 2.84. The work of Gregory et al (2007) focused on the
verification of the switching characteristics at various flow velocity (equivalent to an exit velocity range of $5 \sim 8\text{ms}^{-1}$) and electric power supply levels (at 345mW and 400mW). They also evaluated the jet profile across the output port of the plasma-fluidic oscillator during the jet switching process.

![Diagram of the plasma-fluidic actuator](image)

**Figure 2.84** *Diagram of the plasma-fluidic actuator.* Gregory et al (2007)

Figure 2.85 shows the hotfilm measurements of the output jet profile across the output plane during the jet switching process. As can be seen from Figure 2.85, the first jet profile graph includes the measurements made at a time 20ms after the plasma actuator initiation. Gregory et al (2007) state that prior to this time an overall change in output jet flow has not been detected and suggest the time taken for a separation bubble to grow along the attachment wall is the dominant source of this delay. Figure 2.85 also shows changes in peak velocity and the actual jet profile, as the output jet traverses across the exit nozzle. By comparing the jet profiles at t=20 and t=25, an increase in peak velocity and total momentum can be recognised. Gregory *et al* (2007) concluded that this increase is as a result of an added control flow from the plasma actuator to the main jet flow.

![Velocity measurements across nozzle opening](image)

**Figure 2.85** *Velocity measurements across nozzle opening, Pplenum = 0.15 Torr, Pdissipated = 400mW.* Gregory et al (2007)
Figure 2.86 shows changes in jet switching time (the time taken for the output jet flow to reach 63.2% or one time constant of its final value) with variations in the flow speed and plasma actuation power level. An increase in both the switching time and variance of the calculated switching time can be seen with an increase in flow velocity and/or with a decrease in power level. Gregory et al (2007) state that since the momentum of the induced control flow by the plasma actuator is limited by the available electric power to it, the control authority of the plasma actuator on the min-jet switching reduces with increasing flow velocity and vice versa.

![Figure 2.86](image)

**Figure 2.86 Plots of switching time for varied flow velocities and power settings.** Gregory et al (2007)

Although the results in Gregory et al (2007) present some drawbacks in generating a confident level of control momentum with the plasma actuator and physical integration of the plasma electrodes (a limitation in power levels to the electrodes to prevent them from induced arcing across the electrodes of the confined configuration), it successfully demonstrates a functioning plasma-fluidic oscillator. Yet, Gregory et al (2007) claim that the performance of the plasma fluidic oscillator could certainly be improved with careful review of the geometrical optimisation of the fluidic amplifier and the plasma actuator location, together with the development of a more efficient plasma actuator.

As a closure of this section, the work of Tesar et al (2006) is reviewed in details as this covers the fundamental study of the test model evaluated in this thesis. Readers would have noted that most of recent research efforts are focused on studying vent-fed fluidic oscillator. To the author’s best knowledge, the work of
Tesar et al (2006) is almost an only recent efforts devoted in investigating a control-loaded fluidic oscillator.

The planform of the fluidic model studied in Tesar et al (2006) follows geometrical specifications shown in Figure 2.87 a) with a height of 6mm; photo of a built model can be found in Figure 2.87 b). A distinguishable feature of the model is that it is a pressure driven oscillator unlike others reviewed above, which adopted momentum interaction as a principal in achieving the jet oscillation. Tesar et al (2006) evaluated the performance of the model both as an amplifier and oscillator.

![Diagram and Photo](image)

**Figure 2.87 Geometrical specification and photo of the model.** a) geometry of the amplifier, and b) photograph of the assembled amplifier as used in the experimental investigations. Tesar et al (2006)

They have numerically evaluated the performance of the model as an amplifier and the results are illustrated in Figure 2.88. Tesar et al (2006) conclude approximately 7% of the supply flow will initiate the jet switching when injected through the attached side control port. Also, the computed pathlines of the internal flow as shown in Figure 2.89 illustrate an existence of a standing vortex resulting a jet pumping effect through the low state side output port; approximately 20% of the supply flow.
Tesar et al (2006) studied the frequency bandwidth of the fluidic oscillator with control parameters of feedback loop length ($1m \leq l \leq 52m$) and diameter ($2.5mm \leq d \leq 10mm$) for supply flow Reynolds number (based on power jet nozzle width) range of 1790 to 14340. General output jet response of the fluidic oscillator measured with a pressure transducer is shown in Figure 2.90. One can notice a generation of a very stable modulation with square waveform.
Plot of frequency as a function of feedback loop length at various supply flow level can be found in Figure 2.91. One can note the inverse proportionally reducing jet frequency with increasing loop length. Figure 2.92 shows corresponding changes in the calculated Strouhal number (based on feedback loop length, l, and power nozzle jet velocity, w) as a function of Reynolds number. The term Sh shown in the y-axis of Figure 2.92 is defined in Eq. 2-18 as in Tesar et al (2006).

Figure 2.91 The frequency vs. loop length dependences obtained with the 10mm diameter loop tube at different supply flow rates. Tesar et al (2006)
Figure 2.92 The experimental results obtained with the 10mm diameter loop tube at different flow rates re-plotted in terms of the Strouhal number as a function of Reynolds number. Tesar et al (2006)

\[ Sh = \frac{fb}{w} \quad \text{Eq. 2-18} \]

Tesar et al (2006) multiplied the Strouhal number by the factor of two to enable the interpolation as in Eq. 2-19 and they remark this did not affect the behaviour of the Strouhal number in any way. Based on this derived relationship together with recognition of the exponent of the curve fitting shown in Figure 2.92 being near 1 they conclude the propagation speed is constant for Reynolds number higher than \( \sim 3500 \). In addition, for this Reynolds number regime, they state the Strouhal number is independent of the loop length for the length greater than 5m.

\[ \frac{2Shl}{b} = \frac{wa}{w} \quad \text{Eq. 2-19} \]

The effect of feedback loop diameter on the propagation speed has also been investigated and the results are shown in Figure 2.93. Tesar et al (2006) state the transition to the constant propagation speed regime is noticeable, but for a thin feedback loop a considerable reduction propagation speed is recognised and remark this is due to the thermodynamic characteristic change of fluid within the loop arising from the confinement by the small diameter; this is supported by the results in Figure 2.94.
2.8 Summary of fluidics

Fundamental principals of operation and classification of fluidic devices are reviewed throughout this chapter. Geometrical influences on both the static and dynamic characteristics are well understood and steady-state design methodology is provided.
Recently, considerable research efforts have been devoted in validating a feasibility of fluidic oscillator as an alternative PVGJ actuator for flow control appreciating its operational simplicity and robustness in generation of unsteady pulsing jet. Although, these efforts resulted in notable findings; successful demonstration of separation control; specifying the optimal reduced frequency number for operation; successful development of functioning micro- to meso-scale oscillator giving frequency bandwidth of several kilo hertz; effective removal of habitual coupling between input and output characteristics, most of studies are focused on investigating momentum driven oscillators.

To the author’s best knowledge, publication made by Tesar et al (2006) is almost an only recent work suggesting the use of control-loaded fluidic oscillator for flow control application. Although, Tesar et al (2006) provide some valuable results it requires further analysis as to validate it feasibility.

Thereby, the author experimentally evaluates steady-state characteristics of the oscillator in amplifier configuration and output jet characteristics to verify its frequency bandwidth and governing parameter for jet switching.
3 Experimental apparatus and methods

3.1 Experimental models

The design of the bistable fluidic oscillator and an array of fluidic amplifiers studied in this report are provided by Prof. Vaclav Tesar from the Academy of Sciences of the Czech Republic.

3.1.1 The fluidic oscillator

3.1.1.1 The design

Figure 3.1 and Figure 3.2 show the geometry of the planform and the interaction region of the model respectively. The fluidic oscillator has a power jet nozzle with a width of 2.0mm and two control nozzles with a width of 1.4mm. It has a depth of 8.0mm giving a power nozzle aspect ratio of 4. A specially designed concave bi-cusped shaped splitter is employed to give an extra latching stability and is located at 11.63mm downstream of the exit plane of the power nozzle. The planform of the device is manufactured by laser cutting four 2.0mm thick perspex sheets, then piling them up to meet the required height. The manufacturing of the oscillator is then completed by covering it with a top and a bottom plate each having a thickness of 8.0mm.

All the input and output interconnection ports are made accessible by connecting metal tubes oriented perpendicularly to the top cover plate. The tubes stand 20mm above the upper surface of the top cover plate and have an internal orifice diameter of 6mm, 5mm and 6mm for the supply, control and exit ports respectively. Figure 3.3 shows an assembly of the manufactured bench-top test model. The principle of operations is described in Chapter 2. The time-delay feedback loop internal diameter was fixed to the size of 5mm, while its impedance and capacitance varied according to its length.

This model was tested so as to assess its steady-state dynamic performance such as the control flow requirement in switching the power jet flow, the flow gain and the fan-out ratio as a function of the power jet Reynolds number, $Re_{D_s}$. The characteristics of the oscillatory jets, including the pulse transition, frequency bandwidth and characteristic switching time are also examined.
Figure 3.1 Geometrical specification of the fluidic oscillator planform.
(Dimensions in mm)

Figure 3.2 Geometrical specification of the fluidic oscillator interaction region.
(Dimensions in mm)

Figure 3.3 Photos of the fluidic oscillator test model.
3.1.1.2 Test conditions

3.1.1.2.1 Steady-state analysis
This analysis has been performed to define the control flow rate requirement for deflecting the power jet from the attached side to the opposite wall, gain and fan-out ratio of the actuator as a function of the power jet Reynolds number. The power jet Reynolds number is based on the power nozzle hydraulic diameter and it varies from $8.5 \times 10^3$ to $6.5 \times 10^4$ equivalent to a supply mass flow rate range of 0.7 to 6g/s.

For this analysis, the feedback loop from the existing fluidic oscillator is removed to allow the device to function as a fluidic amplifier, which reacts to an application of a control signal fed directly into the control port. An air supply connection is made to one of the control ports, while the other is open to atmospheric pressure. The volume flow rates applied to both the supply and the control are measured with an analogue flow meter. A precision flow regulation is achieved by the use of needle valves.

A schematic drawing of the experimental set-up used for the steady-state analysis is given in Figure 3.4. To calculate the mass flow rates going through the supply port and control port, fluid density is required along with the measured volume flow rates. The volume flow rates are measured using flow meters. The locations at which the static pressure reading for fluid density calculation are different for the supply port and the control port, as shown in Figure 3.4. For a flowmeter measuring low volume flow rates it is recommended to read a fluid density upstream of the flowmeter, which was the case for the control port flow. On the other hand, the fluid density of the flow going into a supply port is measured at the output port of the volume flowmeter, which was designed to measure high flow rates.

The jet mean velocity is estimated from the hotwire measurements of the exit velocity taken at $1D_o$ downstream of the centre of the output orifice (within the potential core region). If the exit jet Reynolds number based on the exit orifice diameter and jet mean velocity, $Re_{D_o}$, is less than $2.3 \times 10^3$, the flow can be assumed to be laminar. In this case, the Hagen-Poiseuille velocity profile as shown in Eq.3-1 can be assumed and the jet mean velocity, $\bar{u}_m$, is equal to one half of the maximum velocity at the jet centreline.
\[ u_r = u_c \left[ 1 - \frac{r^2}{R^2} \right] \] 

Eq. 3-1

At this low Reynolds number, however, the velocity measured with the hot-wire \( \overline{u}_{c,o} \) is not truly the centreline velocity but rather a space-averaged value over the wire length of the hotwire probe (length of hotwire being 1.25mm). Nevertheless, the measured velocity is found to be only 2.9% lower than the actual centreline velocity if a symmetric jet profile across the orifice and simplified linear mean velocity variation from the wire tips to the centre are assumed. Therefore, the jet mean velocity can be still reasonably taken as one half of the velocity measured with the hotwire. For higher \( Re_{D_o} \) values, since the exit flow is a fully developed turbulent with a top-hat velocity profile, the measured \( \overline{u}_{c,o} \) is assumed to be equal to \( \overline{u}_{m,o} \).

\[ \begin{align*}
\text{at a given supply flow rate, the control flow is gradually increased until the power jet} \\
\text{has completely detached from the originally attached wall side. At the point of switching}} \\
\text{the needle valve for the control flow is locked to a position to maintain its}} \\
\text{flow rate. Then simultaneous measurements of pressure readings and the hotwire}} \\
\text{data at both output ports are made. The data are stored on a PC along with the}} \\
\text{flowmeter readings taken manually for future analyses. Once the data acquisition}
\end{align*} \]
has been completed the power jet is forced back to its initial state, i.e. to the closed control port side.

For the data acquisition from the hotwire, the sampling frequency was set to 80kHz and hotwire signal was low-pass filtered at 50kHz for supply flow rate higher than 2.1g/s ($\bar{\nu}_{c,\rho} \geq 30ms^{-1}$) and 40kHz and 20kHz for lower than 2.1g/s ($\bar{\nu}_{c,\rho} < 30ms^{-1}$) respectively so as to capture the possible turbulent fluctuations. The pressure reading was filtered at 100Hz using a digital filter. The total sampling time was given as five seconds.

### 3.1.1.2.2 Oscillatory jet characteristic analysis

The purposes of this analysis are two-fold: Firstly, to define quantitatively the effects of the flow rate and the length of the time delay feedback loop on the frequency response of the periodic oscillation of the jet; Secondly, to verify the characteristic switching time of the actuator that is determined by the geometrical parameters of the actuator. In addition, at certain selected test points, jet characteristics are also analysed, including the phase-averaged and time-averaged jet centreline velocity, pulse transition characteristics and turbulence intensity over a one cycle.

In the present experiment, the time delay feedback loop between the two control ports is obtained with the use of a plastic tube with a bore size of 5mm with a length varying from 0.3m to 2.0m. The supply flow rates range from 0.6g/s to 5g/s, which gives a range of Reynolds number based on the hydraulic diameter of the tube of $6.5 \times 10^3$ to $7.3 \times 10^4$. The supply flow rate is regulated with a needle valve and measured with an orifice plate flowmeter. 오류! 참조 원본을 찾을 수 없습니다. shows a summary of the test cases and the general experimental set-up is illustrated in Figure 3.5.

<table>
<thead>
<tr>
<th>Supply Flow Rate (g/s)</th>
<th>Feedback Loop Length (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6 to ~5</td>
<td>0.3</td>
</tr>
<tr>
<td>(Power jet velocity from ~ 30ms$^{-1}$ to ~170ms$^{-1}$ with an incremental step of approximately 5ms$^{-1}$)</td>
<td>0.58</td>
</tr>
<tr>
<td>Power nozzle $M \approx 0.5$</td>
<td>0.78</td>
</tr>
<tr>
<td></td>
<td>1.0</td>
</tr>
<tr>
<td></td>
<td>1.16</td>
</tr>
<tr>
<td></td>
<td>1.58</td>
</tr>
<tr>
<td></td>
<td>1.78</td>
</tr>
</tbody>
</table>
Table 3.1 Experiment case summary for the study of the jet oscillation characteristic of a fluidic oscillator.

![Figure 3.5](image)

Figure 3.5 Experimental set-up for the study of the jet oscillation characteristic of a fluidic oscillator.

The jet oscillating frequency is determined as the strongest peak in the power spectrum of the voltage signals measured with a hotwire placed at 1D₀ downstream of the orifice. In order to improve the quality of the spectral measurement, the range of frequencies to be captured was predetermined by estimating the possible resonant frequency of the device with feedback loop length and local speed of sound based on results in Tesar et al (2006); this effectively defined the maximum possible frequency to be ~300Hz for the range of feedback loop length studied in this thesis. Also, oversampled data prior to the final measurements are used to enhance the assurance in determination of the sampling frequency. The final data were acquired at the rate of 3kHz for 2048 points giving a frequency resolution of approximately 1.46Hz. The signals were then low-pass filtered close to the at 1.5kHz mechanically and filtered further with a digital low-pass filter at 1kHz to capture the possible high frequency harmonics. Data are stored for further analysis only when a stabilised spectral measurements are visualised in LabVIEW; each set of 2048 data points were fed to fft function in LabVIEW, then it averaged multiple sets over the time. The pressure measurements for the flowmeter were filtered at 100Hz with a digital low-pass filter.
The sampling frequency selected for the jet velocity analysis data acquisition was 80kHz with a low-pass filter set at 50kHz. The total sampling time was twelve seconds.

### 3.1.2 Fluidic amplifiers in an array

#### 3.1.2.1 The design

The planform of the fluidic amplifier in the array shares the same design of the interaction region as the single oscillator discussed the previous section. Hence it has a power nozzle width of 2mm and a height of 8mm. However, the design of the output ports has been modified so that the spacing between adjacent output orifices can be reduced. The entire array contains five fluidic amplifiers equally spaced across the span. The amplifiers are spaced at 73.96mm in the array and each provide six output ports with a spanwise spacing of 12.33mm with an orifice diameter of 7.6mm as shown in Figure 3.56. The locations of the exit ports of the amplifier are indicated by a cross in the shaded flow path. The two crosses in the empty space represent the exit ports contributed by the neighbouring amplifiers. A streamwise offset between the two control ports is also introduced to allow the control flow to be supplied from two separate chambers.

![Figure 3.6 Geometrical specifications of a fluidic amplifier in the array](image)

**Figure 3.6 Geometrical specifications of a fluidic amplifier in the array**

*Dimensions in mm* (Tesar 2007)
The assembly of fluidic amplifier array is made of four separate pieces; the core plate containing the array of the fluid amplifiers, the middle plate containing the control chambers and jet orifices, the main supply chamber block and a 2mm thick aluminium cover plate (see Figure 3.7). The core plate and the middle plate were manufactured by CNC machining a 9mm and a 14.5mm thick aluminium plate at the Manufacturing and Laser Processing group at the University of Manchester. The two control chambers have a dimension of 24.5mmx12.5mmx556mm. The main supply chamber with a dimension of 22mmx38mmx558mm was manufactured as a separate module and clamped on to the middle plate. Pictures of the manufactured core plate containing an array of fluidic amplifiers and other associated parts can be found in Figure 3.8.

![Figure 3.7 3D drawing showing the general assembly of the test model. Tesar (2007)](image)

### 3.1.2.2 Test conditions
The assembly of the fluidic amplifier array is designed such that it can be mounted as an insert in a 2D NACA0012 aerofoil section which has a chord length of 250mm and thickness of 30mm. The exit orifice diameter is 1.1mm on the aerofoil surface. The orifice diameter of the exit port of the fluid amplifier is reduced from 7.6mm to 1.1mm through a divergent channel of 8mm in length inside the aerofoil.
Hence the initial intention of this study was to examine the oscillatory jet performance of the array of fluid amplifiers with a final exit orifice diameter of 1.1mm. However, owing to the introduction of the extra exit load on the fluid amplifiers, which is caused by the reduction of the exit diameter from 7.6mm to 1.1mm at the outputs of the fluidic amplifiers, a fully pulsed exit jet response was not attainable. This effectively shifted the focus of this project to finding for a minimum possible final exit diameter size which enables a fully pulsed jet for the existing design and modifying this design so as to reduce the exit diameter while meeting the desired fully pulsed jet response. For this reason, six exit nozzles and are attached to the exit orifices on a fluidic amplifier via a mounting plate as shown in Figure 3.8.

![Figure 3.8 Geometrical specification of the exit nozzles for a single fluidic amplifier](image)

The modification on design which is considered feasible in the present study is the aspect ratio of the power nozzle in the fluidic amplifier. This is because the cross-sectional area ratio of the exit orifice to amplifier exit conduit also determines the oscillatory jet amplitude attenuation.

The oscillatory jet characteristics were studied for various sizes of exit orifice diameter ranging from 1.1mm to 3.5mm with a power nozzle aspect ratio varying from 2 to 4. A summary of the test points is shown in Figure 3.8. The performance parameters considered were the jet peak velocity and the frequency response. Therefore similar data sampling setting as that of the oscillatory jet characterisation study for the single fluidic oscillator case was adopted.

In the present study, the jet switching in the fluid amplifier array is governed by an impulsive control signal fed into the control chambers. The fluidic oscillator studied

---

*Figure 3.8 Geometrical specification of the exit nozzles for a single fluidic amplifier* a) exit nozzle profile; b) 3D view of an exit nozzle; c) assembled exit nozzle plate section; d) the fluidic array with the exit nozzles in place.
previously was used as the driving actuator for the array with each of its outputs connected to the control chamber via a fixed feedback loop length of 1.16m. The supply flow level to the fluidic amplifiers and a fluidic oscillator were measured and regulated separately, as illustrated in Figure 3.9.

<table>
<thead>
<tr>
<th>Feedback loop length of fluidic oscillator (L) (m)</th>
<th>AR</th>
<th>Exit orifice diameter (D₀) (mm) [start : step size : end]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.16</td>
<td>4</td>
<td>[1.1 : 0.1 : 3.6]</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>[1.1 : 0.2 : 3.0]</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>[1.1 : 0.2 : 2.6]</td>
</tr>
</tbody>
</table>

Table 3.2 Summary of the test cases for the study of the fluidic amplifier.

Figure 3.9 Experimental set-up for the cascading study of a fluidic amplifier.

3.2 Experimental methods

3.2.1 Constant temperature anemometer (CTA)

Throughout the experiment reported in this report, the fluid flow velocity measurements at the outputs of the actuators were accomplished with the aid of a constant temperature hotwire anemometer. This means of fluid diagnostic method was chosen for its simplicity, reliable accuracy and high frequency response.
The hotwire probe was a standard DANTEC 55P11 single normal wire probe. The sensor was a 5μm in diameter and 1.25mm long tungsten wire. The probe was connected to a TSI IFA 300 constant temperature anemometer containing an on-board signal conditioning unit of the SMARTTUNE™ module and a thermocouple circuit. The SMARTTUNE™ module also automatically optimises the frequency response of the IFA 300, which can be as high as 250 kHz or greater. The analogue output from the unit was converted into a digital signal by a 16-bit PCI 6221 analogue-to-digital converter, which has the best resolution of 0.00153% at the ±5V scale and maximum sampling rate of 250kS/s and stored in a PC for future analysis.

3.2.1.1 Principles of operation

A constant temperature anemometer allows virtually instantaneous measurements of velocity fluctuations in the flow at a given point by quantifying the change as an analogue voltage which varies in a predictable manner at the anemometer output. This fluid diagnostic method is fundamentally based on the forced convective heat transfer from an electrically heated sensor (wire/film) to the surrounding fluid as their rate of exchange varies with fluid velocity, temperature, pressure and phase (density). Thereby if multiple changes of parameters occur simultaneously relevant corrections must be made. In this report only the fluid temperature correction will be considered in the error analysis section. One should also remember the existence of other heat transfer mechanisms apart from the forced-convection which includes conductive heat transfer from the sensor to prongs, radiation heat transfer and buoyant convection.

In the constant temperature mode, the hotwire placed in a feedback circuit as illustrated in Figure 3.10 forms one of the Wheatstone bridge resistors. In this mode, the chosen sensor hot temperature is maintained at a constant value, and hence is the hot resistance. When the wire is exposed to a varying flow condition the selected wire temperature will alter correspondingly due to forced-convective heat transfer. This then generates error voltage of $e_2 - e_1$ as in Figure 3.10, which is compensated by feeding the required current change back to the circuit to maintain the sensor temperature and the resistance at the same level. This results in a change in the voltage output as expressed in Eq. 3-5
The mean wire (sensor) temperature, $T_W$, can be related to the electrical resistance of the hotwire, $R_W$, as in Eq. 3-2

$$R_W = R_a \left[ 1 + \alpha_a (T_W - T_a) \right] \quad \text{Eq. 3-2}$$

$$\alpha_a = \frac{R_0}{R_a} \alpha_0 \quad \text{Eq. 3-3}$$

where the temperature coefficient of the wire material, $\alpha_a$, and wire resistance $R_a$ are the values at the ambient condition while $\alpha_0$ and $R_0$ stand for those values at $0^\circ C$.

A simplified form of a heat transfer relationship for a finite length active hotwire sensor can be expressed as in Eq. 3-4.

$$\frac{I^2 R_W}{R_W - R_a} = A + B U^n \quad \text{Eq. 3-4}$$

Combining the Eq. 3-2 and Eq. 3-3 and substituting into Eq. 3-4 for $R_W - R_a$ and introducing the wire voltage $E_W = IR_W$, where $I$ is the current, yield the heat transfer relation of the hotwire in Eq. 3-5. In practice the Eq. 3-5 is further simplified as in Eq. 3-6 by inclusion of the wire property constants and operation setting constants in the calibration constants $A$ and $B$, which can be easily found by performing a simple calibration analysis. This relationship is the well known King’s law. The exponent, $n$, is a user-selected variable which commonly ranges between 0.4 and 0.45.

$$E_W^2 = \left( IR_W \right)^2 = R_W R_0 \alpha_0 (T_W - T_a) \left( A + B U^n \right) \quad \text{Eq. 3-5}$$

$$E^2 = A + B U^n \quad \text{Eq. 3-6}$$
3.2.1.2 Calibration

For the calibration of the hotwire probe, a 55D41 calibration unit from DANTEC was used by placing the probe at the centre and in the same axial plane of the downstream pressure tap of the venturi nozzle. The calibration unit is capable of giving an accurate velocity variation range of \(\sim 2\text{ms}^{-1}\) to \(\sim 120\text{ms}^{-1}\) and this available full velocity span is used for the probe calibration. The velocity of the flow going through the venturi nozzle is calculated from a differential pressure measurement between the upstream and downstream of the venturi nozzle, made available with a digital micromanometer.

King’s law is used to relate the measured velocity and anemometer output voltage. In order to choose an optimal value for \(n\) in King’s law, the sum of error squared (SES) value is examined by comparing the measured output voltage with calculated output voltage for various \(n\) values ranging from 0.4 to 0.45. A minimum SES is found at \(n=0.43\).

In finding the calibration constants as in Figure 3.11, the voltage reading at zero velocity, \(E_{W,0}\), has been omitted since this reading is purely the result of the buoyant convection rather than a forced convection, but this can be included by multiplying the \(E_{W,0}^2\) by a correction factor of 0.8 as in Eq. 3-7 if required (Bruun 1995).

\[
A \approx 0.8 \cdot E_{W,0}^2 \\
\text{Eq. 3-7}
\]

To account for the possible characteristic disparity between low and high flow speed regions, two different relations are made producing separate expressions for flow below and above \(\sim 30\text{ms}^{-1}\). The calibration data are acquired at every \(\sim 2\text{ms}^{-1}\) up to \(30\text{ms}^{-1}\) then every \(\sim 4\text{ms}^{-1}\) for the high-speed region. The hotwire calibration data are acquired at a sampling frequency of 80kHz for total sampling time of 5seconds. They are then lowpass filtered at 50kHz for high-speed band and 20kHz for low-speed band. A typical calibration curve is shown in Figure 3.11.
3.2.1.3 Error analysis

The major error sources associated with the calculated velocity arise from the calibration unit, linearisation during the calibration and data conversion through the A/D converter. For the uncertainty estimation of the calibration unit used in this project which is based on the differential pressure reading method, the usual relative standard uncertainty value of 2% has been used with normal distribution assumption. The error associated with the linearisation, $\Delta U_{lin}$, is estimated to be 1.25% using Eq. 3-8, where $U_R$ is the measured velocity and $U_C$ is the calculated velocity, and has a normal distribution. The velocity sensitivity of the employed King’s law relation is deduced from Eq. 3-9 and is used in calculating the resolution error, $\Delta U_{res}$, together with the aforementioned A/D board input range, $E_{AD}$, and its data conversion resolution in bit, $n_{AD}$, as in Eq. 3-10. These results are shown in Figure 3.12 and Figure 3.13 and the maximum value reads about 0.23%, which turns out to be negligible. For other associated errors, such as sensor and fluid temperature fluctuation and ambient pressure variation, usual values from Bruun (1995) and Jørgense (2002) are assumed and listed as in 오류! 참조 원본을 찾을 수 없습니다. with the corresponding coverage factors.

\[
U(\Delta U_{lin}) = 100 \cdot \left[ \frac{1}{N} \sum_{i=1}^{N} \left( 1 - \frac{U_R}{U_C} \right)^2 \right]^{\frac{1}{2}}
\]

\[
S_v = \frac{dE}{dU} = \frac{nBU^{n-1}U}{2E}
\]

\[
U(\Delta U_{res}) = 100 \cdot \frac{E_{AD}}{2^{n_{AD}} \cdot dE} \frac{dU}{dE}
\]
Figure 3.12 An example of the velocity sensitivity calculated for the case shown in Figure 3.11

Figure 3.13 An example of the relative standard uncertainty due to the resolution error for the case shown in Figure 3.12
<table>
<thead>
<tr>
<th>Source of error</th>
<th>Estimated standard uncertainty (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Calibrator</td>
<td>1.0</td>
</tr>
<tr>
<td>Linearisation</td>
<td>1.25</td>
</tr>
<tr>
<td>Resolution</td>
<td>≈ 0.23</td>
</tr>
<tr>
<td>Sensor temperature fluctuation (Δ1°)</td>
<td>≈ 0.8</td>
</tr>
<tr>
<td>Fluid temperature fluctuation (Δ1°)</td>
<td>≈ 0.2</td>
</tr>
<tr>
<td>Ambient pressure variation (Δ10kPa)</td>
<td>≈ 0.6</td>
</tr>
</tbody>
</table>

Table 3.3 Summary of the estimated maximum errors in the velocity measurements through hotwire measurements with calibration graph shown in Figure 3.11.

These values are then used in finding the total expanded uncertainty with a normal distribution assumption giving the 95% confidence level using Eq. 3-11. The maximum value is found to be approximately equal to 3.8%.

\[ U_{\text{tot}} = 2 \left( \sum_{i} U(\Delta U_i)^2 \right) \]

\text{Eq. 3-11}

### 3.2.2 Orifice plate flowmeter

Johnston (1963) stated that ‘for a fluidic amplifier having an orientation such as the supply and control flows are injected perpendicularly to the planar plane of the device, the primary switching mechanism is governed by the mass flow rate rather than the pressure’. Hence an accurate means of measuring and regulating the flow rate is required for fluidic devices. A homemade orifice plate was manufactured with a conduit diameter, \( D_P \), of 25mm, which is the minimum allowable size according to British Standards, an orifice diameter, \( d_{or} \), of 12.5mm and an orifice plate thickness, \( E_{or} \), of 0.022\( D_P \). Owing to a difference in size between \( D_P \) and the diameter of the predefined interconnection ports of the test models, it requires a concentric expander at the inlet of the upstream pipe and a reducer at the exit of the downstream pipe. These have an expansion ratio of 4 and a reduction ratio of 0.25. An upstream and downstream straight passage length of 20\( D_P \) and 10\( D_P \) are thus introduced respectively for the flow development. The general geometries are given in Figure 3.14 below.
The aforementioned geometrical values in the design and the flow estimation procedures of the orifice plate are referred to the British Standards (ISO 5167-1 and 2 and ISO 15377). There is a minor departure in the locations of the upstream and downstream pressure tappings from the specifications given in the ISO 15377, which recommends the use of corner tappings, for the pipe $D_p$ ranging from 25mm to 50mm. Owing, however, to the manufacturing complexity, $D_p$ and $D_p/2$ tappings are used. This additional error has been corrected in accordance with ISO 12767. The orifice plate is not bevelled owing to the difficulty in controlling its depth, and so a straight bore orifice was used like that of a bidirectional orifice plate flowmeter.

### 3.2.2.1 Principles of operation

This fluid flow measurements technique is based on the well known Bernoulli's principle. As the fluid is forced out through a relatively small orifice, its velocity and consequently its pressure change. The maximum change occurs at a point referenced as a vena contracta hence giving the maximum pressure difference from the upstream of the orifice plate. This differential pressure reading, $\Delta P$, can then be used in approximating the flow rate, $q_m$, using Eq. 3-12.

$$q_m = C_d \frac{\pi}{4} d_w^2 \frac{\sqrt{2\Delta P_d}}{\sqrt{1-\beta^4}}$$

**Eq. 3-12**
where $\rho_1$ is the upstream fluid density, $\varepsilon$ is the running fluid expansion factor, $\beta$ is the orifice to pipe conduit diameter ratio and $C$ is the discharge coefficient which can be calculated using the Reader-Harris/Gallagher equation as in Eq. 3-13:

$$C = 0.5961 + 0.0261\beta^2 - 0.216\beta^3 + 0.00052(10^6 \frac{\beta}{\text{Re}_{\text{cp}}})^{0.3} + (0.0188 + 0.00634)\beta^{1.3}$$

$$+ (0.043 + 0.080e^{-0.08i} - 0.123e^{-0.12i}(1 - 0.114)}\frac{\beta^2}{1 - \beta^2} - 0.031\left(M_1' - 0.8M_1''\right)\beta^{0.3}$$

**Eq. 3-13**

where;

- $e$ - Orifice thickness (for current design, this equals $E_{or}$)

$$\text{Re}_{D_p} = \frac{4\rho_m\dot{m}}{\pi D_p \mu_l}$$

$$L_1 = \frac{i_1}{D_p}$$

$$L_2' = \frac{i_2}{D_p}$$

$$M_2' = \frac{2L_2'}{D_p}$$

$$A = \left(\frac{19000}{\text{Re}_{D_p}}\right)^{0.8}$$

It can be seen from the Eq. 3-13 that the discharge coefficient depends on the Reynolds number and it therefore needs to be iteratively optimised. This is done by grouping the dependent terms and independent terms in the Eq. 3-13 separately then setting dependent terms to zero by making an initial guess of the Reynolds number as infinite. The new value of the Reynolds number is then determined by substituting the approximated discharge coefficient from the previously estimated value of the Reynolds number into the Eq. 3-14. This iterative analysis is repeated until the convergence criteria in Eq. 3-15 satisfied.

$$\text{Re}_{D_p} = C \frac{\frac{\lambda_l^3}{\mu_l D_p}}{\sqrt{\frac{2\Delta P \rho_l}{\mu_l D_p}}} = C \cdot A$$

**Eq. 3-14**

$$\left| A - \frac{\left(\text{Re}_{D_p,0} / C_{1.0}\right)}{A} \right| \leq 1 \times 10^{-4}$$

**Eq. 3-15**
3.2.2.2 Calibration

As the flow estimation requires iterative approximation of the discharge coefficient, it was calibrated under the same condition as that of the test condition prior to the experiments, enabling the reduction in the computational time during data acquisition. Flow rate during the calibration is estimated with Eq. 3-12 using the iteratively estimated discharge coefficient. Upstream pressure measurements with the downstream temperature measurements are used in determining the density and viscosity. The temperature measurements are made at the downstream of the orifice plate to avoid the possible disturbance that a thermocouple pocket may cause and assuming that the temperature change from that of the upstream is negligible. The flow rate is then plotted against the $\sqrt{\Delta P \cdot \rho}$ as shown in Figure 3.15. The derived equation as in Eq. 3-16 by curve fitting the data points is then used in estimating the flow rate from the pressure measurements during the experiments.

$$q_m = 117.15 \cdot (\Delta P \cdot \rho)^{0.5} + 42.3 \quad \text{Eq. 3-16}$$

![Figure 3.15 The mass flow rate calibration graph of the homemade orifice plate.](image)

3.2.2.3 Error analysis

The expanded relative uncertainty in the mass flow rate measured with an orifice plate flowmeter can be approximated using Eq. 3-17 to achieve the usual 95% confidence level.

$$Q_{\text{uncal}} = 2 \left[ \left( \frac{\delta C}{C} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 + \frac{2}{1 - \beta^2} \left( \frac{\delta D}{D_r} \right)^2 + \frac{2}{1 - \beta^2} \left( \frac{\delta L}{L} \right)^2 + \frac{1}{4} \left( \frac{\delta \Delta P}{\Delta P} \right)^2 + \left( \frac{\delta \rho}{\rho} \right)^2 \right]^{1/2} \quad \text{Eq. 3-17}$$

In this report, only the errors associated with the estimated discharge coefficient have been estimated. Owing, however, to the difficulty in quantifying the quality of
the manufactured orifice plate flowmeter, such as the alignment of the orifice plate and the pressure tappings, circularity and cylindricality of the pipe conduit and orifice plate and orifice sharpness, some of these associated terms involved in determining the overall uncertainty of the determined flow rate have been either neglected or referenced to usual values or maximum allowable values specified by British Standards where possible.

The uncertainty sources included in the approximation of the discharge coefficient errors are the inherent 0.5% arising from the chosen orifice to pipe diameter ratio and an additional 0.91% from the pipe diameter being less than the 50mm, calculated by using Eq. 3-18 (ISO 15377 and 5167:2). Further correction has been made owing to the aforementioned disparity between the pressure tapping location with that of the recommendation using the Eq. 3-19 at every calibration point and over the calibration range (ISO 12767). Also errors arising from possible elastic deformation of the orifice plate have been considered, even though it was expected to be small. The relative change in discharge coefficient owing to plate deformation is calculated as a percentage using Eq. 3-20 before being converted into a percentile error with Eq. 3-21 (ISO 12767).

\[
\frac{\delta C}{C_1} \% = 0.5 + 0.9(0.75 - \beta)
\left(2.8 - \frac{D_p}{25.4}\right)
\]  

\[
\frac{\delta C}{C_2} \% = 25 \cdot \left(\frac{C_{Dp \text{ and } Dp/2}}{C} - 1\right)
\]  

\[
\delta C_{\text{def}} = \frac{100 \cdot \Delta P}{Y} \left[\frac{D_2}{E_{or}}\right]^2 \left[\frac{a_1 D_2}{E_{or}} - a_2\right]
\]  

\[
\frac{\delta C}{C_3} \% = 0.5 + \left|\delta C_{\text{def}}\right|
\]  

where;

\[Y\] - Young’s modulus
\[D_2\] - orifice plate support diameter
\[a_1 = \beta (0.135 - 0.155\beta)\]
\[a_2 = 1.17 - 1.06\beta^{1.3}\]
\[C_{Dp \text{ and } Dp/2}\] - calculated discharge coefficient for \(D_p\) and \(D_p/2\) pressure tappings
\[C_{C}\] - calculated discharge coefficient for corner pressure tappings
The total relative standard uncertainties over the range of the calibration flow rate are shown in Figure 3.16 and the maximum value is around 3%. This value together with other uncertainties gives the total expanded uncertainty of ±2.33% for the estimated mass flow rate with a 95% confidence level.

Figure 3.16 Total relative standard uncertainty in the estimated flow rate over the range of the calibration flow rate
4 The characteristics of the fluidic oscillator

4.1 Steady-state analysis

The primary emphasis of this study is to define a flow gain, $G_F$, and a fanout ratio, $n_F$, of the test model, which are often used as a figure of merit in comparing a dynamic switching response of such devices. To do so, the fluidic oscillator model is modified into a fluidic amplifier as described in Chapter 3 and a steady control flow is introduced to a control port. The control flow volume flow rate is then gradually increased until the switching of a steady attached power jet flow takes place. The control flow rate at the switching point together with the corresponding output flow rates from the both exit orifices of the device are measured at different supply flow rates, allowing the aforementioned performance parameters to be determined.

4.1.1 Control flow rate at the point of jet switching

From the previous work of Tesar et al (2006) and Warren (1963), a ratio of the control to the supply mass flow rate is expected to be around 7% at a $Re_{b_s}$ of $7.6 \times 10^3$ for a model which has a similar interaction region as the current model and it will decrease with an increasing power nozzle Reynolds number, $Re_{D_h}$. Here $Re_{b_s}$ is based on the width of the power jet nozzle width and is calculated using Eq. 4-1. $Re_{D_h}$ is the Reynolds number defined using the hydraulic diameter of the power jet nozzle, $D_h$ as shown in Eq. 4-2. The hydraulic diameter of the power jet nozzle, $D_h$ is calculated using Eq. 4-3.

\[
Re_{b_s} = \frac{\rho U_s b_s}{\mu} = Re_{D_h} \cdot \frac{b_s}{D_h}
\]
\[
Re_{D_h} = \frac{\rho U_s D_h}{\mu} = \frac{2m_s}{\mu(b_s + h)}
\]
\[
D_h = \frac{2(b_s \times h)}{(b_s + h)} = \frac{2AR \cdot b_s}{(1 + AR)}
\]

Figure 4.1 presents the measured control mass flow rates through the active side control port, $c_1$, at the power jet switching point for different power jet Reynolds numbers and its ratio to the supply mass flow rate, $\eta_{m,c1}$. Throughout the analysis, subscript 1 denotes an initial active side of the fluidic amplifier prior to the
completion of the jet switching while 2 denotes an active side after the successful power jet switching has taken place. The measured volume flow rate, \( q_v \), has been converted into the mass flow rate, \( \dot{m} \), with knowledge of the density of the fluid, \( \rho \), as shown in Eq. 4-4. \( \rho \) is estimated from the time-averaged static pressure reading.

\[
\dot{m} = \rho \cdot q_v
\]

\textit{Eq. 4-4}

![Figure 4.1 Variations of the control mass flow rate, \( \dot{m}_{c_1} \), at the switching point and its ratio to the supply flow rate, \( \eta_{m,c_1} \), with \( \text{Re}_{D_h} \).}

Figure 4.1 clearly shows a trend of increasing control mass flow rate requirement at the switching point with an increasing supply flow level. However, a closer examination reveals an abrupt jump in the magnitude of \( \dot{m}_{c_1} \) at \( \text{Re}_{D_h} \) around \( 2.4 \times 10^4 \). This Reynolds number is henceforth referred as the critical Reynolds number, \( \text{Re}_{\text{crit}} \), which divides the supply flow level into a low-band and a high-band Reynolds number regimes. The presence of a characteristic change in the switching dynamics between the two Reynolds number regimes separated by \( \text{Re}_{\text{crit}} \) is more apparent when the control to the supply flow rate ratio, \( \eta_{m,c_1} \), is examined. In the low Reynolds number regime, the control mass flow rate is a constant ratio of 0.071 of the supply flow rate. About 0.087 near \( \text{Re}_{\text{crit}} \), a further increase in the supply mass
flow rate results in a sudden increase in $\eta_{m,c_1}$. Beyond this point, the ratio of the control to the supply flow rate at the switching point decreases almost linearly to 0.066 at $Re_{D_h}=6.5\times10^4$.

Based on the typical $\eta_{m,c_1}$ ratio being 0.05 to 0.15 as mentioned in the literature review, the result shown above suggests that the existing model has a relatively low or a moderate latching stability, i.e., relatively high tendency for power jet switching. The trend of change in $\eta_{m,c_1}$ with the power jet Reynolds number observed in Figure 4.1 at $Re_{D_h} > Re_{crit}$ is in accordance with the conclusion drawn by Warren (1963).

Tesar et al (2006) studied a bistable fluidic oscillator which shares a similar interaction region geometry as that of the current design but with a different power nozzle width, a nozzle aspect ratio and a control to power nozzle width ratio. The most influential parameter which affects $\eta_{m,c_1}$ at the switching point is the control to power nozzle width ratio which was about the half the size of the current model. Although their $\eta_{m,c_1}$ is expected to be higher than that of the current model, their computed value of $\eta_{m,c_1}$ of 7% for $Re_{h_s}=7.6\times10^3$ is in close agreement with the result from the present experiment observed at $Re_{D_h} < Re_{crit}$. Referring to the paper by Savkar (1966), this is perhaps due to a corresponding increase in the relative control to supply jet momentum ratio which arose from the use of a smaller control to power nozzle width ratio in the work of Tesar et al (2006).

As it was described in the literature review, the value of $\eta_{m,c_1}$ of a bistable fluidic wall attachment device is predominantly determined by two factors, i.e. the size of the separation bubble and the latching stability in the device. The separation bubble size near the power jet attachment point is known to be strongly governed by the geometrical parameters, such as the offset and the attachment wall angle, but relatively independent of the power jet nozzle Reynolds number (Johnston, 1963). Furthermore, for a device having a small or zero offset, given the moderate attachment wall angle, the power jet will attach to the wall shortly after leaving the power nozzle. Thereby a negligibly small separation bubble is formed or its formation may even be suppressed within the attached side control nozzle as schematically shown in Figure 2.31 a). On the other hand, the latching stability is
determined by the size and the strength of the standing vortex and since it is formed as a portion of the power jet flow captured by the splitter, its growth is governed by the power jet Reynolds number as well as the shape and the location of the splitter. The standing vortex will influence the latching stability by counteracting the control flow which enters from the attached side control nozzle, hence forces the power jet to detach from the wall. Also depending on the output load condition and the use of vent, a return flow may result which travels towards the upstream output aperture that can effectively counteract the standing vortex, hence reducing the latching stability.

Based on the above discussion, the observed changes in $\eta_{m,s_1}$ with increasing power jet Reynolds number may imply a possible characteristic variation in the evolution of the standing vortex. One possible explanation would be a limitation in the size and strength of the standing vortex by the physical confinement of the device cavity. It is believed that structure of the standing vortex evolves consistently with the supply flow level and hence the control flow levels out to compromise the increasing latching stability in the sub-critical regime. With a further increase in $Re_{D_b}$, the structural evolution of the standing vortex may be halted or stabilised causing that the control rate ceases to depend on the supply flow rate.

4.1.2 The characteristics of output jets

The characteristics of the output jet were evaluated using the velocity data obtained from a single normal probe placed at one orifice diameter downstream of the centre of the exit orifice.

From the statements made by Morris (1973), Kirshner and Katz (1975) and Tesar et al (2006), some induced flow motion is expected to occur at a low state side exit port either in the forms of a spill flow of a small portion of the power jet flow or a suction of the ambient fluid. Morris (1973) states that one of an intrinsic nature of a wall attachment device is that a perfect zero-low state of output signal is unattainable due to their inherent geometrical configuration. Hence a portion of the power jet flow tends to escape through the low state side exit, which can sometimes be disadvantageous. Also, however, Kirshner and Katz (1975) stated that a suction flow is also possible from this side exit port if the pressure at its exit conduit becomes substantially lower than that of the ambient or an interconnected element’s cavity. Furthermore the numerical results of Tesar et al (2006) show that a suction
flow of approximately 20% of the supply flow rate occurs at the switching point with a control flow and about 17% occurs during the steady state operation in the absence of any control signals at $Re_b$ of $7.6 \times 10^3$.

The flow motion that occur at the low state side output holds an equal possibility of being a spill or a suction flow. However, due to a flow directional insensitivity of a single normal hotwire probe, both of the suction and spill cases are considered in estimating its output flow rate, $\dot{m}_{o_b}$, over the complete data analysis. These assumptions are denoted with subscripts SU and SP respectively. For both cases the mass flow rates are calculated using the time-averaged jet centreline velocity, $\overline{u}_{j,\omega}$, but with different fluid density. The fluid density is assumed to be equal to that of the ambient for suction assumption case, whereas the density of the power jet flow is used for spill case.

The time-averaged jet centreline velocity, $\overline{u}_{j,\omega}$, obtained from the hotwire measurements at $1D_o$ downstream of the exit low state side orifice are presented in Figure 4.2 as a function of supply mass flow rate, $\dot{m}_s$, together with a deuced time-averaged spatial mean velocity across an orifice, $\overline{u}_{m,\omega}$, from $\overline{u}_{j,\omega}$. These values describe the induced flow motion after the completion of the power jet switching with a control flow remaining active. Using $\overline{u}_{m,\omega}$ data the output mass flow rates, $\dot{m}_{o}$, from a low state side output orifice are calculated and presented in Figure 4.3 as a function of exit jet Reynolds number, $Re_{D_o}$, defined in Eq. 4-5. This confirms the validity of making the Hagen-Poiseuille assumption by showing that all the calculated $Re_{D_o}$ being lower than $2.3 \times 10^3$.

$$Re_{D_o} = \frac{\rho \overline{u}_{m,\omega} D_o}{\mu}$$  \hspace{1cm}  \text{Eq. 4-5}

The calculated output mass flow rate, $\dot{m}_{o}$, data and their ratio to the supply flow rate, $\eta_{m,\omega}$, of the low state side exit have been plotted in Figure 4.4 as a function of a power jet Reynolds number to verify their correlation. The calculated mass flow rate is found to increase with a supply flow level in general but shows a tendency of its saturation beyond $Re_{D_o} = 4 \times 10^4$ as would expected from the previous analysis of the jet centreline velocity variation. Also it is evident from examining $\eta_{m,\omega}$ variation
that $\dot{m}_{in}$ relative to its $\dot{m}_s$ decreases with increasing $Re_{D_o}$ overall. In addition to this sudden jump in $\dot{m}_{in}$ is observed at $Re_{D_o} = 2.4 \times 10^4$, which coincides with the previously defined $Re_{crit}$.

Figure 4.2 Time-averaged velocity measured at the centre and $1D_o$ downstream of the output orifice at Low state and corresponding time-averaged spatial mean velocity across the orifice assuming Hagen-Poiseuille flow.

Figure 4.3 The calculated mass flow rate through the inactive side exit orifice for both cases of flow being suction and spill as a function of the orifice Reynolds number, $Re_{D_o}$.
Figure 4.4 The calculated mass flow rate through the inactive side exit orifice and its ratio to the supply flow rate plotted against $Re_{D_h}$.

The output mass flow rate from a high state side exit port, $\dot{m}_{v_2}$, has been calculated by assuming $\bar{u}_{v_2}$ as $\bar{u}_{m,v_2}$ since the jet velocity profile is expected to be a top-hat shaped based on their $Re_{D_h}$ being generally in the orders of $O(10^4)$, and this is shown in Figure 4.5. By examining $\eta_{m,v_2}$ an existence of almost a direct proportionality between the high state side output and the supply flow rate can be seen.
Figure 4.5 The calculated mass flow rate through the active side output orifice and its ratio to the supply flow rate as a function of the $Re_{D_h}$.

However, the results shown in Figure 4.5 do not account for the previously verified secondary flow from the low state side output, $\dot{m}_{o_l}$, and the control flow added to complete jet switching, $\dot{m}_{o_s}$. Thereby $\dot{m}_{o_s}$ is modified as in Eq. 4-6 and Eq. 4-7 to account for these flows depending on the aforementioned two different hypotheses in describing the flow directional natures of $\dot{m}_{o_l}$ and the results are shown in Figure 4.6.

\[
\dot{m}_{o_2, SU} = \dot{m}_{o_2} - \dot{m}_{o_l, SU} - \dot{m}_{e_1} \tag{Eq. 4-6}
\]

\[
\dot{m}_{o_2, SP} = \dot{m}_{o_2} + \dot{m}_{o_l, SP} - \dot{m}_{e_1} \tag{Eq. 4-7}
\]

As it can be noticed from Eq. 4-6 and Eq. 4-7, the entrained flow from the ambient through the high state side control port, $\dot{m}_{e_2}$, subsequent to the jet switching has not been included in approximating the modified output flow rate. This is because it is assumed that this entrained flow is dispersed in maintaining a separation bubble at a state of equilibrium that forms at the attached side wall; or more precisely that develops within the active side control nozzle in the case of the current model.
Although a flow directional measurement shall never be interpreted from a velocity measurement of a steady flow which made with a single normal hotwire probe, the author has attempted to define the origin of $\dot{m}_{v_0}$ based on the mass conservation law. Thereby establishing the input to modified output mass flow rate ratio of 1 as a target value and examining the results shown in Figure 4.6, the spill assumption to be an appropriate description for low Reynolds number regime whereas the suction hypothesis describes better for the regime having $Re_{D_s}$ greater than the $Re_{crit}$.

![Figure 4.6](image)

Figure 4.6 The corrected mass flow rate ratio through the active side output orifice for the mass flow rate through the inactive side.

### 4.1.3 Flow gain and fanout ratio

Using the results from the previous sections of this chapter, two steady-state performance parameters of the current model and their correlations with the power nozzle Reynolds numbers can be obtained. The performance parameters considered are the flow gain, $G_F$, and the fanout ratio, $n_F$, which are defined below.

$$G_F = \frac{\dot{m}_{v_0}}{\dot{m}_{v_1}} \quad \text{Eq. 4-8}$$

$$n_F = \frac{\dot{m}_{v_2,S}}{\dot{m}_{v_1,S}} \quad \text{Eq. 4-9}$$

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Where $\dot{m}_{o2}$ and $\dot{m}_{c1}$ are the mass flow rates across the high state side output orifice and the control nozzle at the switching point respectively, and $m_{o2,S}$ and $m_{c1,S}$ are the corresponding total mass flow over the power jet switching period. Thereby the flow gain describes the easiness of the power jet switching and/or the latching stability in the device, whereas the fanout ratio describes the number of passive devices that can be derived by its output. Those passive devices, however, must have a similar size and shape and are operated at the same supply flow condition of the driving device.

The high state side output mass flow rate data presented previously in Figure 4.5 are plotted as a function of its corresponding control mass flow rate from Figure 4.1 and shown in Figure 4.7. The slope of a straight line of a best fit drawn through these data points enables an approximation of an averaged steady-state gain for the current model which found to be equal to 14.86. However, Figure 4.7 also shows a change in the flow gain with an increasing control flow rate and one can recognise that its trend of change and values closely match the inverse of $\eta_{m,c1}$ shown previously in Figure 4.1. This is more evident when $G_F$ is plotted as a function of the supply flow level as in Figure 4.8 in which it appears to firstly have a constant value of 14 at low Reynolds numbers then linearly increases with the power jet Reynolds number. The correlation between the flow gain and the control to supply flow rate ratio re-confirms an existence of a direct proportionality in the device output with an input. The continuous increase of $G_F$ at the high Reynolds numbers noticed in Figure 4.8 demonstrates a decrease in the latching stability which indicates that the jet switching becomes more prone to a control signal with increasing supply flow level.

Since the output of the fluidic amplifier in a steady-state operation is a continuous jet, the averaged steady-state flow gain would effectively describe an averaged steady-state fanout ratio, $n_F$. Hence the output from the high state side of the current model can control up to 14 identical passive devices when they are running at the same supply flow condition. However, it should be stated that this would not describe the fanout ratio of the fluidic oscillator used to drive an array of fluidic amplifiers and this will be further described in the latter section of this chapter.
Figure 4.7 Output mass flow rate at the high state side and flow gain versus control mass flow rate.

Figure 4.8 Flow gain of the fluidic oscillator at the steady state test conditions.

4.2 Oscillatory jet characteristic
In this section, the fluidic amplifier studied in the previous section is transferred to a fluidic oscillator by interconnecting the two control ports together with a time delay feedback loop. A fluidic oscillator in such a configuration is expected to generate self excited pulsing jets from its conjugating pair of outputs with a 180° phase difference. The purpose of this study was to examine the
characteristics of the oscillatory jet, and the effect of $Re_{D_h}$ and feedback loop length on the oscillating frequency and switching time of the jet.

### 4.2.1 Output jet characteristics

The results in this section of the chapter show the correlation between the output jets produced from a conjugating pair of the output orifices and the pulse transitional measurements over a one cycle.

The Figure 4.9 a) shows the instantaneous velocity fluctuations measured simultaneously at the centre of the both exit orifices of the fluidic oscillator at $Re_{D_h}$ of $4.92 \times 10^4$ with a feedback loop length of 1.16m ($\xi = 580$). As expected the periodic oscillatory outputs from a pair of conjugating output orifices show a 180° phase difference. As the power spectral density shows in Figure 4.9 b), the dominant fluctuation frequency is about 100Hz.

The phase-averaged velocity, $\langle u \rangle$, profiles are shown in the Figure 4.9 c) while Figure 4.9 d) showing a phase averaged turbulence intensity, $\overline{T_u}$, calculated over one cycle; usually a synchronised artificial reference signal is used for phase analysis of a pulsing jet, however, due to an instability in velocity waveform a state reference value is used in separating each cycle of pulsation. The phase-averaged velocity profiles are then normalised by the time-averaged velocity, $\overline{\nu_{c,p}}$, and plotted in Figure 4.10 together with state reference values which are used for verifying its transitional characteristics.

It is common for the high and low references values to be 10% below and above of the high and low state values (pulse peak and valley) respectively and being evenly distributed about the middle reference value, however this gave rather an inappropriate high reference value for the intended analysis as it can be seen in Figure 4.10; this effectively distorted rise and decay times of the pulsation. Thereby the high reference value has been systematically defined as 80% of the amplitude whereas the usual 50% and 10% are used for mid and low references respectively.
Figure 4.9 The output jet characteristics of the fluidic oscillator model measured simultaneously at the two exit orifices at $Re_{D_h} = 4.92 \times 10^4$ and $L=1.16$ and approximated $T$ of 10.5ms. a) Instantaneous velocity profile; b) Power spectral density; c) Phase-averaged velocity variation over a one cycle; d) Phase-averaged turbulence intensity over a one cycle.
Figure 4.10 The normalised phase-averaged velocity by the time-averaged velocity over a one cycle together with state reference values

Figure 4.10 shows a square waveform signal with an average duty cycle of 50.6%, which agrees with the theoretical assumption of 50% as employed in Eq. 4-13, and the rise and fall time of 1.65ms and 1.62ms respectively. The wave shape of the oscillating output jet from the existing model is governed by the switching delay time, \( \tau_L \), which is determined by the feedback loop geometry.

4.2.2 The frequency of oscillatory jets

In this section the effects of the supply flow level and the feedback loop length on the frequency response of the periodically oscillating output jet are examined. In addition, the frequency measurements are used to estimate the switching times of the present fluidic oscillator. Referring to the results of Tesar et al (2006) the frequency of the self excited jet oscillation is expected to increase with \( Rc_{D_j} \) while decrease with feedback loop length. Also they have demonstrated an existence of frequency saturation at high supply flow rates and its correlation with a local speed of sound.

Figure 4.11 and Figure 4.12 show the instantaneous jet velocity variations and the corresponding spectra at two different power jet Reynolds numbers with a feedback loop length of 1.16m. The jet velocity variation and power spectral density measurements at \( Rc_{D_j} =4.37\times10^4 \) shown in Figure 4.11 clearly demonstrate an existence of a fundamental frequency supported by the presence of its harmonics in Figure 4.11 b). However, instability in the oscillation period from the instantaneous
velocity measurements at $Re_{D_h}$ of $7.54 \times 10^3$ can be noticed Figure 4.12 a), which in fact indicating occasional switching failures. This is also apparent in the power spectral density shown in Figure 4.12 b) in which a fundamental frequency and its harmonics are less well defined.

This comparison partially demonstrates the existence of an oscillation instability that constantly encountered throughout the experiments at a relatively low Reynolds numbers. Generally, when $Re_{D_h}$ is close to the order of $\sim O(10^4)$ unstable periodic oscillations are noticed. But as the feedback loop length increases, this instability starts to diminish at higher power jet Reynolds numbers. The critical Reynolds number at which this instability starts to diminish has not been established in this study, however, due to the difficulty in defining a quantitative measure of the jet oscillation instability.

One possible cause for this would be related to the flow development of the supply flow, especially for the current flow feeding method. At low supply flow rates, secondary vortices may form (swirl flow) upstream of the power nozzle, which tend to be suppressed as the supply flow rate increases. This swirl flow subsequently disturbs the power jet flow development through the power nozzle hence destabilising its flow entrainment at the downstream of the nozzle.
Figure 4.11 Instantaneous velocity fluctuations and power spectral density for $Re_D = 4.37 \times 10^4$; $L = 1.16$; $t_{total} = 4s$. 
Figure 4.12 Instantaneous velocity fluctuations and power spectral density for $Re_{D_h} = 7.54 \times 10^3$; $L=1.16$; $t_{total}=4s$.

The variations of the oscillating frequency of the output jets for a range of feedback loop lengths in the range of power jet Reynolds number, $Re_{D_h}$, of $6.5 \times 10^3$ to $7.3 \times 10^4$ are shown in Figure 4.13 a). The frequency response at $L=0.3m$ has not been included, due to unstable jet oscillations except at very high Reynolds numbers. This suggests an existence of a possible limit in the minimum length of feedback loop for the generation of a stable oscillating jet. Taking $0.3m$ as the limit for the current design yields an approximated minimum allowable nozzle to length ratio, $\xi$, of 150.

The plots of frequency data in Figure 4.13 a) shows that the frequency increases linearly with an increasing power jet Reynolds number up to $Re_{D_h}$ of $3.5 \times 10^3$. As
$Re_{D_h}$ increases further, there is a reduction in its gradient indicating frequency saturation for all feedback loop lengths studied in this report. Also at a given power jet Reynolds number, a higher jet oscillation frequency was obtained when a shorter time delay feedback loop is used.

The same data are presented by plotting the frequency variations against $L$ for different $Re_{D_h}$ in Figure 4.13 b). At $Re_{D_h} = 1.0 \times 10^6$, the frequency seems to vary with $Re_{D_h}$ linearly whereas it varies inversely proportional reduction at higher $Re_{D_h}$. In Figure 4.13 b), a 4th order polynomial curve fitting has been performed over the data points and used in approximating the frequency responses at a chosen Reynolds numbers for each feedback loop length.

The frequency measurements shown in Figure 4.13 a) are also used in calculating the Strouhal number, $St$, defined as:

$$St = \frac{f \cdot L}{U_s} \quad \text{Eq. 4-10}$$

Where $U_s$ is the mean power jet velocity at the exit of the power nozzle. These values are then plotted as a function of $Re_{D_h}$ in Figure 4.14. Its trend of change closely follows that of the $St$ behaviour of a bluff body in the subcritical range of Reynolds number. At $Re_{D_h}$ of $3 \times 10^4$, $St$ appears to become a constant between 0.6 and 0.8.

The modified Roshko number, $Ro_{mod}$, is defined as

$$Ro_{mod} = St \cdot Re_{D_h} = \frac{f \cdot L \cdot D_h}{\nu} \quad \text{Eq. 4-11}$$

This result shows that there exists a linear relationship between $Ro_{mod}$ and $Re_{D_h}$ which can be described by a straight line found from a curve fitting of the data points.

$$Ro_{mod} = 0.76 Re_{D_h} - 3.57 \times 10^3 \quad \text{Eq. 4-12}$$
Figure 4.13 The frequency measurements of the periodically oscillating jet produced from the fluidic oscillator. a) Frequency variations as a function of $Re_{Dh}$ for different L. b) Frequency variations as a function of L for different $Re_{Dh}$. 
Figure 4.14 Correlation between Strouhal number and $Re_{D_h}$.

Figure 4.15 Correlation between modified Roshko number and $Re_{D_h}$. 
4.2.3 The switching time

The frequency saturation noticed above can be explained by understanding the terms governing the period of the jet oscillation. For a bistable feedback loop oscillators, the oscillation period is usually expressed as in Eq. 4-13.

\[ T = 2 \cdot (\tau_p + \tau_s) \]  \hspace{1cm} \text{Eq. 4-13}

\( \tau_p \) is the control signal propagation time around the feedback loop. The other term \( \tau_s \) is the characteristic switching time of an element.

Since the control signal is transmitted by rarefaction waves and pressure waves, it will always propagate at a local speed of sound and it can be estimated by diving the feedback loop length, L, by the local speed of sound, a, as shown in Eq. 4-14.

\[ \tau_p = \frac{L}{a} \]  \hspace{1cm} \text{Eq. 4-14}

\( \tau_s \) is governed by the input and output characteristics, geometries of the device and the feedback loop. The device geometry and the input and output characteristics will govern the latching stability and the size of separation bubble. On the other hand, the feedback loop diameter and its length determine the pneumatic resistance and capacitance of the feedback loop hence they control the time required for a sufficient level of pressure difference to be achieved to initiate the control signal propagation. Hence \( \tau_s \) can be further subdivided into \( \tau_a \) and \( \tau_L \), i.e. \( \tau_s = \tau_a + \tau_L \),

Here \( \tau_a \) stands for the fundamental switching time of the device determined by the input and output characteristics and device geometry whereas \( \tau_L \) describes the switching time ruled by the feedback loop geometry, henceforth is called a switching delay time; here concept of \( \tau_L \) is introduced, although \( \tau_p \) involves feedback loop length in its definition, to emphasise the geometrical influence of the feedback loop while \( \tau_p \) stressing the thermal property of the flow within feedback loop.

\( \tau_s \) for different feedback loop lengths over the range of Re\( D_h \) are calculated using the oscillation period from the spectrum analysis with Eq. 4-13. These are then normalised by the propagation time, \( \tau_p \), as an effort to verify whether there exists a dominant time parameter at two different Reynolds number regimes. The results
shown in Figure 4.16 demonstrate that \( \tau_S / \tau_p \) for different feedback loop lengths collapse on the same curve and it also asymptotically reduces close to zero as \( Re_{D_h} \) increases.

![Figure 4.16 Correlation between normalised switching time and \( Re_{D_h} \)].

Since \( \tau_S \) is always found to be larger than the propagation time for all test cases at \( Re_{D_h} < 3.0 \times 10^4 \), it is apparent that the period of the jet oscillation is mainly governed by the switching time, where the linearly increasing frequency response has been noticed in Figure 4.13 a). As the switching time reduces and becomes comparable to the propagation time, the frequency saturation starts to initiate when the \( \tau_S / \tau_p \) ratio reaches approximately a value of 1 and the Strouhal number locks on to a constant value.

The saturation frequency of a given fluidic oscillator can be estimated by substituting \( \tau_S / \tau_p = 1 \) into Eq. 4-13, i.e.

\[
\begin{align*}
    f_{sat} &= \frac{1}{2 \cdot (2 \tau_p)} = \frac{1}{4(\xi b/a)} \\
    \text{Eq. 4-15}
\end{align*}
\]

where \( b_S \) is the power nozzle width, \( \xi \) is the feedback loop length to power nozzle width ratio and \( a \) is the local speed of sound.
From Eq. 4-15 one can notice that the local speed of sound not only determines the propagation speed of the control signal time but the saturation frequency as well. Thereby for a given control loaded fluidic oscillator planform with a fixed feedback loop length, i.e., fixed \( \tau_s \) and \( \tau_L \) hence \( \tau_s \), the onset of frequency saturation can either be promoted or postponed by modifying the local speed of sound by altering the thermal properties of the working fluid.

As an effort to estimate the aforementioned fundamental switching time, \( \tau_s \), of the test model, T/2 values are calculated from the oscillation frequency approximated as in Figure 4.13 b) and plotted as a function of feedback loop length with \( \text{Re}_{D_h} \) as a parameter as shown in Figure 4.17. An exponential curve fitting is then drawn through the data points and extrapolated to find the switching time at the zero feedback loop length. The switching time found by an extrapolation is then plotted as a function of a power jet Reynolds number as in the Figure 4.18, which is believed to describe the fundamental switching time of the existing control loaded fluidic oscillator that governed solely by the device planform geometry. The result shows that the fundamental switching time reduces inverse proportionally with increasing power nozzle Reynolds numbers and it seems that the minimum attainable \( \tau_s \) of the current model is approximately equal to 2ms for the range of tested \( \text{Re}_{D_h} \).

![Figure 4.17 Variations of T/2 as a function of the feedback loop length at different \( \text{Re}_{D_h} \) which are used to find the natural switching time](image-url)
of the fluidic oscillator by extrapolation.

![Graph](image)

**Figure 4.18** Approximated fundamental switching time, $\tau_{c,s}$, of the test model at different power nozzle Reynolds number.

### 4.2.4 Assessment of capability in driving other fluidic amplifiers

In the present study, the fluidic oscillator examined here will be used to drive an array of seven fluidic amplifiers which have an identical shape and running at a same supply flow level. In this section, the output mass flow rate over the blowing phase from the fluidic oscillator at three different $Re_{D_h}$ values is calculated using the results in Figure 4.10 in order to verify the capability of this fluidic oscillator in doing so. The chosen $Re_{D_h}$ values are selected to cover the sub critical regime and possible maximum and minimum fanout ratio values based on the previous results in Figure 4.8.

The fanout ratio, $n_F$, at a given operating condition is used as a measure of the fluidic oscillator capability in driving seven fluidic amplifiers simultaneously. $n_F$ is calculated in two different ways. One method involves a comparison of the total output mass flow over the blowing phase with the total control flow required over the switching period. In this case, the control to supply flow rate ratio from Figure 4.1 is used with an assumption that the switching time of the fluidic amplifier is equal to the analytically approximated natural switching time, $\tau_{c,s}$, as in Figure 4.18. The
other method of calculating the fanout ratio assumes the switching time of the amplifier to be comparable to the blowing period. Thereby the control flow rate required at the switching point has only been compared with the output flow rate over the blowing period. However, it must be stated that none of these methods gives an exact fanout ratio of the fluidic oscillator used in such operating condition since the steady-state analysis with an impulsive control signal case is not considered in this study. Nevertheless these two methods cover the two extreme cases of the maximum and minimum possible fanout ratios, thereby the capability of the oscillator in actuating seven fluidic oscillators can be approximated.

Table 4.1 shows the three different $Re_{Dh}$ cases considered and its corresponding fanout ratios. Both $\frac{m_{o,s}}{m_{c,s}}$ and $\frac{m_{o,S}}{m_{c,S}}$ values in Table 4.1 for all Reynolds numbers considered found to be sufficiently higher than the targeted value of 7 and hence the use of the existing fluidic oscillator as a driving actuator of seven fluidic amplifiers is valid under the circumstance that they are not operating at a higher power jet Reynolds number than that of the oscillator.

<table>
<thead>
<tr>
<th>$Re_{Dh}$</th>
<th>$T_{pulse}$ (ms)</th>
<th>$\tau_n$ (ms)</th>
<th>$m_{o,S}$ (mg)</th>
<th>$m_{c,S}$ (mg)</th>
<th>$\frac{m_{o,S}}{m_{c,S}}$</th>
<th>$\frac{m_{o,s}}{m_{c,s}}$</th>
</tr>
</thead>
</table>
| $1.58 \times 10^4$ \ 
($\bar{m_i} = 1.43$ g/s, $f_{act}=32$ Hz) | 15.7 | 5.14 | 19.05 | 0.17 | 36.26 | 11.87 |
| $2.69 \times 10^4$ \ 
($\bar{m_i} = 2.69$ g/s, $f_{act}=62$ Hz) | 8.1 | 2.83 | 19.25 | 0.25 | 32.87 | 11.47 |
| $5.89 \times 10^4$ \ 
($\bar{m_i} = 5.34$ g/s, $f_{act}=105$ Hz) | 4.8 | 1.92 | 30.03 | 0.35 | 42.14 | 16.85 |

Table 4.1 Fanout ratios of the fluidic oscillator calculated at various $Re_{Dh}$.

### 4.3 Assessment for flow control applications

The output jet oscillation characteristics of the fluidic oscillator model studied in this report suggests its possibility to be used as an alternative PVGJ actuator for separation control. In order to deliver the desired flow control effect for a given application, the fluidic actuators are expected to satisfy at least three basic requirements on their performance in terms of jet peak velocity, oscillating frequency and exit orifice diameter.
The jet peak velocity to the freestream velocity ratio is in the range of 1 to 2 at which the pulsed jets are reported to be effective in delaying flow separation and not extensively energy demanding.

The required minimum actuation frequency for the purpose of flow separation control would be to achieve a reduced frequency number, \( f^+ \), of \( \sim 1 \). The reduced frequency number is defined as in Eq. 4-16, where \( f_{act} \) is the actuation frequency, \( c \) is the characteristic length of the lifting surface and \( U_\infty \) is the freestream velocity.

\[
f^+ = \frac{f_{act}c}{U_\infty}
\]

\[\text{Eq. 4-16}\]

Many research results, however, demonstrate that PVGJ based separation control actuators provide an optimal separation control when operated with \( f^+ \) in the range of 0.58 to 2 (Donovan et al, 1998 and Seifert et al, 1996). In addition to this other researchers also showed a successful separation control by actuating at a reduced frequency number in the order of magnitude of \( O(10) \) (Glezer et al, 2005). However, Cerretelli and Kirtley (2009) have successfully demonstrated that the fluidic oscillators produce superior performance in flow control when actuated within \( f^+ \) range of 0.4 to 1.

In order to minimise the disturbance of the exit orifices to the flow on the surface to be controlled, the orifice diameter is expected to be 5 to 20% of the local boundary layer thickness. This usually will require the orifice diameter being in the range of sub millimetre scales. As it is to be discussed in the next chapter, this will place an additional demand on the design of the fluidic actuator in order to produce a fully pulsed with a minimum jet velocity close to zero.

For the aerofoil model to be used in subsonic flow control experiments to be undertaken in a laboratory, the freestream velocity is in the range of 10m/s in a wind tunnel with a typical chord length of 0.3m and boundary layer thickness of 5mm. According to the aforementioned criterion, this would require flow control jets of 10 to 20m/s issues from orifices of 1mm in diameter at an oscillating frequency of 13 to 33Hz.

For flow separation control on the trailing edge flap of aircraft high-lift system at take off or landing, the freestream velocity is in the range of 100m/s with a typical chord
length of 0.5m and boundary layer thickness of 5mm. This would require flow control jets of 100 to 200m/s issues from orifices of 0.5mm in diameter at an oscillating frequency of 80 to 200Hz.

The existing model can be scaled down to the power nozzle width size of 0.25mm, which is a commonly used value for a fluidic oscillator. This then provides an output orifice with a diameter of 0.75mm based on the existing power nozzle width to output orifice ratio of 3. Further reduction in the output orifice diameter approximately by half is also possible by introducing a concentric reduction at its outputs according to the findings from the study of an array of fluidic amplifiers. The scaling in power nozzle width will also improve the maximum attainable oscillation frequency, with an approximated saturation frequency of about 1.1KHz by substituting the modified power nozzle width and the minimum $\xi$ of 150 into Eq. 4-15. These values, which can easily be achieved by minor modifications to current model, assures possible implementation for those flow control applications described above. Moreover, further improvement in frequency bandwidth is possible by reducing the output conduit length, i.e., reducing the convection time.

However, one may find the intrinsic coupling behaviour of a fluidic oscillator between its supply flow rate and the output frequency for a chosen feedback loop length disadvantageous as a potential separation control actuator. This practically presents a difficulty in controlling the output jet to freestream velocity ratio independently from an oscillation frequency. The simplest remedy to overcome this issue would involve use of a secondary driving unit such as a fluidic oscillator to control the actuation of the oscillator. Nevertheless, the most preferable way would be to improve the existing design so that one can attain the required minimum frequency response at a relatively low supply flow rate. This can be achieved by modifying the existing splitter location and its shape and/or the parameters governing $\tau_p$ as complementary efforts to scale reduction of the model as above.

4.4 Summary of findings

**Summary of findings from steady-state analysis of fluidic amplifier:**

- At low Reynolds number regimes ($Re_{D_2} < 2.4 \times 10^4$), the necessary control flow rate to complete the power jet switching is found to increase with the supply flow rate at a constant gradient of change, while it increases with a uniformly decreasing rate of change for higher Reynolds number regimes.
This trend of change in the relative control to supply flow rate ratio at the switching point is in accordance with Warren (1963).

- Approximately 7% of the supply flow is necessary as a control flow to onset and complete the power jet switching for low Reynolds number regime while it increases to a 9% at Re\textsubscript{crit} then starts to decrease uniformly to 6.6% at Re\textsubscript{D_1} = 6.5 \times 10^4. The necessary control flow as a percent of the supply flow in the low Reynolds number regime is in accordance with Tesar \textit{et al} (2006).

- Purely based on the observation of a discontinuity in the functional variation of the control to supply flow rate ratio with the supply flow, the author suspects there exists a point where the size and strength of the standing vortex reaches a stable condition. This hypothesis is made by knowing that the standing vortex is the dominant source of the counteracting force to the control flow momentum entering from the control nozzle forcing the power jet detachment. This certainly requires a validation from possibly an internal flow study.

- An almost a direct proportional relation between the supply flow and the output flow from the high state side exit orifice have been verified.

- An induced flow motion at the low state side exit orifice has been observed. Although attempts to verify its flow nature have been made and states that spill flow occurs at Re\textsubscript{D_1} < 2.4 \times 10^4 while suction occurs at Re\textsubscript{D_1} > 2.4 \times 10^4. Caution must be applied in its application as this solely based on an indirect approximation made using the conservation of mass principle.

- The mass flow from the low state side exit orifice increases with the supply flow but shows a tendency of saturation at Re\textsubscript{D_1} > 4 \times 10^4. Also its output to supply ratio continuously reduces with increasing supply flow.

- The steady-state average flow gain is found by plotting the output flow from the high state side exit orifice against the control flow at the switching point and then by examining the gradient of a straight line drawn across the data point. It was found to be equal to 14 to a first integer value.

- As the output from the high state side exit orifice is directly proportional to the supply flow, the functional variation of the gain is expected to follow that of the inverse of the control to the supply flow rate ratio and the results confirms their correlation.

- To a first approximation the steady-state average flow gain has been considered as the steady-state average fanout ratio.
Summary of finding from study of the control loaded fluidic oscillator:

- The oscillation frequency found to increases with power jet Reynolds number while decreases with feedback loop length.

- Functional variation of the non-dimensionalised frequency numbers with \( Re_{D_h} \) follow that of a self excited oscillation of a bluff body in a cross flow giving a constant strouhal number of 0.7 at \( Re_{D_h} > 3 \times 10^4 \) and linearly increasing Roshko number with \( Re_{D_h} \).

- Instability in the oscillation period has been observed at \( Re_{D_h} < O(10^4) \) and/or with \( \xi < 150 \). The instability in the low Reynolds number regime is believed to be related to the flow development within the power nozzle. The presence of secondary vortices at the upstream of the power nozzle results as a swirl flow which possibly reflected as an unstable jet entrainment at the downstream of the nozzle.

- The non-dimensionalised switching time by the predefined theoretical control signal propagation time demonstrates that the frequency saturation will occur when this value reaches a unity.

- The natural switching time of the existing design has been estimated using the oscillation period data made available from the power spectrum measurements then plotting it against the feedback loop length for \( Re_{D_h} \) range of \( 1.0 \times 10^4 \sim 6.0 \times 10^4 \) with an incremental step of \( 2 \times 10^3 \) and extrapolating the exponential curve fit to find the switching time at zero feedback loop length. This result approximates the fundamental switching time of 2ms.

- The wave shape of the periodically oscillating output jet is found to follow that of a square wave signal due to the presence of the switching delay time, \( \tau_L \), with and average duty cycle of 50.6%.

- A 180° phase difference has been observed from a conjugating pair of exit orifices of the fluidic oscillator.

- The fluidic oscillator is found to produce adequate output flow over its pulse period to be used as an impulsive control signal to drive an array of seven fluidic amplifiers having an identical shape and running at power nozzle Reynolds number not exceeding that of the oscillator.

- Although, fluidic oscillator has intrinsic coupling between output characteristic and supply flow level it can be easily overcome with a careful review of the design to meet the intended application criterions.
An array of Fluidic amplifiers

The results in this section server to show the changes in the pulse characteristics of the output jet from a fluidic amplifier, which are caused by possibly a load mismatching as mentioned in Chapter 3 and to demonstrate a necessity of a design modification of the existing amplifier. Thereby the results shown in this section are only to demonstrate the validity of the remedy methods considered in overcoming the output signal attenuation and caution must be applied as the findings might not be transferable to other applications as the small number of test cases considered.

5.1 Effect of changing exit orifice diameter

The array of fluidic amplifiers was originally designed to accommodate a concentric reduction in its exit orifice diameter from 7.6mm to 1.1mm through a divergent 8mm in length. However, with this setup a fully pulsed output jet can not be achieved at the jet orifice of 1.1mm. Instead an attenuated pulsation equivalent to a continuous jet with a superimposed pulsation about its mean value is obtained.

The supply flow rate was reduced as a first attempt to achieve the fully pulsed jet response. However this was found to not give an expected improvement. Hence it was concluded that the supply flow rate is not responsible for the change of pulse behaviour mentioned above. This led to an option of enlarging the exit orifice diameter of the amplifiers based on the transmission line theory described by Kirshner (1966). Kirshner stated that when pulsing signals are transmitted through a channel terminated with an orifice, attenuation in the signal can occur if the cross-sectional area of the orifice, $A_o$, is not properly matched with that of the chamber at the plane of the orifice, $A_{ref}$. He further stated that there only exists a single matched point area ratio where the signal attenuation can be perfectly removed. According to an illustration provided by Kirshner (1966), if the exit orifice is designed too large relative to the size of the channel conduit then the pulsation will be superimposed to oscillate about a lower mean value while too small orifice diameter will shift the pulsation to occur at a higher mean value. Although the most promising remedy method would involve making the amplifiers to become load insensitive by introducing a vent to its configuration, but this will involve too much modification on
what is already manufactured and furthermore it is not an ideal option for the amplifier's intended application in flow control due to a loss in overall efficiency.

Two additional exit orifice diameters, i.e. \( D_o=3.5\text{mm} \) and \( 2.5\text{mm} \), have been tested and the velocity fluctuations at the jet exit are measured. It can be seen in Figure 5.1 that a near fully pulsed jet is obtained at \( D_o=7.6\text{mm} \) and \( 3.5\text{mm} \). However \( D_o=2.5\text{mm} \), the magnitude of the lowest velocity of the oscillating jet becomes significant.

To assist the comparison of jet performance, the ratio of the jet oscillation amplitude, \( \Delta u_{c,o} \), to its mean velocity, \( \bar{u}_{c,o} \), value is taken as a measure of the extent of the jet pulsation (modulation index). Here the amplitude of jet oscillation \( \Delta u_{c,o} \) is defined as the difference between the values of the high and low peak. In a fully pulsed condition the modulation index should correspond to \( \Delta u_{c,o}/\bar{u}_{c,o} = 2 \). The values of \( \Delta u_{c,o}/\bar{u}_{c,o} \) for different exit diameters are shown in Figure 5.1. At \( D_o=3.5\text{mm} \), \( \Delta u_{c,o}/\bar{u}_{c,o} \) is nearly two confirming that the jet is fully pulsed. At \( D_o=2.5\text{mm} \), however, \( \Delta u_{c,o}/\bar{u}_{c,o} \) is much less than 2. It appears that for the amplifier with an AR value of 4, the minimum exit orifice diameter for getting a fully pulsed jet is around 3.5mm, giving \( D_o/h \) of 0.43.
Figure 5.1 Instantaneous velocity variation measured at the centre and 1DO downstream of an amplifier exit orifice with bS=2mm and AR=4
(a). DO=7.6mm and DO=2.5mm, (b). DO=3.5mm
<table>
<thead>
<tr>
<th>$D_o$ (mm)</th>
<th>$\tau^*/\delta$</th>
<th>$\Delta \tau^<em>/\tau^</em>$</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.6</td>
<td>0.95</td>
<td>2.21</td>
</tr>
<tr>
<td>3.5</td>
<td>0.43</td>
<td>1.97</td>
</tr>
<tr>
<td>2.5</td>
<td>0.31</td>
<td>1.07</td>
</tr>
</tbody>
</table>

Table 5.1 The calculated values of the amplitude to time averaged velocity ratio for various $D_o$ values with $b_s=2$mm and AR=4

### 5.2 Effect of changing aspect ratio

Although it was found that the exit orifice diameter of 3.5mm can provide output signal without signal attenuation, this is still considered to be too large for the intended flow control purpose on the aerofoil. For the aerofoil flow control experiment, a jet orifice of the order of 1mm will be more desirable. It appears that the existing design of the fluidic amplifiers in an array cannot accommodate the expected exit orifice diameter. Therefore in order to achieve a further reduction in the exit orifice diameter the height of the amplifier is reduced from the original 8mm to 4mm, resulting a reduction of the nozzle aspect ratio from 4 and 2.

By reducing the AR from 4 to 2, the minimum exit orifice diameter at which an expected fully pulsed exit jet response achieved is found to be reduced from 3.5mm to 2.1mm. The temporal variations of the jet exit velocity for $D_o=2.1$mm and AR=2 are shown in Figure 5.2 and the comparison of relevant parameters for AR=2 and 4 for $D_o=2.1$mm and 3.5mm are given in Table 5.2.

![Figure 5.2 Velocity variations of output jet with DO=2.1mm and AR=2](image-url)

Figure 5.2 Velocity variations of output jet with $D_O=2.1$mm and AR=2
Although the effect of reduced exit conduit cross sectional area, $A_{\text{ref}}$, observed is in consistence with the description given by Kirshner (1966), $\frac{A_{\text{ref}}}{h}$ found from this study does not gave a constant value for AR=2 and AR=4. Further reduction in the height of the amplifiers has not been considered due to a concern regarding the possible change on the device performance caused by the increased boundary layer effect in the power jet nozzle. Nevertheless, $D_o=2.1\text{mm}$ is seemed a great improvement and an exit orifice of this size is considered acceptable for flow separation control experiments over a flap which has a thicker incoming boundary layer than on the aerofoil model.

<table>
<thead>
<tr>
<th>AR</th>
<th>$D_{O,\text{min}}$ (mm)</th>
<th>$D_o/h$</th>
<th>$\frac{A_{o}}{A_{\text{ref}}}$</th>
<th>$\frac{\Delta u}{\bar{u}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td>(h=8mm)</td>
<td>3.5</td>
<td>0.43</td>
<td>0.16</td>
</tr>
<tr>
<td>2</td>
<td>(h=4mm)</td>
<td>2.1</td>
<td>0.52</td>
<td>0.11</td>
</tr>
</tbody>
</table>

Table 5.2 The calculated values of the amplitude to time averaged velocity ratio at $D_{O,\text{min}}$ for different AR values of 2 and 4 with $b_S=2\text{mm}$.

5.3 Other observations

The instantaneous velocity measurements from a conjugating pair of exit orifices of an amplifier with an AR of 2 and $D_{o,\text{min}}$ of 2.1mm are plotted in Figure 5.3 a) during the supply mass flow rate to a supply chamber being 4.4g/s. This shows that there exists a minor difference between their pulse amplitude, which effectively gave the two different $\frac{\Delta u}{\bar{u}}$ values of 1.74 from the Set side and 2.05 from the Re-set side. This is believed to arise from the fact that the current fluidic amplifier having a multiple output ports.

The control signals to an array of amplifiers were generated by operating the fluidic oscillator at $Re_{D_h}$ of $\sim 3.1 \times 10^4$ with a feedback loop length of 1.0m, which was expected to generate a periodically oscillating control signals at a frequency of 83Hz from an each exit ports. However, the spectral analysis shows the actuation frequency of the amplifiers to be at 127Hz as in Figure 5.3 b). One possible explanation of the discrepancy noticed in frequency synchronisation would involve an extra output load introduced to the fluidic oscillator by an interconnection made
between its output ports and control chamber of the fluidic amplifiers. In Chapter 2 it was briefly described that the jet switching can be initiated by introducing a load on the active side output ports. Thereby an increased output load may have introduced an extra switching force to a fluidic oscillator in the form of returned flow from the high load exit port (expansion wave), which enhances the tendency of the fluidic oscillator for power jet switching compared to its normal operation condition.

Figure 5.3 The output jet characteristics of the fluidic amplifier model measured at the centre and 1DO downstream of a conjugating pair of exit orifices having DO of 2.1mm. a) Instantaneous velocity variation; b) Power spectral density. The supply mass flow rate to the supply chamber of an array is 4.4g/s with a fluidic oscillator running at ReDj of 3.1x10^4 with L=1.0m.
5.4 Summary of findings

- It is found that for the original fluidic amplifier with the nozzle aspect ratio of 4, a fully pulsed jet can only be obtained with $D_o/h$ ratio greater than 0.43.
- Decreasing the nozzle aspect ratio from 4 to 2 leads to further reduction of the minimum orifice diameter with which a full pulsed jet is obtained.
- $D_{O,min}/h$ ratio for a fixed power nozzle width was found to vary with a nozzle aspect ratio, i.e., $D_{O,min}/h=0.43$ for AR=4 and $D_{O,min}/h=0.52$ for AR=2. However, it can be stated that as long as $D_o/h$ ratio is higher than 0.52 a fully pulsed jet can be obtained with a AR larger than 2.
- A 180° phase difference has been observed from a conjugating pair of exit orifices of the fluidic amplifiers. A considerable difference in the pulsing jet magnitude between the 6 exit orifices of a single fluidic amplifier has been observed.
- A disparity between the oscillation frequency of the output jet from an array of fluidic amplifier and the actuation frequency of the fluidic oscillator, which is approximated from its power nozzle Reynolds number, has been observed. The discrepancy in the synchronisation of an actuation frequency is thought to be due to the extra output load introduced to the fluidic oscillator by the interconnection made with the control chambers of the array of fluidic amplifier.
6 Conclusions and future work

The objectives of this study are twofold, a) to examine the jet characteristics of a control loaded fluidic oscillator so as to gain an understanding of their fundamental performance parameters and b) to evaluate the design of the array of fluidic amplifiers and to propose necessary modification for their use on an aerofoil model for flow control. These set objectives have been achieved. The conclusions from this work are stated in this chapter which is followed a suggestions for future work.

6.1 Conclusions

6.1.1 Fluidic oscillator

The steady state analysis showed that the current fluidic oscillator model, in general, requires a control mass flow rate of about 7% of the supply mass flow rate to onset the power jet switching. This demonstrates that the existing model has a relatively high tendency for the power jet switching when compared to a usual value of 5~15%, which can provide broad oscillation frequency bandwidth in the output jet. However at a high Reynolds number, \( Re_{Dp} > 2.4 \times 10^4 \), a constant reduction in the relative control to supply flow rate ratio at the power jet switching point is noticed.

The trend of change and values of the flow gain with an increasing power jet Reynolds number are found to follow that of the inverse of the control flow rate. They were found to increase linearly above the critical power jet Reynolds number, while giving a relatively constant value of 14 at sub critical regime. This discontinuity noticed in trends of changes of the flow gain is likely caused by the limited growth of the standing vortex, which counteracts the control flow entering from the attached side control nozzle, confined by physically by the cavity forming the interaction region.

The fluidic oscillator model has successfully demonstrated a generation of a fully pulsed jet with a square waveform, having duty cycle of 50% and an 180° phase difference between its conjugating pair of output orifice. It was found that the cycle averaged output jet velocity can vary from ~8ms\(^{-1}\) to ~72ms\(^{-1}\) with a maximum of ~50% unsteadiness. The frequency bandwidth of ~10Hz up to ~185Hz was attainable when the shortest available feedback loop length of 0.5m is used for the
power jet Reynolds number range of $6.5 \times 10^3$ to $7.3 \times 10^4$. The maximum cycle peak velocity measured in this study was approximately $185 \text{ms}^{-1}$ and this falls within an acceptable range for practical separation control application.

The most obvious findings to emerge from this investigation are the dependency of the oscillation frequency of the fluidic oscillator on the supply flow rate and the feedback loop length. In general, the frequency of the output jet oscillation increases proportionally with the supply flow rate whereas it reduces inverse proportionally with the feedback loop length. The effect of increasing the supply flow rate is to increase the power jet entrainment of the fluid from its vicinity which in turn reduces the switching time, $\tau_s$, of the device. On the other hand, increasing the feedback loop length provides a larger volume of fluid for the entrainment. As a result of this, the development of a sufficient level of a differential pressure across the feedback loop which is required to onset the propagation of a control signals in the form of a pressure wave and a rarefaction wave is delayed, thereby $\tau_s$ increases. Also, increasing the feedback loop length effectively means the control signal has to travel longer distance and hence the control signal propagation time, $\tau_p$, increases accordingly.

In addition to this, at a power nozzle Reynolds number greater than $3.0 \times 10^4$ frequency saturation starts to occur regardless the length of the feedback loop, which gave a constant Strouhal number of 0.7. Also when the fluidic oscillator operated at low $\text{Re}_{D_h} \sim O(10^3)$ and/or small feedback loop length to power nozzle width ratio, $\xi, \sim 150$, a fluctuation in the pulse period have been noticed. The existence of an instability in the oscillation period at low $\text{Re}_{D_h}$ operation is thought to be due to the swirl flow occurring at an upstream of the power nozzle which tends to be suppressed as the supply flow rate increases.

The normalisation of the experimentally calculated switching time, $\tau_s$, by the theoretical control signal propagation time, $\tau_p$, has gone some way towards enhancing the understanding the principal cause of the saturation in the oscillation frequency of the fluidic oscillator. It is concluded that the oscillation period of the output jet is mainly governed by $\tau_s$ in the Reynolds number regime lower than $3.0 \times 10^4$ whereas $\tau_p$ becomes the major time parameter in the higher Reynolds
number regime as \( \tau_s \) gets smaller than the unity hence resulting a frequency saturation.

The output jet oscillation characteristics of the fluidic oscillator model studied in this report suggests its possibility to be used as an alternative PVGJ actuator for separation control. It produces unsteady output jet with a strong modulation and turbulence intensity of 10~50% of the jet mean velocity during the blowing phase. Also, the possibility of scaling of the current test model to a meso-scale structure promises its reliable augmentation as a flow control actuator for practical application.

6.1.2 The array of fluidic amplifiers

The results from an analysis of an array of amplifiers having a concentric reduction in their output ports demonstrate an existence of change in pulse characteristic in the output jet, which effectively shifted a cycle mean velocity to a higher level while retaining its amplitude. It is apparent by examining the diminishing of the transfiguration in the output jet as the exit orifice diameter is enlarged that the extra output load introduced by the concentric reduction in their output port is the cause of the jet attenuation noticed. It is found that a minimum orifice diameter of 3.5mm is required in order to ensure a fully pulsed jet with a minimum jet velocity near zero with changing the design.

It was also shown that the minimum output orifice diameter that can be accommodated by an amplifier without changing the output jet characteristics, can be reduced by reducing the height of the fluidic amplifier, i.e., the power nozzle aspect ratio. By reducing the aspect ratio of the amplifier from 4 to 2, the minimum orifice diameter is reduced from 3.5mm to 2.1mm. Although the results do not present a constant \( \frac{D_o}{h} \) ratio for the point where the jet attenuation becomes negligible, it can be stated that this ratio generally requires to be higher than 0.52 for the nozzle aspect ratios up to 2. An orifice diameter of 2.1mm is not ideal, but is seemed a great improvement and an exit orifice of this size is considered acceptable for flow separation control experiments over a flap which has a thicker incoming boundary layer than on the aerofoil model.

The design of an array of fluidic amplifiers tested in this study requires the use of a secondary control flow source in order to guarantee a phase difference/synchronisation between the neighbouring amplifiers/outputs due to the
indeterminable initial jet attachment in each fluidic amplifier. However, this removes one of the principal advantages of a fluidic device by requiring a secondary unit which in most cases requires mechanical moving parts to generate pulsing control signal. The results shows when a fluidic oscillator used as a driving unit the switching characteristic of the device can be changed based on the disparity in the actuation frequency for a given supply flow rate. Furthermore a disparity in the pulsing jet magnitude between the 6 output orifices of a single fluidic amplifier used in an array has been noticed when operated at a relatively low power jet Reynolds number.

### 6.2 Suggestions for future work

Although the existing model of the control loaded fluidic oscillator requires some design modification they seem to be a more suitable for the flow separation control application compared to a fluidic amplifier. The drawbacks of a fluidic amplifier would include an intrinsic necessity of a secondary driving element, a relatively large compartment space requirement arising from a necessity of control flow chambers and complexity in its interconnection and load matching between with a driving unit.

When adopting the existing fluidic oscillator for flow control application, the major modification required by the current model involves removal of its bistability in the power jet attachment. For this reason the author suggests to develop a monostable fluidic oscillator where its conceptual schematic drawing is shown in Figure 6.1 a). The monostability of this oscillator is achieved by the restriction introduced in the one side of the control port. A traditional method of achieving a monostability by having different offsets on each side of the power nozzle has not been considered due to a possible pulse characteristic difference, such as a duty cycle, between jets generated from a conjugating pair of output orifices. Its planform may follow that of the fluidic oscillator studied in this report. This idea can be preliminary demonstrated using the existing model by modifying the feedback loop accordingly.

Also for a laboratory demonstration of its capability in delaying flow separation, scaling of the device must be considered to have an acceptable size of the output orifice diameter and to improve its frequency response and oscillation stability at low power jet Reynolds number. Also it would be valuable to build each component of the test model of the fluidic oscillator as a separate module, as shown in Figure 6.1 b), which can allow a geometrical parametric study for performance optimisation.
and study of a scaling effect. In addition to those, by separating them into several modules the complication in manufacturing the interaction region of possibly a microscale model can be realised without using an advanced manufacturing technique. In the development stage of the suggested model, the input and output characteristic analysis using the methods described in Chapter 2 can be incorporated and supported by an experiment if required.

Once the model has been developed and its jet characteristics have been verified, the effectiveness of an array of monostable fluidic oscillators in delaying the separation of turbulent boundary layer will demonstrated over a flap using a surface flow visualisation.

![Diagram](image)

**Figure 6.1 Conceptual drawing of a monostable control loaded fluidic oscillator.**

- a) Planform of the fluidic oscillator;
- b) 3D showing the modules forming the fluidic oscillator.
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