Experimental Investigation of a Bi-Stable Supersonic Fluidic Oscillator

Sichang Xu
University of Windsor

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Experimental Investigation of a Bi-Stable Supersonic Fluidic Oscillator

By

Sichang Xu

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Experimental Investigation of a Bi-Stable Supersonic Fluidic Oscillator

By:
Sichang Xu

APPROVED BY:

______________________________________________
P. Henshaw
Department of Civil and Environmental Engineering

______________________________________________
A. Fartaj
Department of Mechanical, Automotive and Materials Engineering

______________________________________________
G. Rankin, Advisor
Department of Mechanical, Automotive and Materials Engineering

January 11, 2018
Declaration of Originality

I hereby certify that I am the sole author of this thesis and that no part of this thesis has been published or submitted for publication.

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Abstract

Fluidic oscillators are non-moving part fluid valves that generate a self-sustained, stable oscillating flow and pressure. This research involves an experimental investigation of one geometrical configuration of bi-stable Supersonic Fluidic Oscillator (SFO). The experiments are conducted over a range of supply pressure, control channel flow resistance and exhaust chamber pressure values for the purpose of determining their effect on the device oscillation frequency and amplitude. High-speed Schlieren videos of the internal flow field were made and synchronized with experimental pressure values taken at strategic locations, to provide insight into the device switching process. The experimental results were also used to validate a computational fluid dynamics model and conversely, the model was used to better understand the mechanisms responsible for the oscillation.
Dedication

I would like to dedicate this work to Ye Xu, my dearest dad and best friend, and to Jingchun Luo, my dearest mom who gives me the strong and warm support to my life.

I would also like to dedicate this work to my professor Dr. Gary Rankin, not only for his great patience in my academic study, but also for his tender care to me when I was facing serious challenges in my life.
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Nomenclature

A  Area
A*  Throat Area
A_{orifice}  Orifice Discharge Area
C  Constant
c  Control Channel Resistance Coefficient
C_p  Specific Heat at Constant Pressure
C_v  Specific Heat at Constant Volume
D  Plug Hole Diameter
e  System Energy per Unit Mass
F  Dimensionless Frequency
f  Feedback Tank Pressure Frequency
g  Acceleration due to Gravity
h  Specific Enthalpy
k  Specific Heat Ratio
ṁ  Mass Flow Rate
Ma  Mach Number
n  Polytropic Index
N  Number of Cycles
P  Local Static Pressure
P_i  Static Pressure Associated with One Cycle
ΔP  Change of Local Static Pressure
P_{atm}  Atmospheric Pressure
Pe  Entrance Pressure
P_{ec}  Exhaust Chamber Pressure
P_{fbamp}  Feedback Tank Pressure Amplitude
P_{fbamp,max}  Maximum Feedback Tank Pressure Amplitude
\mathcal{P}_{fb\amp}  Dimensionless Feedback Tank Pressure Amplitude
P_o  Total (Stagnation) Pressure
P_s  Supply Pressure
P_{sll}  Lower Limit of Supply Pressure
P_{sul}  Upper Limit of Supply Pressure
P_t  Tank Volume Pressure
Q  Heat Loss
R  Gas Constant
s  Minor Loss Coefficient
SPF  Supply Pressure Fraction
Sx  Standard Deviation
T  Local Static Temperature
T_t  Tank Volume Temperature
t  Time
Δt  Period of One Time Step
t_{95\%}  Student’s t Value with 95% Confidence
To  Total (Stagnation) Temperature
Ts  Supply Temperature
T_t  Target Volume Temperature
U  Uncertainty
u  Specific Internal Energy
V  Volume
v  Flow Velocity
V_{fb}  Feedback Tank Volume
W  Work
Z  Head Loss
α  Orifice Discharge Coefficient
ρ  Fluid Density
Chapter 1 Introduction

Fluidic devices were originally developed in the 1960’s [1] to perform the functions of sensing, control and logic necessary for the operation of pneumatic actuators and motors which provided the fluid power to perform the desired task. The original application of fluidic logic circuits failed due to the advance of electronic logic devices. However, interest has been renewed in flow control applications especially at the micro-scale level [2–4]. A category of fluidic devices simultaneously evolved called “power fluidics” in which the fluid passing through the device provides the power required to perform the desired task [5].

A unique advantage of any fluidic device is that it has no moving parts, which provides reliable working performance with robust design. By utilizing certain flow instability features and a feedback mechanism, a fluidic device can generate periodic self-sustained oscillations. This type of device is called a fluidic oscillator.

The bi-stable supersonic fluidic oscillator is one category of such device, and the main focus of this thesis. This oscillator uses a convergent-divergent nozzle to create a supersonic jet flow that separates from the nozzle walls and attaches to one of the two sidewalls as a result of the Coanda effect. The effect is bi-stable in that attachment can be made to either of the two sidewalls. The feedback mechanism used to switch the jet attachment between walls depends upon the specific oscillator design.

This chapter gives a brief introduction to the fundamental principles of supersonic flow separation and the Coanda effect. The major types of bi-stable fluidic oscillators that have been developed are also introduced in this chapter.
1.1 Supersonic flow separation in a divergent region

A subsonic flow is accelerated to supersonic speed through a convergent-divergent nozzle, and will choke with a Mach number of one at the nozzle throat. The mass flow rate for the isentropic inviscid flow through the nozzle can be calculated using Equation 1, and only depends on the stagnation upstream conditions, type of gas and the throat area.

\[
\dot{m} = \frac{A^*P_0}{\sqrt{T_0}} \sqrt{\frac{k}{R}} \left( \frac{k+1}{2(k-1)} \right)^{\frac{k+1}{2(k-1)}}
\]  

Equation 1

In the case of supersonic flow, there is no effect on the upstream main flow from the downstream condition since the feedback signals (pressure waves traveling at the speed of sound) cannot travel upstream fast enough compared to the downstream flow speed. However, in the near wall region of a viscous flow, the velocity in the boundary layer is much slower than the main flow jet, thus allowing the downstream conditions to be transmitted upstream.

![Diagram of supersonic flow separation in inviscid and viscid flow](image)

Figure 1.1 Supersonic Flow Separation in Inviscid and Viscid Flow

For the inviscid flow, a normal shock can occur in the diverging portion of the nozzle or at the nozzle exit when the ratio of the total pressure before the shock to the
back pressure is in a certain range. However, if the same pressure condition is kept for the viscous flow, the boundary layer may not be able to overcome the large pressure change due to the normal shock wave, and separate from the nozzle walls. The flow separation leads to an inward direction change to the main flow jet and oblique shocks occur as a consequence, as shown in Figure 1.1.

The relationship between the turbulent supersonic flow separation location and the upstream-downstream pressure conditions in the nozzle can be deduced from the empirical relationship given in Equation 2 [6],

$$I \approx \frac{800 + (L - 1.05)^2 (2k - Jk - 1)}{800 + (L - 1.05)^2 (k - 1)}$$

where $I$ is the ratio of the the stagnation supply pressure to the local static pressure just before the oblique shocks, and $J$ is a constant found empirically as 0.55. $L$ is the ratio of back pressure to stagnation supply pressure.

![Figure 1.2 I versus L](image)

If the fluid is accelerated to supersonic speed after the nozzle throat and the back pressure is held constant, a larger stagnation supply pressure results in a smaller value of $L$. According to Figure 1.2, this corresponds to a larger value of $I$. For supersonic
isentropic flow, $A_{local}/A^* \propto I$ as shown in Figure 1.3, and hence a larger cross-sectional area occurs at the separation point which means that it is further downstream because of the diverging area in the flow direction.

Figure 1.3 Values of I and $A/A^*$ versus Mach number

1.2 Coanda effect

When the supersonic flow separates from the divergent wall, a mixing region is formed between the boundary of the jet and the wall. The velocity of fluid in this region is much slower than the main jet and includes flow reversal. As the supersonic flow passes the mixing region, a part of the surrounding fluid is entrained and thus accelerated. This leads to a pressure decrease in the mixing region [7].
If supersonic flow passes through a two-dimensional divergent region as shown in Figure 1.4, and separates from both the upper and lower divergent walls, minor asymmetries make the pressure decrease to be different in the upper and lower mixing regions. This difference leads to a transverse pressure gradient across the supersonic jet with a lower pressure on the side with less entrainment. As a result, the jet will deflect to the lower pressure side. This deflection further increases the transverse pressure gradient. The adverse pressure gradient in the flow direction along the wall of the lower pressure side is decreased due to the local pressure drop, thus the separation point at that side moves further downstream as indicated in Figure 1.5. The opposite phenomenon occurs in the region on the unattached side; the separation point moves to an upstream location due to the larger local pressure increase. Those changes further enhance the asymmetry in the flow field and ultimately the main jet reattaches to the wall of the lower pressure side. The region enclosed by the flow separation and reattachment is called the vortex region or the reattachment bubble. The pressure distribution in the vortex region is approximately constant.
If the device is geometrically symmetric, the jet remains attached to the initial side until some control action causes it to switch to the other output channel and remain in that stable position until a similar but opposite control action switches it back. This type of device is referred to as a bi-stable fluidic switch or amplifier. The term amplifier is appropriate since a low power control jet switches a higher power main jet. The control actions can take many different forms as will be discussed later. If the switch to one side causes a return switching action (feedback) a self-sustained oscillation results. Devices of this type are called bi-stable fluidic oscillators.

1.3 Introduction to major types of bi-stable fluidic oscillator

Most bi-stable fluidic oscillators, such as the one shown in Figure 1.6, are designed with five major components in a planar symmetric layout: the reservoir, the nozzle, the expansion cavity, a pair of outputs at a divergence angle and a pair of control channels attached to the expansion cavity near the nozzle exit. The depth into the page is usually a constant. The reservoir supplies steady flow which is accelerated by the nozzle and separates from the walls of the expansion cavity forming a jet. Due to the Coanda effect, the flow reattaches and passes through one of the outputs. In most cases, the
outputs are formed by placing a splitter at the middle of the diffuser. Most designs also have symmetric channels attached to the divergent walls at the locations near to the vortex regions, to be used for controlling the flow reattachment. These channels are called the control channels.

To maintain the oscillation in the bi-stable fluidic oscillator, it is necessary to cause the jet flow in the sensitive region downstream from the nozzle, to continually switch from one channel to the other and back again. Transverse pressure gradients and momentum changes are two of the mechanisms capable of causing these switches. Based on different control logics, the oscillators can be categorized into four types: the resonance oscillator, the sonic wave oscillator, the relaxation oscillator, and the load oscillator. Figure 1.7 gives a brief illustration of the four bi-stable fluidic oscillator types.
The resonance oscillator as shown in Figure 1.7(A), uses travelling pressure waves in the control channel to sustain the oscillation [8]. As the flow attaches to one side, it sucks air from the control channel on that side and creates an expansion pressure wave. The expansion wave travels to the end of the control channel and reflects as a compression wave which travels back to the jet to increase the pressure at the vortex region and causes the jet to switch and attach to the other side. The performance of the resonance oscillator is determined by the design of the control channels. The length of the control channel and the speed of the waves determine the oscillation frequency. This configuration has been applied as an active flow control actuator for enhancing the performance of aircraft and fluid-flow machinery.
Another approach which uses expansion waves to cause oscillation is to make a loop between the two control channels as illustrated in Figure 1.7(B) [9]. The expansion waves travel along the loop channel and reduce the pressure on the unattached side to reverse the transverse pressure gradient and make the flow switch. This type of the oscillator is referred to as a sonic wave oscillator. Like the sonic wave oscillator, the flow switch and oscillation frequency are controlled by the layout configuration and the dimensions of the control loop.

Like the resonance oscillator and the sonic wave oscillator, the relaxation oscillator changes conditions in the vortex region to maintain oscillation. The relaxation oscillator not only uses the pressure waves, but also a part of the main flow momentum to interact with the jet and vortex region as indicated in Figure 1.7 C. In most of the relaxation oscillators, the sub-branches are connected near the outputs which allow a portion of the mass flow return to the control channels. The resulting pressure waves and the returned mass flow with certain momentum will then travel back to the main flow jet at the vortex region and force it to the unattached side. The performance of the relaxation oscillator is determined by the wave propagation speed and the momentum of the recycled flow, which is related to the location, dimension, orientation and resistance of the sub-branches [2].

The load oscillator is different in that it is the interaction of the flow through the output channel with the vortex region that causes the switch. In this case there is some resistance or load which increases the downstream pressure within the attached output channel. This leads to a local force reducing (opposing) the flow momentum [10, 11] as shown schematically in Figure 1.7 (D). Once the downstream pressure builds up to a
certain level, the velocity in that output channel is reduced. The velocity reduction results in the pressure increase within the vortex region at the attached side which reverses the transverse pressure gradient and forces the flow to switch. After the switch, the built-up downstream pressure dissipates, and the same process occurs on the opposite side of the oscillator. For the load oscillator, the speed and efficiency of the pressurization and dissipation process determine the amplitude and frequency of the oscillation.

One way of causing an increased load is the fluidic micro-bubble generator shown in Figure 1.8. There is no control channel in this design and gas is the working fluid. Each output channel of the oscillator is tangentially connected to a circular chamber with an exhaust hole. An aerator immersed in liquid with tiny outputs is connected to the oscillator exhaust.

![Diagram of the Load Oscillator as a Micro-bubble Generator](image)

Figure 1.8 The Load Oscillator as a Micro-bubble Generator [10, 12]

As the flow attaches to one side, the downstream pressure increases due to the vortex being formed in the chamber attached to that output. This pressure increase pushes the working gas against the external liquid in the aerator and through the tiny outputs
where it is exhausted into the liquid. Meanwhile, the increase of pressure in the chamber ultimately forces the main oscillator flow jet to switch. After the flow switch, the pressure in the chamber starts to dissipate and the liquid in the aerator pushes the gas back. The small amount of gas exhaust through this process becomes a micro-bubble in the liquid. The device can be tuned to control the size and frequency of bubble generation.

1.4 Motivation for the current research

The motivation for this work is an expressed need by industry for a robust method of generating pressure fluctuations of a certain amplitude and frequency in a pressure vessel under extreme conditions of temperature which preclude the use of conventional pneumatic spool-type control valves. The specific industrial application cannot be revealed for proprietary reasons. As the operating fluid is performing the desired work, the oscillator to be considered in this thesis is in the class of a power fluidic oscillator. The absence of moving parts eliminates the possibility of failure at high temperatures due to parts seizing. The oscillating flow momentum exiting each output port is used to generate pressure pulsations of large magnitude and a certain frequency while filling a finite volume space. The large pulse amplitude requires a large output momentum, which is function of supply mass flow rate. A supersonic flow jet is necessary to give the large value of momentum that is necessary. For the application considered it is desirable to be able to adjust the oscillation frequency independently with the mass flow rate which requires the consideration of design parameters other than just the upstream supply pressure. The next chapter includes a review of the literature related to supersonic fluidic oscillators and ends with a presentation of the specific objectives of this thesis.
Chapter 2 Literature Review and Objectives

This chapter begins with a review of some early experimental studies associated with the geometrical design of supersonic fluidic amplifiers. No attempt has been made to include all previous literature, especially research papers concerning subsonic fluidic oscillators, except those which illustrate features common to the supersonic variety. These papers serve to illustrate the importance of various geometrical and operating parameters on the performance of a supersonic bi-stable fluidic oscillator. Next, the supersonic bi-stable fluidic oscillator developed by Hiroki[11] is introduced. It is explained why this design is unique and important regarding this thesis. Some numerical studies of the bi-stable fluidic oscillator designs are then reviewed to show the usefulness in associating the numerical approach with the experimental studies. A CFD model based on Hiroki’s oscillator design that was developed in our laboratory is introduced since it is used later in this thesis to aid in the interpretation of the experimental work reported. Finally, the objectives of this thesis are presented.

2.1 Early experimental studies concerning bi-stable supersonic fluidic amplifiers

Research regarding bi-stable fluidic devices started in the 1960’s, inspired by their robust and reliable operation in fluidic logic circuits. Most studies focused on understanding the fluid motion and pressure variation during the operation.

F.G.Bavagnoli experimentally studied the characteristics of a supersonic bi-stable amplifier (switch) similar to the basic design described in the previous chapter [13]. In this design, the control channel opening is located near the vortex region and the flow switch is achieved by manually opening or closing valves on the control channels. Four
different divergent nozzle angles were investigated with a constant throat area and throat to splitter distance. The locations of control channel vents are also the same and the amplifiers exhaust to the atmosphere. For a constant supply pressure, a smaller divergence angle leads to a longer distance to the vortex region from the throat indicating that the flow separates further downstream. In the case of larger divergence angles, the location of the separation point is less sensitive to the supply pressure change, and the flow switch can be achieved within a larger range of supply pressure. For a fixed divergence angle, as the pressure is increased, the flow continues to be attached to one side, but the vortex region is too far away from the control port, and the switching process is not achieved. If the supply pressure is further increased, flow separation will be symmetric with no jet bending, and hence the Coanda effect does not occur.

F.G.Bavagnoli also measured the pressure in the control channel. As the flow attaches to one side, the pressure at the control channel entrance on that side decreases due to the entrainment of the main flow jet. When the control channel is opened, the pressure in the control channel on the attached side gradually increases until the flow switches. It is found that the response time of the switch process is only about 1 to 2 milliseconds.

R.V.Thompson conducted experiments to study the effect of the back pressure on the switch process of a supersonic bi-stable fluidic amplifier [14]. The amplifier is designed as a basic bi-stable fluidic configuration without the control channels. A shutter is installed at one end of the output, and a pressure relief valve is installed at the end of the other output. The flow is restricted by the shutter, so the back pressure is built up and ultimately forces the switch of the flow jet. The value of the largest back pressure is set
by modifying the discharge of the pressure relief valve. The switch process is achieved by gradually opening and closing the shutter. For three constant values of supply pressure, the locations of the flow separation and reattachment are measured with different back pressure conditions by using flow visualization. It is found that a larger back pressure permits a larger range of supply pressure to achieve the flow switch with the stable flow attachment. With a constant supply pressure, when the flow is stably attached to one side, increasing the back pressure decreases the size of the vortex region and moves the vortex region upstream. The dynamic characteristics of the bi-stable supersonic fluidic amplifier driving a pneumatic piston actuator were also considered however, that work has little to do with the current study and is not discussed here.

In a later work, R.V Thompson experimentally investigated the effect of the geometry on the performance of the supersonic bi-stable fluidic amplifier [7]. The adjustable design parameters for the study are the divergence angle, the control channel width, the divergent wall length and the splitter position. He determined that an increase in the divergence angle decreases the sensitivity of the location of the vortex region to changes in the supply pressure, thus increasing the possible working range of the amplifier. This result agrees with that of F.G.Bavagnoli [13]. The pressure in the vortex region is smallest when the divergence angle is 40 degrees giving the strongest attraction to the main flow which provides the most stable operation. He concluded that the length of the divergent walls must be long enough to permit the vortex region to form on the walls. It is suggested that the minimum wall length exceed the reattachment distance by at least 20%. The position of the reattachment point is not significantly affected by the divergence angle but depends on the throat area and the ratio of the supply pressure to the
back pressure. The tip of the splitter is suggested to be placed on the nozzle center line at a distance from the nozzle exit which is close that of the flow reattachment point.

Empirical data from early studies of supersonic bi-stable fluidic amplifiers helps in understanding the fluid flow phenomena in supersonic fluidic devices such as flow separation and switching. They also give indications regarding changes in important operating parameters, such as supply pressure, control resistance, back pressure and geometrical effects which help in understanding the operation of bi-stable fluidic oscillators.

2.2 Review of the experimental and mathematical studies for the supersonic bi-stable fluidic oscillator

F. Hiroki designed a high power supersonic bi-stable fluidic oscillator for use in a metal fatigue test apparatus which produces a large amplitude periodical bending deflection of a metal plate [15]. This test facility has the advantage of a long-life usage without maintenance due to the absence of any moving parts. The geometry of the prototype is designed based on the early work of the supersonic bi-stable amplifier by R.V Thompson [7]. This oscillator is of the load bi-stable fluidic oscillator type shown in Figure 2.1. Each output channel is “loaded” by restricting a part of the output flow to pass into an enclosed tank, called the feedback tank, which increases in pressure. This process continues until the pressure in the feedback tank is propagated upstream to the jet exit and reaches a value sufficiently large to force the flow to switch to the other output channel. The same process then repeats in the opposite feedback tank and output channel and a self-sustained oscillation occurs.
Experimental results are reported for different supply pressures, feedback volumes and conditions of control port opening. Pressures are recorded at the control channels, the feedback tanks and the junctions of the exhaust channel with the channel to the feedback tanks. Stable oscillation occurred with different ranges of supply pressure for the open and closed control port conditions. For open control ports and a constant feedback tank volume, a higher supply pressure gives a larger oscillation frequency and a higher peak value of the feedback tank pressure. By increasing the feedback tank volume and a constant supply pressure, the oscillation frequency decreased.

J.D McGeachy developed a mathematical model for estimating the performance of a supersonic relaxation type bi-stable fluidic oscillator [16–18] and compared its predictions with his own experimental results. The design of the oscillator is shown in Figure 2.2. When the flow is attached to one side, part of the outflow goes into the feedback port to pressurize the side chamber and cause flow to be released back to the
vortex bubble. On the unattached side, the air is drawn out of the side chamber into the main flow jet. The flow attachment is stable until the pressure difference between the two side chambers reaches a certain value, and the flow then switches to the other side. The device however, does not have a splitter like the traditional oscillator.

![Diagram of McGeachy’s Bi-Stable Fluidic Oscillator](image)

The mathematical model involves a quasi-steady flow assumption, a one-dimensional model for the channel flows as well as approximate equations for the entrainment and reattachment of the main jet. The assumption that switching occurs when the net back-flow from the atmosphere is zero at the outlet port, resulted in reasonable predictions of the channel pressures. The flows entering and leaving the chambers were not accurately predicted which led to frequencies which were only within one order of magnitude compared to the experimental values. The results however, give a similar trend in the variation as the experimental data. Throughout one cycle, the pressure in the bubble region gradually decreases during the stable attachment and detachment process and rapidly increases during the switch process.
2.3 Numerical studies of supersonic fluidic oscillators

Although some work has been reported in literature [19–21] regarding computational fluid dynamic models for fluidic oscillators in general, only a few have considered the supersonic fluidic oscillator [22, 23]. The model presented in Xu et al.[23] was initially created in our laboratory by J.P Martins, independent of his M.A.Sc. thesis. It was developed in connection with an NSERC Engage Grant, for comparison with experimental data of Hiroki [15]. The agreement is shown to be quite good in certain cases showing the same trend of decreasing frequency with increase in feedback tank volume as the experiments. No comparison however, could be made for a higher supply pressure, as the numerical model did not predict an oscillation under those conditions. It was concluded that further experimentation is necessary to determine whether the discrepancy is due to a deficiency in the model or a lack of detail in the description of Hiroki’s experimental operating conditions.

The model has been further refined in connection with the current research however, it is not considered to be the main thrust of this thesis. As important details of the technique are not documented in any other report, they have been included as Appendix 1 in this thesis.

2.4 Objectives of this thesis

The main objective of this thesis is to experimentally investigate the flow field and associated operating characteristics of a supersonic bi-stable fluidic oscillator in the regime of stable oscillation. The information obtained will aid future oscillator design and help formulate a corresponding analytical model. The experiments are conducted in
the Jet/Vortex Lab at the University of Windsor’s Centre for Engineering Innovation. The specific objectives are listed as follows:

1. Construct an experimental oscillator corresponding to the existing numerical model of Hiroki’s supersonic fluidic oscillator design that has been developed within our laboratory.

2. Arrange and calibrate the Schlieren and high-speed photography system to allow visualization of the flow within the oscillator channels.

3. Analyze the images of the flow field and compare with the numerical results to identify the flow features that occur during the oscillation cycle.

4. Investigate the effect of supply pressure, control channel resistance and exhaust chamber pressure on the oscillator performance and operation.
Chapter 3 Experimental Apparatus and Procedure

This chapter begins with a description of the design of the experimental prototype supersonic bi-stable fluidic oscillator, followed by a description of the test facility including the data acquisition and the Schlieren systems. The procedure for one experimental trial and the operating conditions for all trials are then given. Finally, the post-processing techniques applied to the raw data are presented.

3.1 Design of the supersonic bi-stable fluidic oscillator

The design of the prototype includes three major parts: the oscillator internal geometry, the oscillator body and the accessories, and are introduced in the following sections. Electronic copies of the design drawings are contained on an accessory CD kept with the hard copy of the thesis in the Mechanical, Automotive and Materials Engineering Department.

3.1.1. Details of the fluidic oscillator geometry

Efforts are made to reproduce the geometry of Hiroki’s oscillator as closely as is possible. Because of a lack of detailed information, some parts of the final design are approximated, based on the description in Hiroki’s paper. The final design of the flow channels within the oscillator is shown in Figure 3.1.

The semi-rectangular region on the left of the figure is the supply reservoir of the oscillator. This reservoir must be large enough to reduce any flow disturbances caused in the pipes supplying the device. To the right of the reservoir is the convergent-divergent nozzle which is formed by two curved walls. The radius of the walls in the convergent
regions is 18.7 mm and in the divergent region is 23.7 mm. The throat width of this nozzle is 3 mm.

Figure 3.1 Design of the Oscillator Flow Region (Units: mm)

At the exit of the nozzle is the expansion region, which is connected to two control channels and the two output channels. The control channel flow axis is at an angle of 25 degrees from the vertical direction and is 44.58 mm in length and 6.5 mm in width. The two output channels are formed by putting a flow splitter at the middle of the expansion region. The distance from the throat to the tip of the splitter is 29 mm. The divergence angle of the output channels from the horizontal centerline is 20 degrees. The length and width of the output channels are 142.1 mm and 9.9 mm respectively.

Each of the output channels ends in two branches. The ones parallel to the output channels lead to the feedback tanks and are 40 mm long and 5 mm wide. The curved branches from the output channels are connected to the exhaust ports, having a 20.1 mm inner radius and 9.9 mm width. The distance between the two exhaust ports is 40 mm.
To prevent flow from one exhaust interfering with the other one, a bar of 8 mm width is placed at the mid-point between the two exhaust ports. This bar also simulates the metal piece in Hiroki’s work.

The entire flow region is patterned onto a 9.6 mm thickness, 6061 aluminum plate. It is known that the supersonic flow is critically sensitive to the nozzle design, including the nozzle wall roughness, the nozzle convergence and divergence angles, and the width of the throat. At the same time, this oscillator is sensitive to any asymmetries in the flow field. Hence, it is very important to manufacture the flow region accurately. For satisfying the requirements, the flow region is manufactured using wire-cut electrical discharge machining (EDM) technology.

3.1.2. Design and installation of the oscillator body

There are two important requirements for the design of the oscillator body. One is to prevent any leakage of fluid from the oscillator body other than from the exhausts or the control channels. The other one is to provide a clear path for the light to go through the flow field, to allow flow visualization during the experiment.

To achieve these two requirements, a five-layer “sandwich” arrangement is used. The flow region introduced previously (see Figure 3.1) is cut from a 9.6 mm thickness, 6061 aluminum plate. This plate is then enclosed between two 12.6 mm thickness flat optical acrylic plates. These plates have overall outside dimensions of 410 mm length and 210 mm width.

The total light transmission efficiency of the optical acrylic for light striking perpendicular to the surface is approximately 92% for plates 6 mm in depth, which is as good as the fine optical glass [24]. The maximum tensile strength and flexural strength of
optical acrylic are 72.4 and 110 MPa respectively. These properties of optical acrylic ensure the feasibility of the flow visualization and the safety of the experiment under a high-pressure condition.

Two 9.6 mm thickness 6061 aluminum plates are placed on the outside of the previous three layers. The main purpose of these two outside layers is to protect the acrylic layers from distorting and causing leaks. Another purpose is to use those two layers as the adapters for the installation of sensors and other accessories. The outer layers are also used to locate and anchor the splitter on the middle plate, which is “suspended” in the flow region. The center region corresponding to the splitter is connected to the main frame by two 10 mm wide bridges. The two outer layers are cut into the geometry as shown in Figure 3.2.

![Figure 3.2 Design of the Outer Layers (Units: mm)](image)

The five layers are held together using 67 pair of zinc plated steel bolts and nuts. To fix the position of each layer, four pins are used at the corners of the oscillator, and two pins are located in the splitter region. A perspective view of the oscillator body assembly is shown in Figure 3.3.
3.1.3. Design and installation of the oscillator accessories

There are four major accessory components associated with the oscillator: the feedback tanks, the exhaust chamber, the control channel resistance plugs and the air supply system.

The feedback tanks are directly connected to the flow region and formed in three parts: the adapter, the tank body, and the tank cap. All three parts are cylindrical in shape and made of aluminum. The threaded male ends on the adapter parts are fitted into the female ends on the acrylic layers. The threads used here are ¼ inch British Standard Pipe Parallel thread (BSPP), which can provide excellent fluid sealing, but do not have a taper
that may lead to fracture of the acrylic layers. A rubber O-ring is applied at the end of each threaded head for further enhancing the sealing performance.

Each adapter is welded onto a cylindrical tank body with two threaded outputs on the tank walls. One is for installing the pressure transducer and the other one is not used in this study. The inner diameter of the cylinder body is 50 mm and 100 mm in length. The wall thickness of the tank body is 5 mm based on consideration of the inner tank pressure and the welding feasibility. The total fluid volume of each feedback tank is approximately 200 ± 1.7 cc. Figure 3.4 shows one of the feedback tanks. To prevent interference of the tanks with other accessories and the sensors, the two feedback tanks are placed on opposite sides of the main body.

Figure 3.4 Design of the Feedback Tank (Units: mm)
The exhaust chamber is attached to the exhaust end of the oscillator. It provides an enclosed volume to allow the pressure in the chamber to be increased during the experiment. The chamber body is made of 6061 aluminum and formed by one center U-shape plate with two side plates. There are six threaded holes on the center portion, which are used for installing sensors and valves on the chamber to monitor and adjust the exhaust chamber pressure. O-rings are installed in the sockets on the side plates and the edge of the center part for sealing purposes. Figure 3.5 shows the exhaust chamber and its installation. The total fluid volume of the exhaust chamber is approximately 400± 3.4 cc.

Figure 3.5 Design of the Exhaust Chamber (Units: mm)
The control port resistance is to be set by using plugs installed at the entrances of the control channels (control ports). There are three pairs of \( \frac{1}{4} \) NPT plastic plugs with center drilled holes of diameters of 9.12, 7.54 and 6.78 mm to give different values of resistance (discharge coefficient). The length of each plug is 24 mm. The selection of the drilled hole diameter is limited by the plug dimension and the sizes of the available drill bits. Figure 3.6 shows one pair of the plugs.

![Figure 3.6 Control Channel Resistance Plugs](image)

The air supply system connects the laboratory shop air supply to the oscillator supply reservoir. The shop air supply can only provide a constant steady pressure up to a value of approximately 4.5 MPa. The air flow is manually varied by opening/closing a valve which allows the air to pass through a pressure regulator (Arrow Pneumatics R3910G, 6.9 to 861.8 kPa gage) and hose arrangement into the oscillator. The air supply system is schematically showed in Figure 3.7.
3.2 Data acquisition system

Seven OMEGA MM-series pressure transducers are used in these experiments. One gage pressure transducer (Transducer A, MMG150V5B3MC0T4A5CE) with 1 MPa range is placed at the reservoir for recording the supply pressure. Two compound pressure transducers (Transducer B and C, MMCG005BIV5K4MC0T2A3CE) are connected to pressure taps provided in the acrylic plates and located at the middle of the control channels. These are used for monitoring the control channel pressure and have ranges of ±35 kPa. These three pressure transducers are installed on the acrylic layers with ¼ inch BSPP thread to minimize the possibility of acrylic fracture. They are placed on opposite sides of the oscillator to avoid spatial conflict. Two compound pressure transducers (Transducer D and E, MMCG015BIV5K4MC0T4A5CE) with ±100 kPa range are installed on the feedback tanks for recording the tank pressure. Another compound pressure transducer (Transducer F, MMCG015BIV5K2C0T4A5CE) is installed on the exhaust chamber for recording the pressures within a range of ±100 kPa. All the pressure transducers have less than a 0.001 second response time. The measurement uncertainty analysis of the sensors is shown in Appendix 2.

The location of the pressure transducers is shown in Figure 3.8. All sensors are connected to the NI PCI 6251 data acquisition board [25]. The digital data is taken at a rate of 10,000 samples per second and transferred to a custom LabVIEW 2014 program in the computer where they are monitored and stored. The details of the custom LabVIEW 2014 program are shown in Appendix 3.
3.3 Schlieren system

As light passes through a compressible flow field, the amount of light deflection of each ray depends on the density gradient across the ray. If the light is focussed using a lens or mirror and a filter blocks part of the refracted light from the beam in the focal region, the image that results will have a variation in brightness, the gray scale level of which is proportional to the density gradient in the flow field. This flow visualization technique is called the Schlieren method [26]. Multiple images captured by such a system in an unsteady compressible flow field at successive instants of time show the history of the density gradient change related to the fluid motion.
The layout of the Schlieren apparatus used in this experiment is shown in Figure 3.9. The light emanates from a Cree Xlamp XM-L LED source and passes into a converging lens of 460 mm focal length. The light beam converges at the focal point and passes through a spatial filter, which removes undesired aberrations and gives a more uniform intensity in the beam. The filtered light then diverges and passes through a collimating lens with a 175 mm focal length causing the light rays to be parallel. The parallel beam then passes through the oscillator body, which generates refracted light due to the density gradient variation throughout the flow field. The parallel beam is then converged by a de-collimating lens with a focal length of 200 mm and focused onto a knife edge to remove part of the refracted light. Lastly, the modified beam enters the FASTCAM Mini UX100 camera and images captured at a certain frame rate. All the components are installed on a rigid optical beam and adjusted to ensure a straight and consistent optical axis.

![Figure 3.9 Schlieren System](image)

In this experiment, visualization is of interest within the region immediately downstream of the nozzle (Coanda region) where changes in the flow drastically affect the oscillator performance. The parallel beam is adjusted to give a circular viewing area,
4 cm in diameter, within this region. The motion in the flow field is assumed to be limited to the plane containing the output and control channels with little change in the depth direction. If changes do occur in that direction they are averaged.

The knife edge of the Schlieren system is adjusted to obtain an image with a gray scale level suitable for distinguishing density gradients within the supersonic jet exit region. The knife edge adjusting process is achieved by a trial and error method in which the knife edge is adjusted manually. Ideally, the Schlieren image gives a constant brightness in a gray scale if there is no motion in the flow field. Due to the minor deflections and reflections involving the aluminum layers of the test oscillator, some non-homogeneity in brightness is observed in the resulting image. Fortunately, the undesired distortions only occur near the control channel boundary walls and thus give little obstruction to observation of the main flow jet.

The FASTCAM Mini UX100 camera is controlled by the Photron Viewer Version 3.6 software, which can be used to set the exposure time (60 usec), frame rate (10,000 fps) and resolution (640 *320) for Schlieren image capturing [27]. The software is also used to set the START capturing mode of the camera. Recording is started by a digital trigger signal and automatically stops after a certain number of frames. The digital trigger pulse is sent using the custom Labview program mentioned previously. To synchronize the pressure data with the corresponding Schlieren images, the custom Labview program simultaneously turns on data recording from the pressure transducers when the digital trigger is sent to the camera. At 10,000 samples /second, the time delay between the starting of the analog recording to the start of the image capturing is less than 0.1 ms with the NI PCI 6251 data acquisition board [25]. Figure 3.10 is a
photograph of the fully instrumented oscillator prototype in the Schlieren system. More photographs of the experimental facility are included on the accessory CD kept with the hard copy of the thesis in the Mechanical, Automotive and Materials Engineering Department.

![Figure 3.10 Photograph of the Experiment Facility](image)

3.4 Experimental procedure

Three independent variables are investigated in the experiments. The first one is the supply pressure which is controlled using the shop air supply system. The second one is the control channel resistance, which is set using control channel resistance plugs of different diameter. The third one is the exhaust chamber pressure, which is adjusted by opening or closing the valves on the exhaust chamber outlet to the atmosphere. Table 1 includes the values of the variables used for all trials in this study. The range of values is limited by capabilities of the air supply used and the oscillator variable ranges over which stable oscillations occur. All the pressure values are expressed as gage pressures in this thesis.
Table 1 Experimental Variables

<table>
<thead>
<tr>
<th>Study Case #</th>
<th>Supply Pressure (MPa)</th>
<th>Control Channel Plug Diameter (mm)</th>
<th>Average Exhaust Chamber Pressure (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0 to 0.46</td>
<td>No resistance</td>
<td>Open to atmosphere</td>
</tr>
<tr>
<td>2</td>
<td>0.23 to 0.43</td>
<td>No resistance 9.12, 7.54 and 6.78</td>
<td>Open to atmosphere</td>
</tr>
<tr>
<td>3</td>
<td>0.43</td>
<td>No resistance and 9.12</td>
<td>5 to 20</td>
</tr>
</tbody>
</table>

In each trial of the experiment, the light source is firstly turned on and the Schlieren image viewed using the camera. The position and orientation of the knife edge and lenses, as well as the camera settings, are checked and modified in a minor way to optimize the quality of the image. The camera capture is set to START mode [27] using Photron Viewer.

Next, the DC power for the data acquisition system is turned on and the custom LabVIEW program settings mentioned previously are made. The real-time monitors for each of the pressure transducers are checked to that they are giving zero values.

The oscillation is initialized by opening the shop air valve. The discharge of the valve is manually modified until the supply pressure shown in the corresponding monitor reaches the required value. A selected pair of control plugs is installed at the control ports, and the discharge of the valves on the exhaust chamber are manually modified to achieve the target average exhaust chamber pressure value, which is determined using a moving average of 2000 previous sample points. The LabVIEW 2014 program then sends the
trigger signal to the camera and simultaneously starts the data recording. The resulting data and captured images are saved into a specific folder on the computer.

3.5 Raw data treatment

Oscillations appeared under certain combinations of values of the three independent variables. In some cases, the oscillations were erratic and not stable which caused large cycle-to-cycle variations. The objective of the current thesis is to investigate the stable operating characteristics of the devices and not these instabilities and hence it is necessary to establish a criterion to determine if the oscillation variation is acceptable. To determine data acceptability, the raw data from different trials are studied and compared. Figure 3.11 shows representative samples of control channel pressure data for the unstable and stable conditions. The two lines in each figure show the data from the upper and lower control channel pressure transducer. It is obvious that the data on the left gives a much larger cycle to cycle variation compared to one on the right.

```
Supply Pressure = 0.4 MPa,
Average Back Pressure= 11.3 kPa
Without using the resistance plug
```

```
Supply Pressure = 0.4 MPa,
Average Back Pressure= 4.8 kPa
Without using the resistance plug
```

Figure 3.11 Samples of Raw Data for Unstable (left) and Stable (right) Oscillations

The raw data is processed by first determining the mean value for all cycles, \( P_{\text{mean}} \), as well as the maximum and minimum pressures in the \( i^{\text{th}} \) cycle, \( P_{\text{max}} \) and \( P_{\text{min}} \), .
respectively, from the N cycles of the pressure data for the upper and lower control channels. For every trial, a metric, referred to as the overall equivalent ratio, is calculated using these values in Equation 3.

\[
\text{er}_{\text{overall}} = \frac{\sum N \left\{ \left( \frac{\text{abs}((P_{\text{mean}} - P_{\text{min}}) - (P_{\text{max}} - P_{\text{mean}}))}{P_{\text{max}} - P_{\text{min}}} \right) \right\}}{N}
\]  

(3)

This criterion is a measure of the cycle to cycle similarity in the shape of the pressure data as verified by inspection of numerous cycles of different shape. A zero value of this criterion indicates that the pressure profile shapes of all the cycles are identical. A larger value of this criterion means greater cycle to cycle variation in the shape of the pressure profile. For some cycles however, the shape of the pressure data is the same in scale but the cycle period is different thus, a second criterion is needed. To account for cycle period differences the standard deviation of the cycle periods is used for this purpose.

Based on many observations of cycle-to-cycle variation in the pressure data, the largest tolerable overall equivalent ratio is taken as 0.34 and the largest tolerable cycle period standard deviation is 0.0064 second. In Figure 3.11, the left and right data sets correspond to overall equivalent ratios at 0.64 and 0.25 respectively, and to cycle period standard deviation at 0.032 and 0.0024 seconds respectively. Only oscillations simultaneously satisfying these criteria are referred to as stable oscillations and used in the remainder of the thesis.
Chapter 4 Analysis of Results

This chapter presents the experimental results as well as a comparison with CFD results obtained using the in-house model previously mentioned. The numerical model is also used to aid in understanding the experimental results. First, a general understanding of the events including the pressure traces that occur during one example oscillation cycle is obtained. The research is then divided into three studies, which concern three important independent operating parameters: the supply pressure, the control channel resistance, and the exhaust chamber pressure. In each study, Schlieren images and dimensional pressure profiles are shown and the effects on the overall oscillation performance expressed using non-dimensional parameters.

4.1 Flow field patterns and corresponding pressures throughout one sample cycle

One-half of the cycle of the oscillation can be separated into two different time periods. In the first, the flow is consistently attached to one output channel. In the second, the flow is in the process of switching from one output channel to the other.

The start and end of the first period are determined using the Schlieren images. If no apparent transverse motion is observed, the flow is considered to be attached; otherwise, the flow is switching.

4.1.1 Stable attachment of flow to one output channel

In the attached period, the flow pattern includes a stable shock diamond which only experiences minor changes. The feedback tank on the attached side is undergoing pressurization while that on the other side it is being de-pressurized. To understand the
characteristics of the stable flow attachment, the details of the shock diamond flow pattern, the sequences of the recorded pressure changes in the feedback tank and control channel as well as the flow reattachment location are investigated.

4.1.1.1 Shock diamond analysis

The sample Schlieren image shown in Figure 4.1(A) is obtained using a 0.43 MPa supply pressure, atmospheric exhaust chamber pressure, and 6.78 mm diameter control channel plugs. The supply pressure of 0.43 MPa is the maximum gage value of the supply pressure capable of generating stable oscillation in the case that the exhaust chamber pressure is atmospheric and no control resistance plug is used. By holding a constant supply pressure and the exhaust chamber pressure, the 6.78 mm diameter control channel plugs give the largest control channel resistance for stable oscillation. Figure 4.1(A) shows one instance of the main flow jet attaching to the upper output channel in the over-expanded condition. The shock diamond appears after the flow separation.

From right to left in this diagram, the main flow jet separates from the divergent walls extended from the nozzle. It is seen that the upper separation point is located further downstream compared to the lower one. This is because of the lower pressure on the upper side due to restricted jet entrainment compared to the lower side resulting in a more favorable pressure gradient in the flow direction on that side. Following the separation points, the oblique shock waves are seen to cross. The brightness of region 1 is greater than the region before the shocks, due to the greater pressure gradient. Next to the shock waves, the expansion waves cross and the brightness of region 2 is reduced due to its smaller pressure gradient. The following shocks are not as strong as the first one and hence are not as clearly seen; however, the changes of brightness in the following region
suggests the same pressure pattern as the theoretical schematic diagram shown in Figure 4.1 (B).

To avoid any confusion, it must be indicated that the non-uniform brightness in the control channel is not caused by a density gradient. The dark area in the control channel is due to an undesired light refraction from the aluminum surfaces, hence does not change with flow conditions and should be ignored.

Figure 4.1 Experimental Flow Pattern During the Stable Attachment Period

(A) Experimental Schlieren;
(B) Schematic Image of the Resulting Shock Diamond
The plots shown in Figure 4.2 are the results for the case when the flow is stably attached to the upper output channel generated using the numerical model given in Appendix A with a 0.4 MPa supply pressure, atmospheric exhaust chamber pressure, and a value of 10 for the control channel resistance coefficient. The resistance coefficient is represents the overall minor loss coefficient through the control channel, thus the pressure drop is determined by Equation 4 where c represents the value of control channel resistance coefficient:

$$\Delta P = 0.5 * c * \rho * v$$

(4)

If only the resistances of channel entrance and channel geometry are considered, the control channel resistance coefficient has a value of 0.75 (0.25 for flow entrance and 0.5 for 90-degree elbow). The supply pressure value of 0.4 MPa is the maximum value to sustain stable oscillation with atmospheric exhaust chamber pressure and the value of 0.75 for the control channel resistance coefficient. By holding the supply and exhaust chamber pressures constant and increasing the control channel resistance coefficient, the value of 10 is found as the upper limit for stable oscillation. The numerical result gives a very similar flow pattern, although the size of the shock diamond is different.

Figure 4.2 Numerical Flow Pattern During the Stable Attachment Period Schlieren (left); Velocity Magnitude Contour (right)
4.1.1.2 Changes of the feedback tank and the control channel pressures during the stable attachment period

Figure 4.3 and Figure 4.4 give the dimensionless experimental and numerical feedback tank and the control channel pressures during the half cycle when the flow is attached to the upper side. For both the experimental and numerical data, the boundary conditions are the same as those specified in the previous section. The non-dimensional time is a ratio of the local time to the cycle period. The non-dimensional feedback tank pressure is a ratio of the local gage pressure to the maximum gage pressure in the cycle. The non-dimensional control channel pressure is a ratio of the difference between the local control channel pressure and the cycle minimum control channel pressure to the cycle maximum to minimum pressure difference.

Figure 4.3 Experimental Feedback Tank and Control Channel Pressure Data during Stable Attachment
Because of the differences in the operating conditions, the actual values of the frequency and pressure amplitude are not expected to be the same. However, for this thesis it is more important to study the features and trend of the oscillation compared to the actual data values. The maximum gage pressures used to normalize the dimensionless feedback tank pressures were 59.4 kPa and 50.5 kPa in the experimental and numerical cases respectively. The maximum control channel pressure difference used to normalize the control channel pressures were 22.1 kPa and 30.1 kPa in the experimental and numerical cases respectively while the cycle period used to normalize the time was 0.068 secs and 0.06 secs in the experimental and numerical cases respectively. In Figure 4.3 and Figure 4.4 some features are seen to exist in both the experimental and numerical results, even though the actual values of these time locations are different. Five letters are labeled at the time locations where those special features occur in the pressure curves.
Focusing on the control channel pressure curves, “A” is the starting time of the observed “Attached” time period. From time “A” to time “B”, the lower control channel pressure increases but the upper control channel pressure decreases. At time “B”, the lower control channel pressure continues increasing, and the upper control channel pressure also begins to increase. Both the upper and lower pressures increase until time “C” and then they decrease. The decrement of the upper and lower pressures continues to time “D”, where the lower control channel pressure reaches atmospheric. After time “D”, the lower control channel pressure decreases to a negative gage value and the upper control channel pressure begins to fluctuate and slightly increase. The stable attached time period ends at time “E”.

For feedback tank pressure curves, at time “A” the pressure in the upper feedback tank has begun to build up, and the pressure in the lower feedback tank is reducing. From time “A” to time “D”, pressurization of the upper feedback tank and dissipation in the lower tank continues. At time “D”, the lower tank pressure reaches the minimum, which means the lower feedback tank is completely discharged. At point “E”, the upper feedback tank pressure reaches the maximum.

The experimental pressure data and Schlieren images do not give enough information to completely understand these pressure traces. To better understand these pressure curves, image sequences from time “A” to time “D” are studied using the 2D numerical flow contours. Figure 4.5 shows the streamlines and static pressure contours in the oscillator from time “A” to time “B”. In Figure 4.4, the decrease of pressure in the upper control channel is seen from time “A” to time “B”, which is due to the increasingly restricted entrainment of air from the main flow jet after the switch. In the same sequence
at the lower side, the flow direction through the control channel is not changed. The increased lower control channel pressure seen in Figure 4.4, which corresponds to a decreased flow velocity, indicates that the entrainment from the main flow jet is smaller after the flow switch. At the same time, the pressure rises along the lower output channel from point “A” to “B” due to the discharge from the lower feedback tank which is at a high pressure. This discharge also increases the lower control channel pressure.

![Figure 4.5 Numerical Pressure Contour and Streamlines at A and B](image)

After time “B”, the high-pressure flow discharge from the lower feedback tank continues and increases the local static pressure on the lower side of the main flow. The pressure difference between the lower side of the main jet and the upper control channel is hence reduced. Since the flow direction is not changed by this decreased pressure difference, the main flow entrainment to the upper control channel is smaller, which also
explains the upper control channel pressure increase seen in Figure 4.4 after point “B”.
The pressure rise at the lower side of the main flow jet ultimately stops the main flow
entrainment through the lower control channel. The direction of lower control channel
flow is then reversed and dominated by the high-pressure flow discharging from the
lower feedback tank. The lower control channel pressure continues to increase to a value
above the atmospheric pressure value until time “C”.

At time “C”, the lower feedback tank pressure has dropped to a certain level, and
the static pressure on the lower side of the main flow jet begins to decrease. Because the
direction of the upper control flow is not changed in Figure 4.6, the main flow
entrainment through the upper control channel is increased which makes the upper
control channel pressure drop again, as shown in Figure 4.4. The drop of the feedback
tank pressure also decreases the lower control channel pressure after time “C”. The
process continues until time “D”, where in Figure 4.4 the feedback tank pressure is close
to the minimum, and the lower control channel pressure is approximately the atmospheric
value.
Both the experimental and numerical results indicate that the shock diamond is shorten from the time “C” to time “D”, as the upper feedback pressure increases, which is shown in Figure 4.7.
As shown in Figure 4.4, the slope of the numerical upper feedback tank pressure curve reduces from time “C” to time “D”. Not as evident as the numerical result, in the experimental data, the slope of the upper feedback tank pressure slightly decreases around the point where the attached and unattached feedback tank pressure curves cross, as shown in Figure 4.3.

Figure 4.8 Numerical Flow Patterns During the Slope Change of the Upper Feedback Tank Pressure

Figure 4.8 shows the numerical flow patterns throughout this slope reduction. The slope reduction can be attributed to the increase of the pressure difference between the feedback tank pressure and the exhaust chamber pressure on the upper side. When this difference is small, the upper feedback tank is charged by part of the main jet flow. When the tank pressure is increased, a larger pressure difference gives greater resistance to the main flow entering the feedback tank. It is shown in Figure 4.8 that the number of streamlines entering the feedback tank is reduced with the time during this period. Once the feedback tank pressure reaches a certain value, most of the main flow jet is seen to go out through the exhaust port, and only a small portion enters the feedback tank. This can be shown to be the flow entrained from the upper control channel by following the streamlines back to the control channels.
After time “D”, the flow direction in the lower control channel changes again, as shown in Figure 4.9. The flow moves inward from the lower control channel and enters the lower feedback tank. This leads to the increase of the lower feedback tank pressure shown in Figure 4.4. At time “E”, the upper feedback tank pressure reaches the maximum and the flow direction at the upper feedback tank entrance reverses, indicating the beginning of the feedback tank discharge. A large vortex bubble is seen at the entrance of the lower output channel. After time “E”, the flow begins to switch.

![Figure 4.9 Numerical Pressure Contour and Streamlines at E](image)

5.1.1.3 Reattachment locations

As mentioned in the previous sections, the reattachment locations can be studied to determine the stability of the flow attachment in a bi-stable fluidic oscillator [7]. Once the reattachment happens, a vortex bubble will appear surrounded by the main flow jet. The Schlieren system used in the current experiment however, is not suitable for visualizing the vortex formation. In order to further investigate the reattachment during the stable attachment period, the numerical images of streamlines shown in Figure 4.10 are studied. The red circles with “a” and “b” are the locations of reattachment at different
times where the separated flow jet re-attaches to the outside wall in the upper output channel.

![Diagram of flow attachment](image)

**Figure 4.10 Numerical Results with Location Point of Reattachment**

When the flow switch is about to finish, the flow is reattaching at location “a”. A vortex appears between the control port and location “a”. After a short time, the pressure on the unattached side of the main flow jet rises and pushes the main flow jet further toward the upper wall. The curvature of the flow ahead to the reattachment point is reduced, and the size of the vortex decreases. Ultimately, the pressure rise at the unattached side pushes the reattachment to location “b” and reduces the vortex to a negligible size. The flow curvature and the reattachment are not obvious after this time.

4.1.2 Analysis of flow switching

The flow switching process occurs in a very short time period compared to the flow attachment. In the case considered, the total time for one transverse switch is approximately 3.1 ms, which is only about 9 percent of the time for a half cycle (34 ms). The jet swings from one output channel to the other and causes the pressure changes in the feedback tanks and the control channels.
4.1.2.1 Transverse motion of the flow switch

Figure 4.11 shows the transverse jet motion at three times during the flow switch from the upper to the lower side. A detached shock can be seen just ahead of the splitter when it is impacted by the main jet. This shock will influence the pressure drop along the output channel and may affect the switching.

Figure 4.11 Experimental Schlieren Images of the Flow Switch

Figure 4.12 includes the numerical result of static pressure contours and streamlines, which show that the vortex bubble enlarges on the previously attached side (upper) during the flow switch. In the middle of the switch process, the sizes of the vortex are the same on the upper and lower sides. When the switch is finished, the size of the vortex bubble reverses from the previous attachment condition.

Figure 4.12 Numerical Images of the Flow During Switching
4.1.2.2 Change of the control channel pressure and the feedback tank pressure during the switch

Figure 4.13 shows the experimental pressure data during the switch. The upper feedback tank pressure decreases due to discharge of fluid from the tank. The lower feedback tank pressure increases with a larger slope compared to the late portion of previous stable attachment period, due to the tank inflow contributed by the main jet. The upper control channel pressure increases while the lower control channel pressure decreases. These pressure traces continue to the beginning of the stable attachment time period.

![Figure 4.13 Experimental Pressure Data During the Flow Switch](image)

By studying the Schlieren images obtained at various times throughout a sample cycle with the corresponding pressure data for both the experiments and corresponding numerical result, a general understanding of the oscillation flow patterns and the resulting pressure traces is obtained. It is seen from the data however, that for different supply pressures, control channel resistances, and average exhaust chamber pressures, the
oscillation frequency and amplitude are different. It is necessary to further investigate the effects of changing those operating conditions. Each is considered separately below.

4.2 The effect of supply pressure

The experimental trials in this study are conducted with multiple supply pressures. The exhausts are open to the atmosphere without the chamber and no plug is used at the control ports. Comparable numerical trials are conducted for the various supply pressures, with exhaust chamber pressure equal to the atmospheric value and a value of 0.75 for the control channel resistance coefficient. The value of 0.75 for the resistance coefficient is considering the minor losses only from the channel entrance and the channel shape. The pressure values from the numerical results are different in magnitude than the experimental values due to the simplifying assumptions used in the model. The trends in shape of the curves are the same which is sufficient for the purpose of analysis.

4.2.1 Effect of supply pressure on the Schlieren images of the jet flow field

The Schlieren images shown in Figure 4.14 correspond to three different supply pressures at the instance when the unattached (lower) control channel pressure is at its

![Schlieren images](image)

Figure 4.14 Effect of Supply Pressure on the Experimental Schlieren Images of the Jet at Maximum Control Channel Pressure

The Schlieren images shown in Figure 4.14 correspond to three different supply pressures at the instance when the unattached (lower) control channel pressure is at its
maximum value. When the supply pressure is increased, the separation point of the main flow jet is pushed downstream, and the size of the exit shock wave is increased. This agrees with the theory discussed in Chapter 1 [6].

Since the exhaust chamber pressure is a constant, when the supply pressure increases, the momentum of the main flow jet increases. Therefore, the ratio of the transverse force on the jet caused by the control channel vacuum to the main flow thrust force is reduced and the deflection angle of the flow jet decreases. With the decreased deflection angle, the interference between the splitter wedge and main flow jet becomes more significant during the stable attached period.

Stable oscillations are seen to only occur over a certain range of supply pressures. When the supply pressure is lower than the range, the flow consistently attaches to one side (output channel) and there is no switch. If the supply pressure is above the range, the flow deflection becomes negligible, and the jet is equally separated by the splitter. In this study, the supply pressure range for stable oscillation is from 0.23 MPa to 0.43MPa. A larger supply pressure also leads to a greater oscillation frequency, increasing from about 16 Hz to 32 Hz.

\[ P_s = 0.15\text{MPa} \quad f=16\text{Hz} \]

\[ P_s = 0.23\text{MPa} \quad f=23.2\text{Hz} \]

\[ P_s = 0.40\text{MPa} \quad f=32.4\text{Hz} \]

Figure 4.15 Effect of Supply Pressure on the Numerical Schlieren Images of the Jet at Maximum Control Channel Pressure
The numerical model gives similar trends for the flow pattern as found in the experiments, as shown in Figure 4.15. The range of stable oscillation frequency in the numerical case is approximately the same as the experimental data. The numerical supply pressure range values found in this case are however, different (from 0.17 to 0.40 MPa) to the experimental results.

4.2.2 Effect of supply pressure on feedback tank and control channel pressures

An average cycle is determined for each experimental trial by using the Woody Average technique, which is explained in Appendix 2. The number of the cycles used for the average is 10 in each case.

Figure 4.16 shows the result of Woody averaging of the feedback pressure data for three different supply pressures. Because the same pressure change repeats on each side of the oscillator, each curve just shows the average of the half cycle transient pressure data of the two sides after applying a 180-degree phase shift to one side to synchronize the peak value. The trend of increasing oscillation frequency with supply pressure agrees with the observation from the Schlieren images. When the supply pressure is 0.23 MPa, the oscillation frequency is 16.06 ± 0.21 Hz, and the pressure amplitude is 31.62 ± 0.07 kPa. As the supply pressure increases to 0.33 MPa, the oscillation frequency and the pressure amplitude becomes 26.62 ± 0.14 Hz and 35.11 ± 0.25 kPa respectively. When the supply pressure reaches the upper limit, which is 0.43 MPa in this case, the oscillation frequency and the pressure amplitude becomes 32.02 ± 0.33 Hz and 25.97 ± 0.29 kPa respectively.
Figure 4.16 Effect of Supply Pressure on Feedback Tank Pressure Variation over a Cycle (Experimental)

The change of the feedback tank pressure curve shape can be explained using the simplified theory of the tank filling process, which is introduced in Appendix 1. The increase of oscillation frequency is consistent with a decrease in the tank filling time since the oscillator switch from one output to the other is related to the feedback tank pressure. Tank filling time is reduced by accelerated feedback tank pressurization, caused by the increased mass flow rate into the tank due to a decrease in the main jet deflection and an increase in the choked mass flow rate through the nozzle throat at higher supply pressures. The increased oscillation frequency with a higher supply pressure produces an increase in the maximum feedback tank pressures, but reduces the feedback tank pressure amplitude because the discharge period is decreased.

Change in the supply pressure not only changes the frequency and amplitude of the feedback tank pressure, but also make its profile shape different. With the low supply
pressure, the gradual pressure increase before the rapid rise, which is seen in time period D to E in the previous section, is more obvious compared to the high supply pressure data. This indicates that the time delay between the complete feedback tank pressure discharge on the unattached side and the end of the feedback tank filling on the attached side is reduced with the increase of the supply pressure. This result can be understood because the oscillation frequency, which is inversely proportional to the tank filling time, increases with the supply pressure.

Even though the actual pressure values are different, the numerical feedback tank pressure results shown in Figure 4.17 give the same trend that the increasing supply pressure decreases the oscillation frequency. Different from the experimental results, the peak value of the feedback tank pressure does not continually increase with the supply pressure. This may be due to the homogeneous pressure distribution assumption within the feedback tank used in the simulation and the different orientation of the feedback tank installation on the oscillator in the experiments (perpendicular to the output channels) than in the numerical model (parallel to the output channel).
Figure 4.18 shows the average cycles of the control channel pressure data at three different supply pressures. As in the case of the feedback tank pressures the results for only one control channel in each trial is shown since the shape of the curve for the other channel is the same with the exception that it is 180 degrees out of phase with the one shown. Because of this, the first half of the curve represents the time that the main jet is attached to the channel being measured while the second half of the curve represents the time that the jet is unattached to that channel. As expected, the frequency of the control channel pressure signal is identical to the feedback tank pressure. For the 0.23, 0.33 and 0.43 MPa supply pressures, the maximum vacuum control channel pressures are 5.80 ± 0.22, 6.77 ± 0.24 and 8.45 ± 0.36 kPa respectively, and the average control channel pressures are -3.09 ± 0.02, -3.68 ± 0.03 and -5.11 ± 0.04 kPa respectively.
Figure 4.18 Effect of Supply Pressure on Control Channel Pressure Variation over a Cycle (Experimental)

The rate of control channel pressure change is related to the intensity of the entrainment by the main flow jet, which depends mainly on the main jet velocity. The maximum vacuum control channel pressure is determined by the control channel pressure change rate and the time length of the stable attachment. When the supply pressure is increased, the attachment time decreases due to the increase in frequency, but the flow velocity increases. Similar to the feedback tank pressure, there is a supply pressure value that yields an optimized combination of the frequency and flow velocity. However, from the data in Figure 5.18, it is seen that the maximum vacuum control channel pressure just slightly increases, as the supply pressure changes from the 0.43 to 0.33 MPa (32 to 23 Hz), and significantly increases when supply pressure changes from 0.33 to 0.23 MPa (23 to 16 Hz). This seems to indicate that the change in the maximum vacuum control channel
pressure is more sensitive to changes in the flow velocity than to changes in the oscillation frequency.

The feature of control channel pressure in time period C to D described in the previous section, where the attached side control channel pressure gradually decreases, can be seen with the 0.23 and 0.33 MPa supply pressures, but does not appear with the 0.43MPa supply pressure. With 0.43MPa supply pressure, the cycle period is shorter, and there is not enough time for the attached side control channel pressure to slowly reduce before the flow switch.

The numerical control channel pressure results give the similar pressure traces and shown in Figure 4.19. The increasing supply pressure increases the maximum vacuum control channel pressure.

![Figure 4.19 Effect of Supply Pressure on Control Channel Pressure Variation over a Cycle (Numerical)](image-url)
4.2.3 Dimensionless study of the effect of the supply pressure on the oscillation over the useful range of oscillation

As mentioned in Chapter 2, the oscillation amplitude and frequency are the most important parameters of the oscillator performance for the application of interest in this thesis. Three non-dimensional parameters are used in the study.

The first parameter is the dimensionless frequency, which relates the feedback tank volume, the supply mass flowrate, the supply density and the oscillation frequency as given in Equation 5.

\[ F = \frac{f \cdot V_{fb} \cdot \rho}{m} \]  

(5)

Oscillations only occur over a certain supply pressure range between a lower limit, \( P_{sll} \) and an upper limit, \( P_{sul} \). These values are used to define a dimensionless pressure which represents the fraction of the pressure range being considered. This is referred to as the supply pressure fraction, SPF, as given by Equation 6. This makes the range of values from 0 to 1.

\[ SPF = \frac{(P_s - P_{sll})}{(P_{sul} - P_{sll})} \]  

(6)

The feedback tank pressure amplitude is made dimensionless using the maximum amplitude within the range as given in Equation 7.

\[ Pf_{b \, amp} = \frac{Pf_{b \, amp}}{Pf_{b \, amp \, max}} \]  

(7)

Figure 4.20 shows the dimensionless frequency versus the supply pressure change ratio. Both the experimental and numerical results show that the frequency increases with the supply pressure. The uncertainties are shown as error bars, but cannot be seen since the values are very small.
Figure 4.20 Dimensionless Frequency versus Fraction of Useful Supply Pressure Range

Figure 4.21 is a plot of dimensionless feedback tank pressure amplitude versus supply pressure fraction with the uncertainty error bars. Both the experimental and numerical results give upward convex shapes. An optimized supply pressure value is approximately 0.5 for generating the maximum feedback pressure amplitude. This figure shows that when the supply pressure approaches the limits, the feedback pressure amplitude decreases.
Figure 4.21 Dimensionless Feedback Tank Pressure Amplitude versus Fraction of Useful Supply Pressure Range
4.3 The effect of control channel resistance

For this experimental study, the supply pressure is held constant at different values in the range of 0.23 to 0.43MPa. The oscillator exhaust port is open to the atmosphere with no chamber attached. Three pairs of the resistance plugs are applied to the control ports for each of the constant supply pressures. The corresponding numerical trials are conducted with supply pressures ranging from 0.33 to 0.46 MPa with atmosphere exhaust chamber pressure. In the numerical work, the control resistance is changed by modifying the resistance coefficients. As mentioned in the previous study, comparison between the numerical model and experimental results focuses on the trend rather than comparisons at specific values of supply pressure.

4.3.1 Analysis of Schlieren images with different control resistances

By holding the supply pressure constantly at 0.4MPa, the Schlieren images are captured with different resistance plugs at the instant the unattached (lower) control pressure reaches the maximum and are shown in Figure 4.22.

![Schlieren Images](image)

Figure 4.22 Effect of Control Channel Resistance on the Experimental Schlieren Images of the Jet at Maximum Control Channel Pressure

Because the supply pressure and exhaust chamber pressure are constant, the size of the shock diamond does not change for different control resistances. Decreasing the
diameter of the hole on the plugs increases the flow resistance through the control channel. With a higher flow resistance, a larger vacuum pressure is generated in the attached control channel. As shown in the previous section, in the case of no plug, the flow interferes with the splitter during the flow attachment. Introducing plugs, enhances the attraction due to the transverse pressure difference which increases the flow deflection angle.

As the control resistance increases, the main flow attachment point is closer to the control channel. The oscillation frequency decreases due to the larger amount of angular travel of the main jet during the switch. By decreasing the plug diameter to 6.78mm, the oscillation frequency drops to approximately 15Hz.

The numerical results for the case of a 0.4MPa supply pressure and atmospheric exhaust chamber pressure with different control channel resistance coefficients give the same trend in the flow pattern changes as shown in Figure 4.23.

![Numerical Schlieren Images of the Jet at Maximum Control Channel Pressure](image)

Figure 4.23 Effect of Control Channel Resistance on the Numerical Schlieren Images of the Jet at Maximum Control Channel Pressure

4.3.2 Effect of control channel resistance on feedback tank and control channel pressures

Figure 4.24 shows the experimental feedback tank pressure variation over a cycle determined using the averaging technique previously described, for different control plugs and a 0.43 MPa supply pressure. The results for only one feedback tank is shown
since the other tank shape is the same with the exception that it 180 degrees out of phase with the one shown. For the 6.78, 7.54 and 9.12 mm diameter plugs and the case of no plug, the oscillation frequencies are 15.17 ± 0.14, 18.34 ± 0.12, 26.90 ± 0.16 and 32.02 ± 0.33 Hz respectively, and the feedback tank pressure amplitudes are 65.83 ± 0.25, 56.70 ± 0.29, 31.38 ± 0.24 and 25.97 ± 0.29 kPa respectively.

Figure 4.24 Effect of Control Channel Resistance on Feedback Tank Pressure Variation over a Cycle (Experimental)

Since the mass flow rate does not change significantly due to the constant supply pressure and constant exhaust chamber pressure, the feedback tank pressure amplitude change is dominated by the filling time, which is inversely proportional to the oscillation frequency. As the oscillation frequency reduces, the feedback tank pressure amplitude increases.
Figure 4.25 Effect of Control Channel Resistance on Feedback Tank Pressure Variation over a Cycle (Numerical)

The numerical results for feedback tank for a 0.4 MPa supply pressure and atmospheric exhaust chamber pressure with different values of control channel resistance coefficient show similar trends in Figure 4.25

Figure 4.26 Effect of Control Channel Resistance on Control Channel Pressure Variation over a Cycle (Experimental)
Figure 4.26 shows the experimental result of the control channel pressure with different control plugs and a 0.43 MPa supply pressure and atmospheric exhaust chamber pressure for one cycle. For the 6.78, 7.54, 9.12 mm diameter plugs and the no plug case, the maximum vacuum control channel pressure are 19.41 ± 0.59, 17.00 ± 0.54, 10.30 ± 0.53, 8.46 ± 0.36 kPa respectively, and the average control channel pressures are 8.31 ± 0.12, 7.49 ± 0.06, 5.83 ± 0.03 and 4.9 ± 0.04 kPa respectively.

When the control resistance increases, the oscillation frequency is seen to decrease. When the oscillation frequency decreases, the time increases for the gradual decrease in the control channel pressure before the rapid drop due to the switch during the unattached time (second half of the time period shown in Figure 4.26). The control pressure amplitude increases with the control resistance due to the larger vacuum generated in the control channel.

![Figure 4.27 Effect of Control Channel Resistance on Control Channel Pressure Variation over a Cycle (Numerical)]
The numerical results for channel pressures for a 0.4 MPa supply pressure and atmospheric exhaust chamber pressure with different values of control channel resistance coefficient show similar trends Figure 4.27 as the resistance coefficients are increased.

4.3.3 The effect of control channel resistance on the supply pressure range for stable oscillation

Until this point in the thesis the plug diameter and the resistance coefficient have been used to indicate the state of the control channel resistance. They however, don’t have the same units and therefore comparison could only be qualitative. A measure is needed that represents changes in the control channel resistance that is common to both the experiment and numerical work. Although not ideal, the maximum suction (vacuum) pressure in the control channel is such a value and will be used in the following graphs.

The range of supply pressure for stable oscillation changes with the control resistance for both the experimental and numerical results as shown in Figure 4.28 with uncertainty error bars for the experimental case. The supply pressure lower limit increases with the control resistance. Unfortunately, due to the restriction of the shop air pressure, the upper limit for high vacuum control pressure in the experimental case cannot be determined. However, the result with the low vacuum control pressures indicates the trend that the supply pressure upper limit increases, but the supply pressure range decreases with the increased control channel resistance. The numerical results are seen to agree with this trend.
Figure 4.28 Supply Pressure Ranges of Stable Oscillation for various Control Channel Pressure Values

Figure 4.29 shows the experimental feedback tank pressure amplitudes for different supply pressures with uncertainty error bars. The trends found with 0.43MPa supply pressure are still valid with other supply pressures. The feedback tank pressure amplitude is increased by increasing the control resistance with a constant supply pressure, which is also agreed by the numerical result shown in Figure 4.30.
Figure 4.29 Effect of Maximum Vacuum Control Channel Pressure on Feedback Tank Pressure Amplitude (Experimental)

Figure 4.30 Effect of Maximum Vacuum Control Channel Pressure on Feedback Tank Pressure Amplitude (Numerical)

Figure 4.31 shows the experimental feedback tank frequencies for different supply pressures with uncertainty error bars. The frequency is decreased by increasing the control channel resistance with a constant supply pressure. The numerical result, shown in Figure 4.32, gives the same trend. The oscillation frequency ranges are almost the same for both the experimental and numerical results with different control resistances considered in this study.
4.3.4 Dimensionless study of the effect of the control channel resistance on the oscillation over the useful range of oscillation

Three non-dimensional parameters are calculated in this analysis. The first one the dimensionless frequency, as defined in the section 4.2. The second one is the dimensionless vacuum control channel pressure, which is the ratio of the maximum vacuum control channel pressure to the supply pressure. The third one is the dimensionless feedback tank pressure amplitude, which is the ratio of the feedback tank amplitude to the supply pressure.
The dimensionless frequency is shown decreasing with the increase of the dimensionless vacuum control channel pressure for a constant supply pressure in Figure 4.33 with uncertainty error bars. For the same dimensionless frequency, the dimensionless vacuum control channel pressure is almost a constant for 0.33, 0.36 and 0.40 MPa supply pressures and significantly increased when the supply pressure rises to 0.43 MPa, which indicates that there is serious nonlinearity between the maximum vacuum control channel pressure and the supply pressure for generating a constant dimensionless frequency. For supply pressures from 0.43 to 0.46 MPa, the dimensionless vacuum control channel pressure increases in a minor way compared to the change from 0.40 to 0.43 MPa supply pressures for a constant dimensionless frequency. The numerical result as shown in Figure 4.34, gives a similar trend in that the dimensionless frequency
is decreased by the increase of dimensionless vacuum control channel pressure for a constant supply pressure. Likewise, a similar large rise in the value of the dimensionless vacuum control channel pressure for a constant dimensionless frequency is seen between supply pressures of 0.36 to 0.40 MPa.

As mentioned previously, the maximum vacuum pressure in the control channel is determined by the attachment period, the control channel resistance and the suction by the main flow jet, which can be seen by looking at the dynamic pressure around the inner control channel port near the expansion region. The dynamic pressure around the inner control channel port does not vary in a linear manner with the supply pressure and can be affected by different operating conditions hence, a nonlinearity between the maximum vacuum control channel pressure and the supply pressure with a constant dimensionless frequency can be expected. The significant rise of the dimensionless vacuum control channel pressure for a constant dimensionless frequency at a certain supply pressure may be caused by the change of the shock diamond shape and location, which would give different dynamic pressures in different sections of the expansion and compression waves.

The dimensionless feedback pressure amplitude increases with the dimensionless vacuum control channel pressure and is shown in Figure 4.35 with uncertainty error bars. The numerical result agrees with this trend as shown in Figure 4.36.
Figure 4.35 Dimensionless Feedback Tank Pressure Amplitude versus Dimensionless Vacuum Control Channel Pressure (Experimental)

Figure 4.36 Dimensionless Feedback Tank Pressure Amplitude versus Dimensionless Vacuum Control Channel Pressure (Numerical)
4.4 The effect of exhaust chamber pressure

In the experiment, selection of the exhaust chamber pressure is restricted by the limited range of supply pressure and the maximum discharge from the exhaust chamber. Consequently, the operating conditions can be only selected in a limited range for stable oscillation. The supply pressure is held at values of 0.43 MPa. The average exhaust chamber pressure in the chamber is therefore in the range 5 to 20 kPa. One set (9.12mm diameter) of control channel resistance plugs are also used in this section to obtain limited information regarding the effect that control channel resistance has on the results. The operating conditions for the numerical trials however, are set with a 0.4 MPa supply pressure over a wider exhaust chamber pressure range of 0 to 30 kPa with 0 to 14 for the control channel resistance coefficient. As mentioned previously, the trends of the oscillation performance rather than the actual values are of primary interest.

4.4.1 Analysis of Schlieren images for different average exhaust chamber pressures

Figure 4.37 shows the Schlieren images captured at the instance of maximum unattached control pressure with different values of average exhaust chamber pressure. As the exhaust chamber pressure increases, the shock diamond is squeezed, and the main flow deflection is increased, resulting in the reduction of the oscillation frequency. The flow separation point remains almost constant with changes in the exhaust chamber pressure.
Figure 4.37 Effect of Average Exhaust Chamber Pressure (Pc average) on Experimental Schlieren Images of the Oscillator Flow Field

For the 0.43 MPa experimental supply pressure without control resistance plug, the range of exhaust chamber pressure for stable oscillation is from 0 to 20 kPa. The exhaust chamber pressure above this range leads to the permanent flow attachment to one of the output channels.

Figure 4.38 shows the numerical Schlieren images for 0.4 MPa supply pressure with a value of 0.75 for the control channel resistance coefficient. The result gives the same trend of flow patterns as the experimental data.
4.4.2 Feedback tank and control channel pressure variation with average exhaust chamber pressure

Figure 4.39 shows the feedback tank pressure data for one average cycle for different average exhaust chamber pressures. When the average exhaust chamber pressure increases, the oscillation frequency decreases but the feedback tank pressure amplitude increases. For the average exhaust chamber pressure values of 5, 10 and 20 kPa, the feedback tank pressure amplitudes are 43.54 ± 0.19, 49.72 ± 0.19 and 55.55 ± 0.24 kPa respectively, and the frequencies are 24.63 ± 0.14, 19.69 ± 0.17 and 15.29 ± 0.18 Hz respectively.

![Figure 4.39 Effect of Average Exhaust Chamber Pressure on Feedback Tank Pressure Variation over One Cycle (Experimental)](image)

The numerical results of feedback tank pressure show the same trends in Figure 4.40. The oscillation frequency is decreased, and the feedback pressure difference is increased by increases in the exhaust chamber pressure.
The control channel pressure variation over one cycle is shown in Figure 4.41 for different average exhaust chamber pressures. For the average exhaust chamber pressure values of 5, 10 and 20 kPa, the maximum vacuum control channel pressures are $7.45 \pm 0.29$, $6.39 \pm 0.64$ and $4.88 \pm 0.48$ kPa respectively. The small spike portion of the attached control pressure, which is around time point C defined in section 4.1 and labeled by red circles in Figure 4.41, disappears when the average exhaust chamber pressure reaches 20 kPa.
Figure 4.41 Effect of Average Exhaust Chamber Pressure on Control Channel Pressure Variation over One Cycle (Experimental)

The numerical results of control channel pressure give the same trends in Figure 4.45. For the high values of average exhaust chamber pressure, the spike of the unattached control pressure after the switch is not obvious.

Figure 4.42 Effect of Average Exhaust Chamber Pressure on Control Channel Pressure Variation over One Cycle (Numerical)
Figure 4.43 shows both the feedback tank and exhaust chamber pressures over one cycle for an average exhaust chamber pressure of 20 kPa. It shows that the oscillation frequency in the exhaust chamber is twice that of the feedback tank pressure. This is to be expected as during one cycle, the exhaust chamber receives flow from two output channels while each tank only receives flow from one channel. The exhaust chamber pressure is at a minimum when the feedback tank pressure is close to the maximum value as the supply flow can either go into the exhaust chamber or a feedback tank. The maximum of the exhaust chamber pressure appears during the rise and drop of the feedback tank pressure.

![Figure 4.43 Comparison of Feedback Tank and Exhaust Chamber Pressure Variation Over One Cycle (Experimental)](image)

The numerical results shown in Figure 4.44 are in general agreement in shape but not magnitude.
Averaged exhaust chamber pressure variation over a cycle for different average exhaust chamber pressure values are shown in Figure 4.45. The exhaust chamber pressure oscillation amplitude and cycle period increase with the average exhaust chamber pressure. For average exhaust chamber pressures of 5, 10 and 20 kPa, the amplitudes are 2.46±0.03, 3.76±0.04 and 5.22±0.04 kPa respectively.
The numerical results of the exhaust chamber pressure also show that the amplitude increases with the feedback tank pressure amplitude in Figure 4.46.

![Figure 4.46 Effect of Average Exhaust Chamber Pressure on Exhaust Chamber Pressure Variation Over One Cycle (Numerical)](image)

4.4.3 The effect of average exhaust chamber pressure on oscillator performance

In Figure 4.47, the experimental oscillation frequency is seen to decrease with an increase of the average exhaust chamber pressure for a constant supply pressure and a constant control channel resistance. By holding a constant supply pressure, the oscillation frequency is decreased by an increase of the control channel resistance at different average exhaust chamber pressures. This indicates that the trend found with a zero-exhaust chamber pressure for different control channel resistances is also valid with a non-zero gage exhaust chamber pressure.
Figure 4.47 Effect of Average Exhaust Chamber Pressure on Feedback Tank Frequency (Experimental)

Figure 4.48 shows that for the constant supply pressure and exhaust chamber resistance, the experimental feedback tank amplitude is increased by the increase of the average exhaust chamber pressure. This can be explained as follows: when the average exhaust chamber pressure is increased, resistance to flow through the exhaust is greater, thus the mass flow rate into the feedback tanks increases during the filling process.

By holding a constant supply pressure, the feedback tank amplitude is increased with the increase of the control channel resistance
Figure 4.48 Effect of Average Exhaust Chamber Pressure on Feedback Tank Pressure Amplitude (Experimental)

Figure 4.49 gives the experimental results of the maximum vacuum control channel pressure for different average exhaust chamber pressures. The maximum vacuum control channel pressure is seen to decrease with an increase of the average exhaust chamber pressure. This is attribute to the overall increase of the upstream and downstream pressures.

Figure 4.49 Effect of Average Exhaust Chamber Pressure on Maximum Vacuum Control Channel Pressure (Experimental)
Figure 4.50 shows that for a constant supply pressure, both the increase of the average exhaust chamber pressure and the increase of the control channel resistance leads to an increase of the exhaust chamber pressure amplitude. The same trend is seen in the feedback tank pressure amplitude shown in Figure 4.48, indicating that the exhaust chamber pressure amplitude is positively correlated to the feedback tank pressure amplitude.

![Graph showing exhaust chamber pressure amplitude versus average exhaust chamber pressure](image)

**Figure 4.50 Exhaust Chamber Pressure Amplitude versus Average Exhaust Chamber Pressure (Experimental)**

4.4.4 Dimensionless study of the effect of the exhaust chamber pressure on the overall oscillation performance

The dimensionless frequency and the dimensionless feedback tank pressure amplitude are determined as mentioned previously. The dimensionless exhaust chamber pressure is defined as the ratio of the exhaust chamber pressure to the supply pressure. Figure 4.51 shows that the dimensionless frequency decreases with an increase of the
dimensionless exhaust chamber pressure for different control channel resistances, and with the decrease of the control resistance if the dimensionless exhaust chamber pressure is held constant, which is in agreement with the numerical results shown in Figure 4.52.

![Figure 4.51 Dimensionless Frequency versus Dimensionless Exhaust Chamber Pressure (Experimental)](image1)

![Figure 4.52 Dimensionless Frequency versus Dimensionless Exhaust Chamber Pressure (Numerical)](image2)

The dimensionless feedback tank pressure amplitude is increased with an increase of the dimensionless exhaust chamber pressure, as seen in both the experimental (Figure 4.53) and numerical (Figure 4.54) results.
Figure 4.53 Dimensionless Feedback Tank Pressure Amplitude versus Dimensionless Exhaust Chamber Pressure (Experimental)

Figure 4.54 Dimensionless Feedback Tank Pressure Amplitude versus Dimensionless Exhaust Chamber Pressure (Numerical)
Chapter 5 Conclusions and Recommendations

In conclusion, the objectives of this thesis have been met in the following ways:

1) An experimental prototype has been designed and constructed to further investigate Hiroki’s design and validate the corresponding 2D CFD model.

2) A supersonic fluidic oscillator test facility has been constructed which includes a combined Schlieren and high-speed photographic system for flow visualization, pressure transducers and computer controlled data acquisition arrangement to record pressures at the supply reservoir, control channels, feedback tanks and the exhaust chamber.

3) Experiments have been conducted to obtain the operating condition ranges necessary for stable oscillation. A procedure has been established to identify stable oscillating performance measure which allows a quantitative determination of the limits of stable oscillation.

4) A careful analysis of the experimental pressure data and Schlieren images, obtained with one set of operating conditions yielded a detailed description of the flow field and pressure variation over one complete cycle of operation. Results from the numerical model with similar operating conditions were used to help interpret the data. Highlights of this description include 1) identification of two distinct portions of each half of the switching cycle; a stable jet attachment portion and a switching portion (only 10% of the total time), 2) a quasi-steady over-expanded supersonic jet shock-diamond pattern during the attached period and 3) the formation an oblique shock pattern around the tip of the splitter during the switch.
5) The experimental and numerical results obtained by changing the supply pressure while holding the exhaust chamber pressure and control channel resistances constant indicate that:

i. Stable oscillations only occur there within a range of supply pressure,

ii. If the supply pressure is smaller than the range, the flow will permanently attach to one side and,

iii. If the flow is larger than the range, the flow will be equally separated into two outputs,

iv. There is a decrease in the size of the shock diamond and the flow deflection angle for an increase in the supply pressure,

v. There is an increase in the oscillation frequency with increases in the supply pressure,

vi. The maximum value of the feedback pressure amplitude near the middle of the supply pressure range

6) The experimental and numerical results obtained by holding the exhaust chamber pressure constant while increasing the control channel resistance shows an increase in both upper and lower limit of the supply pressure range for stable oscillation. The range of supply pressure however, is smaller with a larger control channel resistance.

7) The experimental and numerical results obtained by holding the supply and exhaust chamber pressures constant while increasing the control channel resistance

i. Does not change the size of the shock diamond significantly, but the flow deflection angle is increased because a larger vacuum is generated in the control channel.
ii. Decreases the oscillation frequency and

iii. Increases the feedback tank pressure amplitude.

8) Both the experimental and numerical results show that for a constant supply pressure and control channel resistance, changes in the average exhaust chamber pressure:

i. Indicate a range of average exhaust chamber pressure that produces a stable oscillation.

ii. The size of the shock diamonds and the deflection angle increases with an increase in average exhaust pressure,

iii. The frequency of oscillation is decreased with increases in average exhaust chamber amplitude, and

iv. The feedback tank pressure amplitude and the exhaust chamber amplitude are increased by increases in the average exhaust chamber pressure.

9) The numerical model results are in general agreement with the experimental results. Any differences may be attributed to some of the assumptions made in the lumped parameter model and the limitations of the exhaust chamber pressure conditions in the experiment.

The following recommendations are made for future research in this area.

1. The compressed air supply for the test facility be changed from the building “shop air” system to a bank of compressed air (or nitrogen) tanks. This will overcome the experimental limitations on the supply pressure value and allow validation of the numerical model over a wider range of operating conditions.

2. A new test oscillator should be designed which allows changing the volumes of the feedback tank and exhaust chamber, an effect that is not considered in the current
work. A different geometry of the oscillator should also be constructed to investigate how the geometry parameters such as outlet port orientation affect the oscillation frequency and amplitude.

3. One-dimensional and lumped parameter mathematical models of the supersonic oscillator which can quickly give a reliable prediction of the oscillator performance should be developed and their accuracy determined. This would shorten the extensive design times which are currently required using two-dimensional CFD techniques. These models may also allow device optimization for specific applications.
References


Appendix A. Numerical Model of the Bi-Stable Supersonic Fluidic Oscillator

This appendix gives the description of the CFD model used in this thesis, followed by the equation development and logic of the User Defined Function applied in the lumped parameter model to determine the transient pressure change in the feedback tanks and exhaust chamber. The ANSYS Fluent 17 code is used.

A.1 Description of the numerical model

The model is set in a 2D planar system shown in the Figure A.1. The solution domain of the model is based on the design of the flow region in Hiroki’s work after making certain reasonable assumptions which will become evident. The mesh of this domain consists of 28671 hexahedral elements and the transient solution is found using a double-precision density-based implicit AMG solver, with a spatial discretization and transient formulation that is of second order accuracy. The k-omega sst viscous model is utilized in the calculation. The wall roughness for the solid walls in the model is taken to be 0.046 mm.
The oscillator reservoir is assumed to be at a homogeneous pressure and hence is not included in the solution domain. The supply pressure inlet is set to the reservoir pressure value. The control channels are open to the atmosphere and hence the inlet ports are set as pressure inlets at atmospheric pressure. A porous jump boundary is located in each of the control channels to allow a setting of the channel resistance in the form of a constant minor loss coefficient. The porous jump is located 29.1 mm away from the channel inlet. The location of the jump has been shown not to influence the solution. The minor loss coefficient is estimated to be 0.7 which is consistent with the minor loss for a 90-degree elbow and tank entrance.

The feedback tanks are not included in the solution domain but are simulated using two external lumped-parameter models for the tank filling and discharging process. The models interact with the CFD through the use of User Defined Functions (UDF). The
UDF scans and records the total pressure in the region immediately before the tank inlet and compares it with the pressure in the tank. The flow in the tank is assumed to be stagnant and uniform throughout the volume. If the total upstream pressure is larger than the tank pressure, flow is allowed to fill the feedback tank; otherwise, the feedback tank is full or discharging. For either the charging or discharging process, once the downstream to upstream total pressure ratio is lower than the choking criteria which is 0.528 for an ideal gas, the flow is assumed choked at the tank inlet. On each feedback tank, a Boolean logic condition is applied to simulate the opening or closing of the tank.

The transient solution is run with a 0.1 millisecond time step and the mesh is auto-adapted based on the pressure gradient, every 15 time steps. At the end of each time step, the pressure change in the feedback tank is determined using the mass flow rate through the tank inlet at that instance as found from the CFD solution. Equation A1.1 is for calculating the transient pressure under isothermal assumption. The development of this equation is introduced in the next section. The polytropic index “n” in this equation is 1.4 for an isentropic process and 1 for an isothermal process. The parameter “T<sub>o</sub>” is the stagnation upstream temperature for the isentropic condition and is the tank temperature for the isothermal condition.

\[
\Delta P = n \cdot \frac{R \cdot T_o}{V_{cv}} \cdot \dot{m} \cdot \Delta t
\]  

\( A1.1 \)

In order to study the effect of the strength and oscillation of the downstream exhaust chamber pressure, a chamber is created as a lumped parameter model connected to the exhaust boundaries. The logic of this lumped parameter model is as the same as for the feedback tanks, except there it includes an orifice for changing the average exhaust pressure in the chamber. By using different combinations of supply pressures and orifice
discharge coefficients, different average exhaust chamber pressures can be achieved. Equation A1.2 is used for calculating the pressure change through the orifice discharge.

\[ \Delta P = \alpha \cdot A_{orifice} \cdot \sqrt{2 \cdot \frac{P_o}{V_{cv}}} \cdot \rho \cdot \frac{R + T_o}{V_{cv}} \cdot \Delta t \]  

A1.2

The details of the lumped parameter models for both the feedback tank and exhaust chamber are shown in the next section.

After giving a constant supply pressure, a constant average exhaust chamber pressure and constant control channel resistances, the simulation is initiated with one feedback tank closed to the environment and the other one open. This leads to a pressure difference between the two feedback tanks and creates non-symmetries in the flow field. If the appropriate supply to exhaust pressure ratio is given, the main flow jet will attach to one side due to Coanda effect and can be treated as a steady flow problem. After 0.01 second simulation time period, the valve on the feedback tank opening to the environment is closed, and the corresponding tank pressure starts to build up. This pressurization process continues until the tank pressure reaches a certain value, then the main flow jet is forced to the other channel. The same process repeats and the self-sustained oscillation begins.

From the simulation result, it is found that the oscillation occurs in a range of the supply pressure when the exhaust ports are at atmospheric pressure. When the supply pressure is smaller than 0.12 MPa, the main flow jet will always attach to one side and cannot be switched. When the supply pressure is larger than 0.4 MPa, the flow jet will symmetrically split to both output channels thus no oscillation occurs. Within the range of supply pressure, the constant boundary conditions make self-sustained oscillation occur at a constant frequency. The oscillation frequency increases with a decrease of the
feedback tank volume. The tendency of tank pressure change due to tank volume change is the same as described in Hiroki’s work. However, in Hiroki’s work, the oscillator is claimed to be operating in the 0.3 to 0.6 MPa supply pressure range. At the same time, when the supply pressure is the same, the oscillation frequency and its rate of change with tank volume are different between the simulation result and Hiroki’s data. The comparison is shown in Figure A.2.

![Figure A.2 Comparison of Oscillation Frequency with Different Feedback Tank Volumes between CFD Result and Hiroki’s Data](image_url)

The numerical model proves that this supersonic oscillator is capable of producing stable oscillation under the appropriate boundary conditions. The numerical results give the similar pressure trace and flow pattern as Hiroki’s work, but differences are observed between the numerical results and Hiroki’s data with the same boundary conditions. Even though these differences cannot be explained well due to the lack of the information from Hiroki’s experiments, the numerical model gives the confidence for building and conducting a new experiment to investigate this fluidic bi-stable oscillator design further.
A.2 Equation development to determine instant pressure change in a finite volume

This section gives the equation development based on a simple tank filling problem. The equations are later used to determine the pressure change in the feedback tank and exhaust chamber and are applied into the UDF for the lumped parameter model.

A simple tank filling/discharging problem is considered as shown in Figure A.3.

![Figure A.3 Schematic Diagram of Simple Tank Filling Problem](image)

In this system, the conservation laws must be satisfied, and a polytropic process is assumed. The mass and energy conservation laws and the polytropic process in this system are described in Equation A1.3 to A1.5:

**Mass conservation:**

\[
\dot{m}_{\text{inflow}} = \frac{d}{dt}(\rho \ast V)_{\text{tank}} = \left[\frac{V}{R} \ast \frac{d}{dt}\left(\frac{P}{T}\right)\right]_{\text{tank}}
\]

\[
= \left\{\frac{V}{R} \ast \left[P \ast \left(-T^{-2}\right) \ast \frac{d}{dt}(T) + \left(T^{-1}\right) \ast \frac{d}{dt}(P)\right]\right\}_{\text{tank}} \quad \text{A1.3}
\]

**Energy Conservation:**

\[
\frac{dE}{dt} = \frac{dQ}{dt} - \frac{dW}{dt} = \frac{d}{dt}\left(e1 \ast \rho_cv \ast dV\right)_{\text{tank}} - (e2 \ast \dot{m}_{\text{inflow}}) \quad \text{A1.4}
\]
Polytropic process:

\[
\begin{align*}
(T * P_{\frac{1-n}{n}})_{\text{polytropic}} &= C \\
\frac{d}{dt} (C) &= [T * \frac{d}{dt} (P_{\frac{1-n}{n}}) + P_{\frac{1-n}{n}} * \frac{d}{dt} (T)]_{\text{polytropic}} = 0 \\
\frac{d}{dt} (T)_{\text{polytropic}} &= \left[ -T * \left(\frac{1-n}{n}\right) * P_{\frac{1-2n}{n}} * \frac{d}{dt} (P) * P_{\frac{n-1}{n}} \right]_{\text{polytropic}} \\
&= \left[ \frac{n-1}{n} * \frac{T}{P} * \frac{d}{dt} (P) \right]_{\text{polytropic}} \quad \text{A1.5}
\end{align*}
\]

If the filling/discharge process is isothermal, the temperature in the system will be a constant and equal to the tank fluid temperature.

Based on Equation A1.5:

\[
\begin{align*}
(T * P_{\frac{1-n}{n}})_{\text{polytropic}} &= C = (T * P_{\frac{1-n}{n}})_{\text{tank}} \\
\frac{d}{dt} (T)_{\text{tank}} &= \left[ \frac{n-1}{n} * \frac{T}{P} * \frac{d}{dt} (P) \right]_{\text{tank}} \quad \text{A1.6}
\end{align*}
\]

Substituting Equation A1.6 into Equation A1.3 gives:

\[
\begin{align*}
\dot{m}_{\text{inflow}} &= \left\{ \frac{V}{R} * \left[ P * (-T^{-2}) * \frac{n-1}{n} * \frac{T}{P} * \frac{d}{dt} (P) + (T^{-1}) * \frac{d}{dt} (P) \right] \right\}_{\text{tank}} \\
\dot{m}_{\text{inflow}} &= \left\{ \frac{V}{R} * \left[ -T^{-1} * \frac{n-1}{n} * \frac{d}{dt} (P) + T^{-1} * \frac{d}{dt} (P) \right] \right\}_{\text{tank}} \\
\dot{m}_{\text{inflow}} &= \left\{ \frac{V}{RvT} * \left(\frac{1-n}{n} + 1\right) * \frac{d}{dt} (P) \right\}_{\text{tank}}
\end{align*}
\]

Ultimately:

\[
\begin{align*}
\dot{m}_{\text{inflow}} &= \left[ \frac{V}{RvT} * \frac{1}{n} * \frac{d}{dt} (P) \right]_{\text{tank}} \\
\frac{d}{dt} (P)_{\text{tank}} &= n * \left(\frac{RvT}{V}\right)_{\text{tank}} * \dot{m}_{\text{inflow}} \quad \text{A1.7}
\end{align*}
\]

So the instantaneous pressure change in a tank can be determined by Equation A1.7 for the isothermal system.
If the filling/discharging process is adiabatic, the temperature will not be constant in the system, thus Equation A1.6 cannot be obtained. In the adiabatic system, the heat loss is zero, and the following conditions are obtained in the energy conservation:

\[
\frac{dQ}{dt} = 0; \quad \frac{dw}{dt} = 0; \quad e1 = u; \quad e2 = h - g * Z
\]

Substitute these conditions into Equation A1.4, then:

\[
0 = \frac{d}{dt}(U * \rho_{\text{tank}} * V_{\text{tank}}) - (h - g * Z) * \dot{m}_{\text{inflow}}
\]

\[
0 = \frac{d}{dt}(cv * T_{\text{tank}} * (\frac{P}{R*T})_{\text{tank}} * V_{\text{tank}}) - (h - g * Z) * \dot{m}_{\text{inflow}} \quad \text{A1.8}
\]

Since the heat capacity and the tank volume are assumed constant:

\[
\frac{d}{dt}(cv) = 0; \quad \frac{d}{dt}(V_{\text{tank}}) = 0
\]

Equation A1.8 becomes:

\[
V_{\text{tank}} * \frac{cv}{R} * \frac{d}{dt}(P)_{\text{tank}} = (c_p * T_{\text{inflow}} - g * Z) * \dot{m}_{\text{inflow}}
\]

\[
\frac{d}{dt}(P)_{\text{tank}} = (c_p * T_{\text{inflow}} - g * Z) * \dot{m}_{\text{inflow}} * \frac{R}{cv * V_{\text{tank}}} \quad \text{A1.9}
\]

Consider Z is as the minor loss, then:

\[
Z = s * \frac{v^2}{2} * \frac{1}{g}
\]

Equation A1.9 becomes:

\[
\frac{d}{dt}(P)_{\text{tank}} = \frac{c_p}{cv} * \frac{R*T_{\text{inflow}}}{V_{\text{tank}}} - \frac{1}{2} * \frac{R}{cv} * \frac{cv^2}{v_{\text{tank}}^2} * \dot{m}_{\text{inflow}} \quad \text{A1.10}
\]

If the minor loss of the inflow/outflow is neglected, then:

\[
\frac{d}{dt}(P)_{\text{tank}} = \frac{c_p}{cv} * \frac{R*T_{\text{inflow}}}{V_{\text{tank}}} * \dot{m}_{\text{inflow}} \quad \text{A1.11}
\]

The instantaneous pressure change in a tank can be determined by Equation A1.11 for the adiabatic system.
A.3 Logic in the lumped parameter models to determine transient pressure change

The pressure in the feedback tanks can be considered as a tank filling/discharging problem with one open port as shown in Figure A.4. By using Equations A1.7 or A1.11 given in the previous section, the instant pressure change for one time step in the feedback tanks can be determined.

![Figure A.4 Schematic Diagram of Feedback Tank Filling/Discharging](image)

The logic shown in Figure A.5 uses the instant pressure change in one time step to calculate the updated pressure in the next time step, thus generating the transient feedback tank pressures.
The pressure in the exhaust chamber can be considered as a tank filling/discharging problem with three open ports as shown in Figure A.6.

Figure A.6 Schematic Diagram of Exhaust Chamber Filling/Discharging
For each of the ports, Equation A1.7 or A1.11 can be applied. The instant pressure change in the chamber is the combination of the results from the three ports. The logic shown in Figure A.7 uses the instantaneous pressure change in one time step to calculate the updated pressure in the next time step, thus generates the transient exhaust chamber pressure.

The logic introduced in this section is coded into one UDF file and used to give appropriate transient boundary conditions for the feedback pressures and the exhaust chamber pressure. The reason to integrate the logic into one file is the convenience of coding for a multi-thread parallel calculation. The actual code of the UDF is included in an accessory CD, which is kept with a copy of the thesis stored in the University of Windsor Mechanical, Automotive and Materials Engineering Department.
Figure A.7 Logic to Calculate Transient Exhaust Chamber Pressure
Appendix B. Uncertainty Analysis

This appendix explains how the uncertainties are calculated for the quantities used in this thesis. For each average cycle pressure result shown in this thesis, the uncertainty is made of two parts. One is the uncertainty from the measurement system which includes the error from the sensors and the acquisition board. The other one is from the cycle to cycle variation in the experiment.

B.1 Uncertainty analysis for the measurement system

The design stage error of the measurement system is calculated based on the manufacturer specification of the pressure sensors and the AC converter. The pressure transducer B, C, D, and E are all labeled at the accuracy of 0.2\% of full scale, which includes the error from resolution and other possible instrument errors. The pressure transducers A and F have an accuracy of 0.08\% of full scale. The resolution of the A/D converter is 16 bit, which means the bias uncertainty of each pressure transducer from it can be calculated as the full-scale pressure measurement range times 0.5/4096.

The overall design stage uncertainty of each pressure transducer is calculated by Equation A2.1:

$$U_d = \sqrt{(U_{\text{transducer}}^2 + U_{\text{AC converter}}^2)}$$  \hspace{1cm} \text{A2.1}

The design stage uncertainty of each pressure transducer is shown in Table 2:
Table 2 Design Stage Uncertainties of the Pressure Transducers

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>Measurement Full Scale (kPa)</th>
<th>U_{transducer} (kPa)</th>
<th>U_{AC convertor} (kPa)</th>
<th>U_d (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1000</td>
<td>0.8</td>
<td>0.244</td>
<td>0.8</td>
</tr>
<tr>
<td>B</td>
<td>+/- 100</td>
<td>0.4</td>
<td>0.024</td>
<td>0.4</td>
</tr>
<tr>
<td>C</td>
<td>+/- 100</td>
<td>0.4</td>
<td>0.024</td>
<td>0.4</td>
</tr>
<tr>
<td>D</td>
<td>+/- 35</td>
<td>0.14</td>
<td>0.0086</td>
<td>0.14</td>
</tr>
<tr>
<td>E</td>
<td>+/- 35</td>
<td>0.14</td>
<td>0.0086</td>
<td>0.14</td>
</tr>
<tr>
<td>F</td>
<td>+/- 100</td>
<td>0.16</td>
<td>0.024</td>
<td>0.16</td>
</tr>
</tbody>
</table>

The measurement precision error, which comes from the electrical noise and the power supply disturbance, is estimated by repeating the measurement under the no-flow (zero-gauge state) condition. For each trial, 30,000 data points are taken to obtain one measurement, and ten random repeated measurements are made. By taking zero as the true mean value of the data, the corrected sample standard deviation of each trial is calculated. The measurement precision error of each pressure transducer is the average of the ten, corrected sample standard deviations.

The measurement precision error of each pressure transducer is shown in Table 3:

Table 3 Precision Errors of the Pressure Transducers

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>Measurement Precision Error (kPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.2065</td>
</tr>
<tr>
<td>B</td>
<td>0.1937</td>
</tr>
<tr>
<td>C</td>
<td>0.2</td>
</tr>
<tr>
<td>D</td>
<td>0.1161</td>
</tr>
<tr>
<td>E</td>
<td>0.95</td>
</tr>
<tr>
<td>F</td>
<td>0.1549</td>
</tr>
</tbody>
</table>
The total uncertainty at the zero gauge state for the SFO can be estimated by Equation A2.2:

\[
U_{\text{static}} = \left[ U_d^2 + (t \times \text{Precision})^2 \right]^{1/2}
\]

A2.2

If 95\% probability is assumed, the total uncertainties calculated for the six pressure transducers are 2.45 kPa (transducer A), 0.569 kPa (transducer B), 0.552 kPa (transducer C), 0.416 kPa (transducer D), 0.267 kPa (transducer E) and 0.343 kPa (transducer F).

B.2 The average cycle and the variable uncertainty in a trial

One pressure data set in a stable oscillation trial can be divided into multiple cycles by using the local peak pressure value in each cycle. Using the Woody’s adaptive filter system which iteratively corrects the inter-trial latency, the average cycles of the pressure data at two sides of the oscillator is acquired for each trial [28]. The results for the average cycles on the two sides shifted so that they are in phase, synchronized by the peak value and then averaged.

Meanwhile, the variable precision errors in the trial are determined from the divided cycles. The variable uncertainties in the average cycle are calculated as the uncertainty of the mean values by Equation A2.3:

\[
U_x = \sqrt{\left( \sqrt{t_{95\%} \times S_x^2} \right)^2 + U_{\text{static}}^2} / N_{\text{cycles}}
\]

A2.3

The actual MATLAB codes for calculating the average cycle and the uncertainty are included in the accessory CD kept with the hard copy of the thesis in the Mechanical, Automotive and Materials Engineering Department. The codes read the data from Excel.
files and can work once all the code files are put into one folder. The main code is named “Uncertainty Analysis”. 
Appendix C. Labview Programing

This appendix shows the Labview program used in the experiment for this thesis work. Both the front panel, which is for the actual operation in the experiment, and the block diagram, which is the logic of the program, are introduced.

C.1 Front panel

The full view of the front panel is shown in Figure C.1. This front panel is used to control the measurement and monitor the data from the pressure transducers.

![Full View of the Front Panel](image)

Figure C.1 Full View of the Front Panel

The upper left corner of the front panel, as shown in Figure C.2, is the main control region. The “start” button is for turning the program on. The sampling rate and the required total number of samples can be set at the right side of the control region. Once the start button is clicked, the data acquisition starts and will complete when the total number of samples is satisfied. The sub panel in the middle is for setting the
physical channels of the digital output to trigger the camera. The device name under start button is for the physical channel of the analog input from the pressure transducers. The “write to file” button is to control for the captured data to be written on the disk.

Figure C.2 Upper Left Corner of the Front Panel

The lower left part of the front panel, which is shown in Figure C.3, includes the signal form setting of the digital trigger, the table of the recorded data and the program error report. The camera trigger can be modified by changing the width of the signal and the delay time between the start button and the actual capturing. In the recorded data table, each column is for one specific pressure transducer. The data in one row is captured at the same time.

Figure C.3 Lower Left Part of the Front Panel
The right side of the front panel, shown in Figure C.4, is for the monitors of all the pressure transducers and the fast Fourier transform (FFT) results. For each of the monitors of the feedback tank and control pressure transducers, there is also a numeric indicator to show the estimation of the oscillation frequency from the FFT. The monitor at the lower left corner is for checking the FFT calculation of the upper feedback tank pressure.

Figure C.4 Real-Time Monitors in the Front Panel
C.2 Block diagram

The full view of the block diagram is shown in Figure C.5. Each section of the block program will be introduced next.

Figure C.5 Full View of the Block Diagram
Figure C.6 shows the sub-block diagram for calculating the delay and width of the digital signal, which is the trigger for the camera capturing, and transferring the digit signal from Boolean to real number format.

![Figure C.6 Sub-block Diagram 1](image)

Figure C.7 is for setting the data acquisition system, includes the data range, sensor type and the physical channel connection.

![Figure C.7 Sub-block Diagram 2](image)
Figure C.8 is for synchronizing the analog input and the digital output and sending the error message to the front panel.

![Figure C.8 Sub-block Diagram 3](image1)

Figure C.8 Sub-block Diagram 3

Figure C.9 is showing the sub-block diagram for transferring, sending and recording the analog input from the pressure transducers.

![Figure C.9 Sub-block Diagram 4](image2)

Figure C.9 Sub-block Diagram 4

The Labview code is included in the accessory CD kept with the hard copy of the thesis in the Mechanical, Automotive and Materials Engineering Department.
Vita Auctoris

NAME: Sichang Xu

PLACE OF BIRTH: Kunming, Yunnan, China

YEAR OF BIRTH: 1990

EDUCATION:

University of Windsor, Windsor, Ontario
2010-2014 B.A.Sc.

University of Windsor, Windsor, Ontario