Reduction Of Vortex-driven Oscillations In A Solid Rocket Motor Cold Flow Simulation Through Active Control

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REDUCTION OF VORTEX-DRIVEN OSCILLATIONS IN A SOLID ROCKET MOTOR COLD FLOW SIMULATION THROUGH ACTIVE CONTROL

by

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B.S. Southwestern Oklahoma State University, 2001

A thesis submitted in partial fulfillment of the requirements for the degree of Master of Science in the Department of Mechanical, Materials, and Aerospace Engineering in the College of Engineering and Computer Science at the University of Central Florida Orlando, Florida

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Abstract

Control of vortex-driven instabilities was demonstrated via a scaled-down, cold-flow simulation that modeled closed-end acoustics. When vortex shedding frequencies couple with the natural acoustic modes of a choked chamber, potentially damaging low-frequency instabilities may arise. Although passive solutions can be effective, an active control solution is preferable. An experiment was performed to demonstrate an active control scheme for the reduction of vortex-driven oscillations. A non-reacting experiment using a primary flow of air, where both the duct exit and inlet are choked, simulated the closed-end acoustics. Two plates, separated by 1.27 cm, produced the vortex shedding phenomenon at the chamber’s first longitudinal mode. Two active control schemes, closed-loop and open-loop, were studied via a cold-flow simulation for validating the effects of reducing vortex shedding instabilities in the system. Actuation for both control schemes was produced by using a secondary injection method. The actuation system consisted of pulsing compressed air from a modified, 2-stroke model airplane engine, controlled and powered by a DC motor. The use of open-loop only active control was not highly effective in reducing the amplitude of the first longitudinal acoustic mode, near 93 Hz, when the secondary injection was pulsed at the same modal frequency. This was due to the uncontrolled phasing of the secondary injection system. A Pulse Width Modulated (PWM) signal was added to the open-loop control scheme to cor-
rect for improper phasing of the secondary injection flow relative to the primary flow. This addition allowed the motor speed to be intermittently increased to a higher RPM before returning to the desired open-loop control state. This proved to be effective in reducing the pressure disturbance by approximately 46%. A closed-loop control scheme was then tested for its effectiveness in controlling the phase of the secondary injection. Feedback of the system’s state was determined by placing a dynamic pressure transducer near the chamber exit. Closed-loop active control, using the designed secondary injection system, was proven as an effective means of reducing the problematic instabilities. A 50% reduction in the FFT RMS amplitude was realized by utilizing a Proportional-Derivative controller to modify the phase of the secondary injection.
I dedicate my thesis to the women in my family who have always been an inspiration to me!

To my grandmother Elene Ward who taught me how to be gracious when faced with adversity.

To my grandmother Margarett Hancock who taught me what it means to be open-minded.

To my mother Johnnie Sue Hancock who gave me a love for knowledge, learning and truth.

To my twin sister Jeni Suezette Ward whose gentle spirit and unbounded determination inspires me everyday.

To these women I dedicate my work.
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9.5 Frequency spectrum of the primary flow’s characterization signal compared with the closed-loop control signal shown in Fig. 9.4.
Combustion systems can be adversely affected by instabilities caused by internal acoustic resonances. An SRM is a combustion system that is especially vulnerable to acoustic instabilities because of its natural geometrical configuration [3]. The internal geometry of SRMs is conducive to vortex shedding to occur. Inhibitors from segmented grain assemblies as well as various grain geometries, provide ideal impingement surfaces for vortices to shed at low frequencies. The frequencies caused by vortex shedding can be at or near the natural acoustic modes of the motor itself. These frequencies will excite one of the motor’s natural acoustic modes which will cause an increase in the resonance amplitude. This phenomenon can be potentially harmful to the motor itself as well as the payload that it is carrying. The resonant frequencies produced can be transferred to the payload and damage sensitive systems.

Mechanical systems that suffer from the negative effects of vortex shedding require a technique that will eliminate this phenomenon. Passive control, such as strategic inhibitor placement, have been used to eliminate vortex shedding. Such solutions consist of modifying
the physical geometry of the system with the intent of removing all characteristics that would contribute to the development of shedding vortices at a resonant frequency. However, this solution is dependent on continuous system redesigns and tedious amounts of analysis [4,5]. An alternate solution is to utilize an active control scheme. In order to elevate pressure disturbances found in various mechanical systems such as Solid Rocket Motors (SRMs), turbines and ramjets, a secondary injection method is developed to be used to cancel out acoustic noise in any closed-ended chamber. Utilizing pulses of air as the control actuation, the pressure disturbance caused by vortex shedding is suppressed. Destructive interference between the pressure disturbance, caused by vortex shedding and the suppression system’s flow, the two longitudinal modal frequencies, cancel when they are both superimposed and 180 degrees out of phase. When these conditions are achieved the disturbance from vortex shedding will be eliminated. The success of this control concept will show that vortex shedding instabilities found in certain mechanical systems can be reduced or eliminated by using a pulsed flow as the secondary injection technique and implementing an active control scheme.

A more specific description on how vortex shedding in a closed-ended chamber, such as a SRM, creates a pressure disturbance increase is illustrated in Fig. 1.1. The vortex driven instabilities that give rise to the amplified pressure disturbances are caused by a continuous flow impinging on a protruding surface. When the flow impinges on the protruding surface the flow area is decreased and turbulent conditions are created. The pressure in the downstream region is reduced and a pressure disturbance is created. Shown in Fig. 1.1,
the chamber pressure shows a signal response that reflects the excitation inside the cham-
ber. This response contains the frequency that is being excited. When the vortex driven
frequency couples with the natural acoustic frequency of the chamber the overall pressure
variance increases. The elimination of the excited frequency response will specifically be
evaluated and controlled.

When a suppression method, such as pulsed air flow injection, is implemented the pressure
variance should decrease. Figure 1.2 shows how the pressure’s variance signal would appear
with the secondary injection system employed. When a properly phased wave of air is injected
into a chamber where a vortex shedding pressure disturbance exists, destructive interference
will occur. The pulsed air flow must be injected 180 degrees out of phase with the disturbance
pressure wave in order to achieve optimal reduction in the pressure disturbance.

Figure 1.1: Conceptual diagram of how vortex shedding causes disturbances in SRMs.
Figure 1.2: Conceptual diagram of how secondary injection will reduce the pressure disturbances in SRMs.

The development of an experimental cold-flow simulation was first needed in order to test the control theory in the elimination of vortex shedding instabilities. Chapter 3 addresses a detailed description of the experimental setup and sets the foundation of the overall experiment. Both the primary flow and secondary flow designs are developed and discussed. Then in Chapter 4 the primary flow or cold flow simulation is demonstrated. Without a proper simulation of vortex shedding excitation a control implementation would be fruitless. The validity of using such a setup to reproduce the vortex shedding phenomenon must be shown to be feasible. Following the successful characterization of the primary flow an analogous pressure signal is matched by the secondary injection flow in Chapter 5. The characterization of the injection mechanism was done so as to define the secondary injection flow to match the primary flow’s pressure signal within the smallest error possible. This
flow is next introduced into the flow chamber along with the primary flow so as to see how it influences the reduction of the primary flow. Control is added to the secondary injection flow by using a LabView virtual instrument program design to implement open-loop control schemes followed by closed-loop control scheme. The hardware integration with the software is developed in Chapter 6. All software and hardware control was developed using LabView. It was also used for all data acquisition and analysis. The LabView software development is described in detail in Chapter 7. In Chapter 8 both the open-loop control and closed-loop control schemes are evaluated for their ability to cause a reduction in the pressure disturbance caused by vortex shedding.

Open-loop and closed-loop control schemes are explored as possible means of actively eliminating vortex shedding. It will be seen that using a closed-loop control scheme is effective in reducing the pressure disturbances caused by vortex shedding. The methods and control schemes that are examined should not be limited to just SRMs but can be implemented in other systems that experience vortex driven instabilities. This technology could be a promising solution for many systems facing such problems in addition to SRMs. The successful use of active control methods in vortex shedding instability elimination would provided Engineers with greater design flexibility, options, and reliability.
CHAPTER 2

LITERATURE REVIEW

Particular attention to motor instabilities started to surface between prediction models and real systems when discrepancies were found. Motors that had been developed had failed due to low-amplitude pressure oscillations but had been modelled as perfectly stable. Discrepancies between prediction and actual implementation suggested that some phenomenon was happening that was not being predicted. Inaccuracies in the models were such that even after correcting propellant-combustion characteristics, major problems still persisted [6]. Thus, new research efforts were turned towards understanding SRM instabilities. An essential tool for understanding this phenomenon is predicting the instabilities via an accurate simulation model or the development of a control scheme that will be able to physically eliminate and adapt with varying velocity flows. Brown and Dunlap [6] reported discrepancies found between predicted results and actual test data. They theorized that it must be due to some significant source of acoustic energy that has not been recognized. They found that simultaneous pressure oscillations were observable when vortices were formed.
The frequency at which vortices are shed depends on the mean flow velocity [7] and is often at or close to the natural acoustic modes of the duct where they are formed. The low-frequency, low-amplitude oscillations that are attributed to vortex shedding are harmless on their own, however, when the frequency that is being shed is at or near the chamber’s natural acoustic mode, coupling will take place causing an increase in amplitude and resonance. Resonance is a very undesirable phenomenon in SRMs as it can cause damage to the onboard payloads, decrease the thrust of the motor itself and cause catastrophic failures [6]. When engineers are designing a new SRM they are careful to design the motor so that the likelihood of resonance is kept at a minimum.

There are various types of instabilities common to combustion systems. These include acoustic instabilities, shock instabilities, intrinsic instabilities, and system instabilities. A detailed description of each instability was reviewed by Barrere and Williams [8]. Although other combustion instabilities do exist within the combustion chamber, the focus will be devoted to acoustic instabilities and will offer their finding on this specific case. Barrere and Williams report in [8] that the amplification sources for acoustic instabilities usually are concentrated close to the propellant surface, with in the combustion zone. The longitudinal mode occurs most easily when the energy release due to combustion takes place in a zone where the pressure variations are large, that is, in a system of standing waves, at a pressure antinode. Frequencies corresponding to longitudinal modes have been observed in chamber of all types. In propulsion chamber, the exhaust is generally supersonic and the fundamental mode corresponds to a pipe closed at both ends.
New ideas and innovative techniques are developed and studied by researchers for the elimination of vortex-driven instabilities in various mechanical systems. A controlled solution for the elimination of vortex-driven instabilities is looked at currently as a possible solution which will provide design engineers with a viable option to this problem than past, passive solutions. A review of the problem as well as the research that has been done to date on vortex shedding instabilities is presented in the following sections.

2.1 Cold-flow Simulation Techniques

When performing experiments in a laboratory environment, it is convenient to use cold gases as the working fluid for a scaled-down SRM simulation. The cold gas is advantageous because it is nonreactive and it will produce the flow characteristics belonging to SRMs and other combustion systems. When utilizing cold gas as the working fluid, the temperature fluctuations are not accounted for, but this is not a primary concern when working with the system acoustics. Cold-flow models have proven to work very well in simulating the acoustic phenomena found in SRMs [7]. Culick and Magiawala were the first to use a cold-flow simulation as an appropriate means of reproducing this phenomenon and demonstrating it in a laboratory environment [9]. They proved through their studies that placing pairs of rings a necessary distance apart, at the node of the fundamental mode of a chamber, allowed for the production of vortex shedding pressure oscillations. They also conducted tests which modified the distance between plates and the mean flow velocity to study the relationship
between the vortices being shed and the acoustic modes of the chamber, proving that the frequencies produced by the vortices are proportional to the mean flow velocity, $U$, over the distance between the plates, $l$, [9]

$$f_t \propto \frac{U}{l} \quad (2.1)$$

Disturbances are shown to grow in magnitude if the energy gain is greater than that lost through radiation at the end of the duct. Strong passive fluctuations may be set up in that way [10].

The significance of combustion oscillations is that such oscillation can be harmful to an extent ranging from mild noise, through causing flame leakage and vibration of doors or loose parts, up to the extreme when they cause a high-pressure combustion chamber to explode or a low-pressure one to implode. It was reported by Thring in [11] that oscillatory combustion had been observed to give increased combustion intensity for the combustion of liquid drops and gas-borne slid-fuel particles, and can give an enormous increase in the rate of convective heat transfer from the flame to the walls of the combustion chamber. It was also reported that combustion instabilities occur in liquid-propellent engines as well and solid-propellent motors.
2.2 Passive Control

The discovery of instabilities in any design after production can be very time consuming, difficult, and expensive to resolve [4]. Both passive and active control methods have been visited by engineers and researchers alike to produce a means of finding an easy yet efficient way of solving this problem.

Passive control techniques are the most common methods of dealing with combustion instabilities to date. Passive control encompasses different methods in altering the physical systems being studied. Schadow and Gutmark [5] showed different methods of passive control for dump-type combustors. Decoupling the mechanisms by driving them out of phase and changing the geometric design of the inlet duct so that small-scale turbulence is produced instead of large-scale turbulence were some of the passive approaches tested. Strategically placing plates at pressure antinodes and reducing the combustor length by adjusting the inlet was a method of geometric modification to reduce the effects of pressure instabilities. It was shown by Schadow and Gutmark in [5] that this was an effective means of reducing pressure instabilities. However, this method is only practical if the geometry of the combustor can be varied according to the acoustic criteria.

The Titan IV Motor went through two redesigns before it successfully completed structural testing since it was experiencing pressure fluctuations that were not common in the analysis for prefiring predictions. These pressure fluctuations were particularly noticed because the motor had a catastrophic failure in both of the tests done before successful com-
pletion after the total redesign. The Titan IV motor tests proved the need for some type of combustion instability control [12].

While removing inhibitors or flow constrictions in the motors seems like a simple and effective means of passive control, this is not always the case. Acoustic instabilities have been seen in motors where no inhibitors were used. In the Titan IV upgrade, inhibitors were eliminated all together, yet pressure instabilities were still present. It was later found by Dotson et al. that the acoustic feedback can also induce pressure oscillations [13]. Therefore a geometric adjustment to the motor’s design is not always the complete solution, requiring careful attention when designing motors such that instabilities are minimized yet efficiency is maximized. This approach can be a very tedious and costly endeavor. Thring reported in [11] a possible cure for eliminating combustion instabilities. One way is to introduce additional damping, by baffles in the combustion chamber, or a hole in the wall of the combustion chamber. Another is by using a higher air pressure an velocity in the inlet port to reduce the effect of small combustion-chamber-pressure fluctuation upon the instantaneous air-inlet rate. Also move the fuel inlet away from the axis can inhibit the generation of combustion instabilities.

Sivasegarm and Whitelaw in [14] proved that the suppression of oscillations in ducts with an acoustically closed end by proper location of an orifice is possible for upstream length exceeding 14 diameters in 40-mm ducts, and appears to be achieved by the interaction between a quarter-wave frequency in the duct section between the orifice and the flame
holder and a half-ave in the duct section upstream of the orifice. Also, a smooth convergent-divergent nozzles could be used in place of orifices for the purpose of suppressing oscillations.

2.3 Active Control

Active control schemes have been evaluated through various experimental trials. As early as 1952, active control methods for the reduction of combustion instabilities were considered [7]. One means of suppressing vortex-driven oscillations is via a secondary injection method. Some other types of actuation methods have been used, such as a loud speaker. Loud speakers are the common way found in the literature for instability control [10,15–17]. In particular, a digital phase-shift in the pressure signal response was outputted to the chamber via a loud speaker. However, Gulati and Mani [18] showed that this would only work for very low-amplitude instabilities and would not be a good means of reducing pressure instabilities in practical combustors (< 5% peak-to-peak fluctuations) [7].

Another factor in combustion system which contributes to instabilities caused by vortex shedding is the heat of the combustion process itself. This mechanism by which energy can be transferred was proposed by Lord Rayleigh in 1878 [19]. He proposed that, “If heat be given to the air at the moment of greatest condensation, or be taken from it at the moment of greatest rarefaction, then vibration is encouraged. On the other hand, if heat be given at the moment of greatest rarefaction or abstracted at the moment of greatest condensation, the vibration is discouraged.” Sterling and Zukoski in [19] concluded that, “if equal amount
of heat are added in and out of phase with the pressure oscillation then the vibration is neither encouraged nor discouraged by the heat addition”.

Active feedback has been used to reduce the characteristic combustion instabilities of systems. The use of software to observe pressure oscillation characteristics such as amplitude, frequency, and phase was used in the feedback processes. The so called observer, developed by Sattinger et al. in [20], could detect modes of the instabilities and modify the fuel flow into the combustion chamber. This method of actuation proved to reduce the pressure oscillation by factor of 4.

Different types of active control methodologies are developed for the elimination of vortex-driven instabilities in combustion systems. An adaptive control strategy was demonstrated by Padmanabhan et al. [17] which utilize actuators which simultaneously control volumetric heat release and pressure fluctuations and sensors that measure combustor performance. Their approach was based on the Downhill Simplex algorithm. This strategy sets out to minimize what Padmanabhan et al. call a cost function and keep it at a minimum. The cost function is related to the pressure fluctuations and the volumetric heat release. As pressure fluctuation decreased and/or the volumetric heat release increases the cost function decreases, therefore, the cost function is at a minimum when the combustor performance is optimized.

Davis et al. in [21] used a hybrid algorithm which combines the hill-climbing and downhill simplex algorithms. This control algorithm modified the combustor condition for geometrical manipulation.
Petersen and Murdock [2] used air pulsed at a fixed frequency as the actuation method. They proved that open-loop control was obtainable through the use of this method of actuation, proving that the vortex shedding mechanism can be directly affected by this type of injection method. They theorized from their results that the vortex shedding mechanism can be directly controlled once a more efficient control algorithms was developed [22]. The present research is a continuation of the work of Petersen and Murdock, with the goal of demonstrating a closed-loop, active control technique using secondary injection.
CHAPTER 3

EXPERIMENTAL DESIGN

The experimental design is comprised of two distinct parts. The first is the design is what the author will call the primary flow. It is designed to model the vortex shedding instability phenomenon seen in flow systems. The cold flow simulation will reproduce, in a controlled and scaled down setting, the pressure disturbance behavior that is present in various mechanical systems, particularly SRMs. The secondary injection, or what will at times be referred to as the secondary flow, is a specially designed flow actuator that will be controlled using a LabView virtual instrument. This design will be implemented and used to rid the simulated cold flow system of its vortex-driven disturbances. The following two sections will lay out specific design details for both the cold flow simulation and the secondary injection system.
3.1 Cold Flow Vortex-Driven Simulation Design

The cold flow simulation is designed to produce vortex shedding disturbances. The orifice plate spacing and the chamber’s exit diameter are the two major design variables that must be adjusted to produce the resulting vortex shedding pressure disturbances.

The chamber design was inspired by previously reviewed designs and adapted from the design developed by Petersen and Murdock [2, 7, 22]. While Petersen and Murdock’s cold-flow rig operated under vacuum, the new setup described herein is pressurized. An overall cylindrical design was chosen over the plexiglas rectangular duct used by Petersen and Murdock in [7] because it allowed for experimental variables to be easily adjusted. A conceptual drawing of the experimental chamber is shown in Fig. 3.1. Petersen and Murdocks square chamber design have more point to seal which made it more complicated when opening the chamber to perform plate adjustments. The shape of the chamber itself does not change the phenomenon that is being observed and controlled. The interest is in the longitudinal pressure waves in the chamber not the radial pressure waves.

![Figure 3.1: Cold flow simulation conceptual chamber design.](image)

Given that the objective was to show the feasibility of using active control to eliminate vortex shedding instabilities; the simulation setup centered exclusively on the acoustic re-
duction of the phenomenon. This allowed for a simpler design; since no flow visualization was necessary at this stage. The motivating factors for the overall cold flow experimental design were pressure considerations and orifice plate placement. The chamber should be able to withstand gauge pressures that range from 0 psi to approximately 30 psi. It is capable of producing the vortex shedding that would couple with the natural acoustic modes of the chamber causing the acoustic instability phenomena seen in SRMs and other closed ended systems. The following complete chamber design is relevant to these two main attributes.

3.1.1 Constant Design Parameters

![Figure 3.2: Cold-flow model.](image)

The experiment required that the chamber be leak free so that choked conditions could be reached. When leakage did occur, silicon gel was used to stop the escaping flow. It was important for the system to be a close-ended system so that a standing wave would be produced at certain physical conditions. Poly vinyl chloride (PVC) pipe was chosen as the working material for the chamber because of its availability and its capability to
withstand gauge pressures up to 100 psi. Using PVC as the working material also allowed fewer places for leaking to occur because there are fewer points of connection. Adjoining parts were minimized as much as possible for this very reason. Figure 3.3 shows how the whole chamber appeared after design and system configuration.

Figure 3.3: Cold-flow experimental setup.

An important design dimension is the chamber length. This length governs the natural acoustic modes of the system. The natural acoustic frequency is dependent on the length of the chamber by

\[ f_i = \frac{an}{2L}, \quad n = 1, 2, ... \]  

(3.1)
where \( n \) is the longitudinal acoustic mode integer, \( L \) is the total length of the duct and \( a \) is the speed of sound. From Eq. 3.1, \( f_1 \) is inversely proportional to the length \( L \) of the chamber for a constant acoustic mode integer. Therefore, the longer the length of the chamber the lower the frequency. The chamber length was designed so that lower frequencies could be produced by the system. The length of the system was a primary design requirement due to the limitations of the secondary injection system. The experimental frequency needed to be designed for a frequency below 200 Hz. To allow for lower frequencies a chamber length of 179.39 cm was used. This produces a first longitudinal acoustic mode at 98 Hz, which is within our design limitations. Specific chamber dimensions are shown in Fig. 3.2.

Other important design elements that are shown in Fig. 3.3 are the flow valves, the side injection, the coupler, clamps, and pressure gauges. The two primary flow valves are used to needle the air flow into the chamber. When the experiment is fully deployed, these two valves are fully open. The side injection is shown more clearly in Fig. 3.4.

Air flow from a building reservoir is split into two flows that enter into the chamber from opposite sides. The side injection creates a steady stream of air flow which aids in creating a uniform air flow inside of the cold flow chamber. The entry air flow must be placed before the injector plate. The injector plate, which is made of plexiglass, has a series of small holes placed uniformly apart to create a uniform flow. The injector plate placement can be seen inside the chamber in Fig. 3.2. The injector plate is the point where the chamber begins. It is placed 4.13 cm from the point of the primary injection. Therefore, the chamber entry wall to exit wall is 175 cm. The injector plate has a diameter of 4.04 cm.
The cold flow experimental chamber was mounted to a wooden platform by v-clamps. It was extremely important that the whole apparatus be held together tightly or the pressure from the air flow would cause the chamber to separate at the coupling section which holds the fore and aft sections of the chamber together as well as houses the orifice plates. The wooden platform and v-clamps that hold the chamber can be seen in Fig. 3.3.

Special taps are placed at the aft end of the chamber to mount the steady state pressure gauge and piezoelectric PCB 106B pressure transducer that was used in characterizing the pressure waves formed from the flow field inside the chamber. These are located at a distance of 15.24 cm and 5.08 cm, respectively, from the aft end of the chamber and can be seen in Fig. 3.5. These sensors will be used to measure the pressure perturbations inside the chamber.

Over all the chamber design of the cold flow simulation is relatively simple. The structural design’s simplicity ensures less points of failure and easy of modification and fine tuning.
3.1.2 Variable Design Parameters

As stated before, the chamber’s design variables include the orifice plate spacing and the exit diameter. These two design parameters are what is adjusted to produce the vortex shedding phenomenon inside the cold flow chamber.

Spacing of the orifice plates contributes to the vortex shedding occurring at a frequency close to one of the longitudinal acoustic modes of the chamber. Because of this, thought was given to the easiest possible means of adjusting this parameter with accuracy. It was apparent that the spacing between the two plates would have to be modified frequently until the vortex shedding conditions were realized. The spacing of the orifice plates is a crucial tuning parameter in the experiment.
The spacers were used to adjust the distance between each orifice plate. A PVC coupling piece was used to hold the spacers and the orifice plates together under pressure. A double o-ring was machined for both sides of the coupler such that no leakage would occur. This worked very well for stopping all leaks in this area.

The orifice plates were a specific design consideration. The orifice plates were designed to be 0.64 cm thick with a sharp knife edge created by a tapered inner hole of diameter 1.91 cm. The orifice plates were tapered on the side that faces away from the flow. This allows for the flow to impinge on the sharp knife edge. This sharp edge simulates the effective flow through a SRM. Figure 3.6 shows the orifice plate design.

![Figure 3.6: Conceptual drawing of the orifice plate.](image)

The orifice plates’ outer diameter corresponded to the inner diameter of the chamber. The plates were epoxied inside two of the spacers that were cut so that they made a continuous smooth transition through the chamber. The spacers which were cut to 0.3175 cm in width, could be used in combinations to adjust the distance \( l \) between the orifice plates inside the coupler. The spacers had an outer diameter of 4.83 cm and an inner diameter of 4.04 cm. The spacers were just a continuation of the chamber through the coupler and were used to...
separate the orifice plates by a distance. The coupler used was a standard PVC coupler which was milled out to a inner diameter of 4.04 cm so that the spacers and orifice plates would fit perfectly inside. The coupler was sealed by double o-rings located on each of the connecting ends of the chamber.

The maximum distance that the orifice plates could be placed apart was 1.59 cm. This dimension was limited by the coupler used. The coupler was 6.35 cm in length. After the double o-rings were put in place, a total of 2.86 cm remained as working space. The orifice plates were each 0.635 cm wide leaving the amount of distance to vary the plates in reference to one another at 1.59 cm. Initially, the separation that could be obtained from this setup seemed sufficient. This assumption was based on the preceding experimental setup done by Petersen and Murdock. To obtain the acoustic resonance at approximately 200 Hz, on orifice spacing of 3.2 cm with a chamber length of 85 cm was used. As seen in Fig. 3.2, it was chosen to have the new chamber length of 175.26 cm which is almost double that used by Petersen and Murdock. This parameter design was decided upon because it allowed for the chamber acoustic modes to be realized at lower frequencies. This was ideal because the secondary injection scheme that was utilized as the means of eliminating the acoustic disturbances was limited to frequencies below 300 Hz. Effects of the secondary injection system on the overall design are discussed in more detail in Chapter 4. The flow diameter of the chamber as seen in Fig. 3.3 is 4.04 cm. The diameter of the overall chamber was somewhat arbitrary therefore a standard size PVC pipe was chosen.
The place where the flow exits the chamber was also a parameter that had to be modified to meet experimental conditions. The exit was located at the aft end of the chamber and was located in the middle of another end cap. The design of the exit diameter is discussed in Chapter 4.

### 3.2 Secondary Injection Design

There are two separate flows that must enter into the chamber. One is the primary flow that is responsible for the formation of the vortex shedding phenomena of interest, and the second is the secondary flow. The secondary flow is used to eliminate or control the vortex shedding mechanism caused by the primary flow in the chamber. The flow for both will enter the chamber at the front end of the chamber. Swagelok connections were used to facilitate this point of entry.

![Model of the secondary injection system chamber entrance.](image)

The secondary flow enters at the very front end of the chamber in the middle of the end cap. The secondary flow begins at the injector plate. A tube fitting is set flush with the
injector plate where the secondary flow will enter into the chamber. Figure 3.7 shows how the secondary flow enters into the chamber.

The secondary injection design was in need of a fast acting flow valve. The valve was need to intake air and output it at a particular frequency. The author used a converted model airplane engine to act as the flow valve. The whole setup was limited by how quick a pulse could be created. This system was only able to work in the frequency range of 0 to 200 Hz.

The secondary injection consists of a two-stroke model airplane engine that is powered by a DC brushless motor. The two-stroke model airplane engine is used as an intake piston to output a pulse of air at a set frequency. This secondary injection hardware setup is based on that employed by Petersen and Murdock [2]. The model setup is shown in Fig. 3.8.

Figure 3.8: Secondary injection flow setup.
A secondary injection flow is used as the suppression method in eliminating the acoustic instabilities caused by vortex shedding in the duct. The DC motor is controlled by a Minarik MM23201 Series DC controller box. The motor is able to spin at a maximum of 2500 rpm. By itself it would be able to produce frequencies no greater than 41 Hz. A gear reduction of 7.2:1 is used, resulting in a maximum achievable frequency of 300 Hz. The two-stroke converted model airplane engine is rated up to 18,000 rpm, therefore, the secondary injection flow system is limited by the motor and not the engine. A closer look at this configuration can be seen in Fig. 3.8

A 0.32-cm copper tube is connected to the air intake valve (Fig. 3.7). The copper tube is flush with the primary injector plate. This allows for the secondary flow to start at the beginning of the primary flow, at $x = 0$. A needle valve is placed between the intake piston and copper injector tube. This valve is used to modify the flow rate of air that enters the duct.
CHAPTER 4

CHARACTERIZATION OF THE PRIMARY FLOW

As introduced in Chapter 3, the primary flow is the label given to the flow which is responsible for the formation of the vortex shedding phenomenon. The goal to be achieved from the cold flow experimental setup was to reproduce vortices that shed at frequencies close to or at the natural acoustic modes of the chamber in which they occur. When these vortices are shed at frequencies close to the modes of the chamber, they excite these modes and an increase in the amplitude can be seen.

The flow initiates from an external pressurized air reservoir coming from the building’s supply. The flow is injected into the chamber via side injection. The flow passes from the pressurized reservoir into the fore end of the chamber via two ports located on opposite side of the chamber. As soon as it enters the chamber it is contained in a space of approximately 1.91 cm where it resides until it passes through the entrances plate creating a more uniform flow field. It continues to flow downstream until it reaches the pair of orifice plates which simulate the impingement mechanisms inherent to systems such as SRMs, turbines, and ramjets.
A back pressure of 1 atm was maintained to ensure that choked conditions produce a pseudo-wall at the exit. The chamber now behaves like a closed-closed chamber, meaning closed at the inlet and closed at the outlet. With this configuration a standing pressure wave is created inside the chamber.

Orifice plates are strategically placed near an acoustic pressure antinode, which is located approximately at the center, to produce vortex-driven oscillations. In order to justify the orifice plate design and placement, it is useful to examine the theory behind the physical phenomena that contribute to the vortex shedding instabilities in various systems. The process of vortex shedding creation is shown in Fig. 4.1, where $U$ is the velocity flow, $l$ is the orifice spacing, $D$ is the chamber diameter, and $d$ is the orifice plate diameter. Vortex shedding will occur in a closed duct when a gas or fluid flow impinges on some protruding or slotted surface, causing a restriction in the upstream flow which reduces the flow area and causes the upstream pressure to rise [4].

![Vortex Shedding in a Duct](image)

Figure 4.1: Vortex Shedding in a Duct.
After the formation of vortices, due to the first orifice plate, the turbulent flow continues to traverse downstream until it again impinges on the second orifice plate. Here an acoustic signal is formed. This acoustic signal is reflected back upstream. As it travels upstream it aids in the organization of the vortices that are forming. This process is called acoustic feedback. These vortices continue downstream where the process is repeated again. When the vortices create an acoustic signal that is close to that of the natural acoustic modes of the chamber in which they occur, coupling takes place. When coupling occurs between the acoustics caused by the shedding vortices and the chamber’s natural acoustic modes there will be an increase in amplitude and resonance [23]. This process, which is shown in Fig. 4.1, causes additional acoustic energy in systems, leading to unwanted instabilities.

Table 4.1: Calculated longitudinal acoustic modes of the cold-flow simulation experimental setup.

<table>
<thead>
<tr>
<th>Axial Mode</th>
<th>$f_{cal,1}$, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>98</td>
</tr>
<tr>
<td>2</td>
<td>196</td>
</tr>
<tr>
<td>3</td>
<td>294</td>
</tr>
<tr>
<td>4</td>
<td>392</td>
</tr>
<tr>
<td>5</td>
<td>490</td>
</tr>
</tbody>
</table>

The fundamental longitudinal frequency modes, $f_1$, of the flow chamber can be calculated by,

$$f_1 = \frac{an}{2L}, \quad n = 1, 2, ...$$  \hspace{1cm} (4.1)
where \( n \) is the longitudinal acoustic mode integer, \( L \) is the total length of the duct and \( a \) is the speed of sound. The calculated modes of the duct used for this experiment can be found in Table 4.1. Vortices should be shed around the values shown in Table 4.1 so that acoustic coupling occurs inside the chamber resulting in an increase in amplitude. Table 4.2 shows the experimental study done to obtain vortices shedding at 98 Hz.

<table>
<thead>
<tr>
<th>( d_{\text{exit}}, \text{cm} )</th>
<th>( l, \text{cm} )</th>
<th>( f_s, \text{Hz} )</th>
<th>RMS Amp</th>
<th>( F_{\text{Amp}} )</th>
<th>( P_{\text{gauge, psi}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.32</td>
<td>0.32</td>
<td>93</td>
<td>0.04</td>
<td>100</td>
<td>18</td>
</tr>
<tr>
<td>0.32</td>
<td>0.64</td>
<td>289</td>
<td>1.7</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.32</td>
<td>0.95</td>
<td>289</td>
<td>0.11</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.40</td>
<td>0.64</td>
<td>480</td>
<td>0.12</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.40</td>
<td>0.95</td>
<td>480</td>
<td>0.12</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.04</td>
<td>0.64</td>
<td>289</td>
<td>0.08</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.04</td>
<td>0.95</td>
<td>480</td>
<td>0.11</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.04</td>
<td>1.27</td>
<td>480</td>
<td>0.13</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.04</td>
<td>1.59</td>
<td>289</td>
<td>0.09</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.52</td>
<td>0.32</td>
<td>93</td>
<td>0.025</td>
<td>100</td>
<td>15</td>
</tr>
<tr>
<td>0.20</td>
<td>1.27</td>
<td>93</td>
<td>0.065</td>
<td>100</td>
<td>18</td>
</tr>
</tbody>
</table>

Due to the limitations of the suppression method, the excited mode needed to be near 100 Hz corresponding to the first longitudinal mode. Initially, tests were done by holding the exit diameter at 0.32 cm. A shedding frequency of 93 Hz was seen with this exit diameter at an orifice plate separation of 0.32 cm. However, a signal conditioner amplification factor of 100 was necessary to get an RMS amplitude of 0.04 volts. Vortex shedding was achieved by this setup which proved that with certain conditions this setup was capable of producing acoustic instability phenomena. Noting that the setup was sufficient for experimentation, another series of tests was done to determine if a stronger acoustic signal could be produced.
by the setup. By varying the spacing of the orifice plates, reducing the orifice diameter and reducing the mean flow velocity, the desired excitation of the first longitudinal mode was obtained.

The exit diameter was varied in order to decrease the mass flow rate,

\[ \dot{m} = \rho A v. \]  \hspace{1cm} (4.2)

By decreasing the diameter the flow velocity the area increases and the pressure also increases. A series of experiments was done by first varying the spacing between the two orifice plates. In Fig. 4.2 a simplified representation of orifice plate separation is shown. The orifice plate diameter, \(d\), was fixed at 1.91 cm and \(l\) was the dimension that was varied. For this particular setup, the easiest excited mode was at approximately 289 Hz, the third natural longitudinal acoustic mode of the chamber. With the same exit diameter, a stronger signal was not found around 98 Hz. Therefore, it was concluded that for an exit diameter of 0.32 cm, the orifice plates spacing, \(l\), would have to be increased to achieve better results. However, due to the experimental design, the orifice plate spacing was limited to 1.59 cm.

![Figure 4.2: Plate spacing model.](image)
The distance that the orifice plates needed to be placed with an exit diameter of 0.32 cm was double of what was achievable with the design. A series of trials was performed to decrease the exit diameter so that there would be a decrease in internal flow velocity and an enhanced vortex shedding signal at 98 Hz. It can be seen in Table 4.2 that the 1st longitudinal mode was excited using an orifice plate separation of 1.27 cm and an exit diameter of 0.2 cm, resulting in a RMS amplitude of 0.065 with an amplification factor of 100. Figure 4.3 shows the spacers and the impingement plates that were used in the actual experiment.

The output signal from the final characterization of the primary flow is show in Fig. 4.4 and Fig. 4.6. A dynamic pressure signal for the baseline vortex shedding instability is provided in Fig. 4.4.
Figure 4.4: Dynamic component of the primary flow pressure signal. \( P_c = 18 \text{ psig} \).

The peak-to-peak fluctuations represent 3.3\% of the average \( P_c \). Figure 4.5 shows the pressure fluctuations in reference to the chamber pressure. The corresponding frequency spectrum is shown in Fig. 4.6, where the 93 Hz mode is shown to have a 0.065 RMS amplitude, typically.

The results clearly show that the vortex shedding mechanism was exciting the first longitudinal chamber mode at 93 Hz, which is very close to the calculated first longitudinal mode of 98 Hz which can be seen in Table 4.1. The primary flow characterization was accomplished with satisfying results. This became the signal that was actuated upon.
Figure 4.5: Percent pressure fluctuations of the vortex induced acoustic signal with a $P_c = 18$ psig.

Figure 4.6: Frequency spectrum of the primary flow.
CHAPTER 5

CHARACTERIZATION OF THE SECONDARY FLOW

From the primary flow experiment, the desired frequency that must be suppressed was 93 Hz. Data reported by Petersen and Murdock [2] showed a calibration for the DC brush motor controller voltage input as shown in Fig. 3.8. Referencing Fig. 5.1, to pulse the intake valve at 93 Hz, the voltage input to the DC brush motor controller must be 734 mV.

It was found by experimentation that to pulse the intake valve at 93 Hz, the DC motor must be powered at 770 mV. The calibration used from Petersen and Murdock proved reliable with an approximate 5% error. The resulting pressure signal produced from the secondary injection system is seen in Fig. 5.2 followed by Fig. 5.3 which show a dominant frequency at the desired 93 Hz required for suppression of the acoustic oscillations produced by the primary flow.

The secondary flow was adjusted through experimentation to match the primary flow’s characteristics. The secondary flow’s motor controller input voltage was varied until the 93 Hz frequency resulted. Subsequently, the amplitude of the primary flow needed to be
produced by the secondary flow so that total destructive interference is realized in the system. Recall that the primary flow’s amplitude was 0.065 RMS amplitude so this is the amplitude required of the secondary injection flow system. Substantial amplitudes can be reached using the secondary injection flow with this experimental setup. A needle valve located between the intake piston and the copper tube had to be adjusted to ensure that the the measured RMS amplitude of the secondary injection setup, seen on the frequency spectrum graphs, was at the RMS amplitude of the primary flow. The needle valve was varied until approximately 0.065 RMS amplitude was achieved.

However, it was discovered that the connection between the secondary flow and the primary flow was causing losses in the secondary injection system. There was a substantial
back flow from the primary flow. Due to the fact that these two systems are coupled the secondary injection’s characteristic amplitude was increased to adjust for the system losses. A 0.08 RMS amplitude was the designed characteristic amplitude of the 93 Hz signal for the secondary injection system. Figure 5.3 provides a dynamic chamber pressure trace with the secondary injection on at 93 Hz and no primary flow. The corresponding FFT is shown in Fig. 5.3. Along with the 93 Hz signal that is shown in Fig. 5.3 an additional peak at 193 Hz results as a harmonic of the secondary injection system.
Figure 5.3: Secondary injection flow FFT at 93 Hz, corresponding to pressure trace in Fig. 5.2.
CHAPTER 6

EXPERIMENTAL HARDWARE

The complete experimental setup consists of three main parts, the primary flow system, the secondary flow system, and the data acquisition system. The following sections provide details about each individual system’s hardware components and lays out the framework for integrating all three systems together.

6.1 Primary Flow Hardware Setup

The primary flow hardware setup consists of two pressure sensors, a signal conditioner, and two flow valves. The two pressure sensors, a piezoelectric PCB pressure transducer and a steady state pressure gauge were used to measure the dynamic pressure and internal chamber pressure respectfully. The steady state pressure gauge was not electronically connected to any other hardware. The steady state pressure transducer was monitored visually to ensure that an internal chamber exit pressure of 15 psig, allowing for a choked exit condition. Figure
Figure 6.1: Primary flow hardware configuration.

6.1 shows the sensor and flow valve locations on the actual cold-flow chamber where as Fig. 6.2 shows a closer view of how the pressure sensors were connected to the chamber.

Once a choked condition was produced this parameter was no longer adjusted. The pressure sensors were both threaded into the top side of the chamber at the exit. The piezoelectric PCB series 106B pressure transducer, seen in Fig. 6.3, is a highly sensitive sensor that measures the pressure perturbations in the air flow.

The dynamic pressure signal was measured by the PCB 106B pressure transducer mounted 15.24 cm from the chamber’s exit. This pressure transducer utilizes quartz plates to sense
the pressure perturbations. The fact that the quartz is rigid results in negligible diaphragm motion producing a sensitive sensor. Table 6.1 show the calibration data provided by PCB Piezotronics for the pressure transducer. Its sensitivity being 303.5 mV/PSI [1], which was used during the resulting analysis.

The signal conditioner is a digitally controlled amplifier. The signal conditioner provided a constant current excitation to the transducer over the signal line and decoupled the sig-
Table 6.1: Piezoelectric PCB model 106B, serial 6540, pressure transducer calibration test data [1].

<table>
<thead>
<tr>
<th>INPUT (PSI)</th>
<th>OUTPUT (mV)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.00</td>
<td>613</td>
</tr>
<tr>
<td>4.00</td>
<td>1213</td>
</tr>
<tr>
<td>6.00</td>
<td>1829</td>
</tr>
<tr>
<td>8.00</td>
<td>2421</td>
</tr>
</tbody>
</table>

Figure 6.3: Schematic of piezoelectric PCB pressure transducer, model 106B.

The signal from the DC bias voltage, the voltage in which the AC dynamic signal is centered or superimposed. The signal conditioner was the direct link between the piezoelectric pressure transducer and the data acquisition system.
6.2 Secondary Flow Hardware Setup

The secondary injection system’s hardware setup consists of a grounding isolator, Minarik MM23201 motor controller, a DC motor, a 2-stroke model airplane engine and a series of flow valves. A grounding isolator had to be added to the motor controller setup to prevent a short when all systems where connected to the computer via the BNC-2090 adapter chassis.

Figure 6.4: Ground isolator connection diagram.

Once the motor controller was reconfigured to be powered with an input voltage from the Virtual Instrument (VI) controller, grounding became an issue. Therefore the ground isolator was wired between the interface BNC-2090 chassis and the Minarik MM23201 motor controller, which can be seen in Fig. 6.7 and a clearer view of this setup can be seen in Fig. 6.6. Figure 6.4 shows how the ground isolator box was configured and Fig. 6.5 shows the
actual ground isolator box that was used in the experimental hardware setup. The voltage output to the motor was calibrated by tuning at the ground isolator. This provided a linear voltage output from the control VI to the motor controller box. The Minarik MM23201 motor controller converts the input AC signal from the controller VI to a DC signal which powers the DC motor. A Minarik MM23201 Series DC motor controller is connected to the the DC Motor that powers a two-stroke converted model airplane engine by a 7.2 : 1 gear reduction. The Minarik MM23201 Series DC motor controller requires 5 amps and a variable voltage input from 0-1.4 volts. The motor is able to spin at a maximum of 2500 rpm. The
Figure 6.6: Input/Ouptut signal hardware configuration.

gear reduction of 7.2:1 that is used between the DC motor and the two-stroke converted model airplane is rated up to 18,000 rpm (Fig. 3.8).

The secondary injection flow must pass through three valves which are placed between the engine and the chamber entrance. The first and third valves are used to turn the flow on and off and when the experiment is running are left completely open. The second valve is a needle valve that is used to reduce or amplify the amount of air flow from the secondary injection system. The primary flow hardware and the secondary injection hardware utilize the BNC-2090 adapter chassis to interface with the computer and LabVIEW software that are used for data acquisition and control.
6.3 Data Acquisition Hardware Setup

The data acquisition hardware setup consisted of a BNC-2090 adapter chassis and a Dell 2.4 GHz computer with 240 MB of RAM. Figure 6.6 shows the BNC chassis that was utilized to bridge hardware from the primary and secondary flow system with the data acquisition system. Connections between all hardware and software were done through the a National Instruments BNC-2090 adaptor chassis. Data collection from the pressure transducer was done using a National Instruments AT-MIO-16XE-10 board (16 channel, 100 kHz, 16 bit).

6.4 Hardware Integration

The primary flow, secondary flow, and data acquisition individual systems must be integrated into one functioning system. The hardware was configured such that data could be both sent and retrieved from the data acquisition system and computer controller. Figure 6.7 shows how these three systems are integrated. The hardware integration meets at the data acquisition BNC 2090 chassis. The data acquisition system must interface with both the primary and secondary flow systems for system analysis, characterization, and control.

The piezoelectric pressure transducers is connected to the BNC chassis. An output voltage from the pressure transducer sent to the data acquisition system via the BNC 2090 chassi. The signal is then passed to the AT-MIO-16XE-10 board, which is configured for recognition. Data processing and control output is done by a computer algorithm developed
as a virtual instrument. The computed output is then passed from the data acquisition system to the secondary injection system. The signal is first passed through the grounding isolator then onto the motor controller which translates the control signal into a motor power. The motor turns the intake valve that creates the pulsed flow. This complete system consisting initial of three individual systems work together in the elimination of vortex-driven disturbances.
CHAPTER 7

CONTROL SOFTWARE DEVELOPMENT

Computation, data acquisition and control were accomplished by using LabView 7.0 software. The controller software was developed in Laboratory Virtual Instrument Engineering Workbench (LabVIEW) which is a combination of hardware and software distributed by National Instruments. LabVIEW is a graphics-based programming language uniquely called G-programming.

The software is dependent on feedback from the pressure transducers to calculate the new input signal to the motor controller. The LabVIEW Virtual Instrument (VI) software was developed to control the phase of the pulsed flow frequency. The desired suppression state would result if the secondary injection’s flow frequency were 180 degrees out of phase with the vortex shedding frequency produced by the primary flow. This would allow the two responses to destructively interfere with each other resulting in a zero pressure signal at the frequency of interest. Hence, the vortex shedding frequency would be suppressed and eliminated. Data was acquired with a data acquisition rate of 9300 Hz.
The development of the pressure suppression control VI can be summarized into four sections. The software logic is respective of the section order. The following sections of this chapter will develop the functionality of the software in detail.

The open-loop control software and the closed-loop control software are similar and share some of the same functionality. The difference between the two software’s configurations is in the way the control output value is found. Figure 7.1 depicts how the logic flows from one subfunction to another in the open-loop software and Figure 7.2 depicts how the logic flows from one subfunction to another in the closed-loop software.

Figure 7.1: Open-loop software design.
The open-loop software’s input and output are not connected. The information coming from the sensors and being read by the data acquisition setup is independent of what is being inputted to the motor controller. The signal analysis gathers the sensor data and it is read into the memory and then outputted graphically to the user. The voltage that is being sent to the motor controller via the computer software is a constant value. In the actual program that was developed in LabView one VI was created that could be modified to be either the open-loop case or the closed-loop case. A button could be activated that would disconnect the control aspect of the closed-loop case and would just supply the desired constant voltage to the motor controller.
The closed-loop software uses the same signal analysis and control output functionalities, however, the control section is much more complex. The software development for the closed-loop controller will now be discussed in more detail. Figure 7.2 shows how the software is divided into four section and into seven functionalities. The first section, signal analysis, which contains functions which perform necessary input channel configuration, data acquisition and signal manipulation. The second section, control correction, uses the acquired data from the pressure sensors to calculate the new input value to the motor controller for the desired response. The third section, phase control, uses a Pulse Width Modulated (PWM)
signal to alternate the controller switching on and off at a set frequency. The final section, control output, sets up the physical channel to send the resulting modified analog voltage signal as the new input to the motor controller.

The complete development of the software VI can be divided into the following seven functions; 1) Acquire the signal from the pressure transducer in order to monitor the pressure perturbations in the chamber, 2) Take the FFT of the acquired signal in order to observe the frequency spectrum as well as the amplitude produced at the excited frequency, 3) Extract the amplitude of the FFT response at the excited frequency, 4) Calculate the error based on the assumption that the amplitude should be ideally zero at the excited frequency, 5) Employ a PD controller based on the calculated error, 6) Implement a Pulse Width Modulated (PWM) signal to control the amount of time that the controller is on, 7) Input the new control voltage to the control system. The logical flow of the LabVIEW VI is shown in Fig. 7.3. Each of these seven functions will be discussed in detail under their specific section.

7.1 Signal Analysis

Function 1 in Fig. 7.3 can be seen in more detail in Fig. 7.5. This function is responsible for obtaining the signal from the pressure transducer (sensor data). This function is set up for a voltage input. The maximum value, minimum value, physical channel where the hardware is located, number of samples and the rate of acquisition must be defined.
Figure 7.4 shows the front panel of the VI which is a graphical interface that was created to view results from the VI program and for user configuration of certain blocks in the VI program. The front panel includes any input variables that the user must define to complete the definition of the function block diagram panel. This is where such parameters from function 1 are defined. For the presented experimental setup, function 1’s parameters are defined as follows:
• maximum value = 8.2
• minimum value = -2.5
• physical channel = ai0
• number of samples = 1000
• rate of acquisition = 9300
• units = volts

The input signal comes from the pressure transducer. The pressure transducer sends a voltage signal to the (PCI) card where the analog to digital conversion is performed. The signal is then recognized by the computer software as a discrete value. The input signal

Figure 7.5: Input voltage from pressure transducer.
is configured to be obtained at 9300-Hz sampling rate. The sampled data is read by the software as a vector of 1000 values. The sampled data is then passed into a Fast Fourier Transform (FFT) Virtual Instrument (VI) block. Here the FFT is defined to give the best response possible.

This newly acquired data must be analyzed to obtain any useful information out of it. A FFT block included in LabVIEW was used to find and view the frequency response of the measured signal from the piezoelectric pressure transducer. Figure 7.6 shows this block and the configuration that was used for analysis.

![FFT block configuration](image)

Figure 7.6: Fast Fourier Transform (FFT) of the pressure signal.

Function 2 shows that the FFT block used was a spectral measurement of RMS magnitude, a Hanning window, an exponential weighting with 20 sampling averages.
The next function of importance to the functionality of the control software is the Extract Value block which is shown as function 3 in Fig. 7.3 and is shown in detail Fig. 7.7. Extract Value is another VI provided by LabVIEW. This function is importance for extracting the piece of data that is of interest to the programmer.

![Extract Value Block](image)

Figure 7.7: Extract amplitude of dominate frequency.

This block is used to find the amplitude at the frequency of 93 Hz which is defined because of the primary flow characterization.
7.2 Control Correction

The amplitude, if the controller were perfect, would ideally be zero. The actual RMS amplitude at 93 Hz is used to calculate the error of the system which is a function of the RMS amplitude (although the phase is the control variable). The difference between the actual value and the ideal value is calculated. The function that is used to calculate the error can be seen in Fig. 7.8. In the experiment, the absolute value of the error was taken since the actual value would never be a negative value.

![Diagram showing the calculation of error](image)

**Figure 7.8:** Calculate error.

The error value is then fed to a PD controller. Figure 7.9 shows the way the PD controller is developed in LabVIEW.

First, the proportional gain value is multiplied by the error that was calculated in the previous window. The derivative term is calculated by taking the difference in error over time, multiplied by the derivative gain value. The last function adds the proportional and
derivative terms together to calculate the total controller value. The PD gain values must be inputted by the user. This is ideal in that the PD controller can be changed in real time from the front panel. Therefore, the program doesn’t have to be stopped to modify the controller’s gain values.

7.3 Phase Control

The PD control output is multiplied by a PWM signal which was optimized for the controller. Utilization of the PWM signal allows for the controller to basically be turned on and off for a certain amount of time. This adjustment or modification to the signal will allow for the phase to be changed when needed and monitored when not in need. This value is the new input to the motor controller which in turn powers the DC motor and the
engine. Function 6 is a Signal generator VI. Figure 7.10 shows this function and its tuning parameters. A square wave signal has been generated with a frequency of 200 Hz, a duty cycle of 50% and amplitude of 1. The timing was tuned to match that of the system, meaning the samples per second are 9300 Hz, and the number of samples is 1000.

![Simulate Signal](image)

Figure 7.10: Apply control at a specific time, multiplication of PWM.

### 7.4 Control Output

There is an upper and lower bound set by the motor controller hardware. The motor controller works from 0 to 1.5 volts. However, the upper limit was set to 1.25 volts in order
to not stress the motor itself. The DC motor and engine are exceptionally sensitive to heat so care must be taken to keep them cool. One way to avoid overheating the DC motor or the engine is to not use the maximum voltage setting and not exceed any time limits. The input to the motor controller must be configured like for the input seen in Fig. 7.5.

Figure 7.11 shows the configuration for the output signal. The following are the limits set by the output block configuration:

- maximum value = 1.4
- minimum value = 0.0
- physical channel = ao0
- units = volts
The limiter that is mentioned above was programmed separately. The conditions for that limiter were programmed by using True and False functions which can be seen in Fig. 7.12.

This whole process is placed into an infinite loop that is stopped by a programmable control button. Therefore, this process is continuous. The phase of the two pressure waves is continuously adapted.
ACTIVE CONTROL SOLUTIONS

Active control solutions differ from passive control solutions in that an active controller is a powered system, rather than un-powered system such as strategic inhibitor placement. Passive solutions usually consist of modifying the geometrical characteristics of the system where as an active solution would make use of a measured signal and characterization of the pressure. [24]

An active control solution is studied as the method of controlling the phasing of the secondary injection system for the removal of the acoustic disturbance developed by the primary flow in the cold flow simulation. An active solutions is more desirable than a passive one because it is more adaptable to changing systems. Passive solutions, although sometimes effective in acoustic instability elimination, involve extensive re-design causing the loss of valuable time and money. SRMs vary from motor to motor and the motor itself changes during operation. The varying commercial and industrial challenges placed on SRM designs, as well as the nature of the changing functional demands, make active control a more viable choice than passive solutions.
The result of the perfect removal of problematic acoustic disturbances would require ideal control of the secondary injection’s phase and amplitude. The following is the development, implementation and testing of active control schemes to accomplish the best possible control for the elimination of the acoustic disturbances. Both open-loop and closed-loop schemes were visited for possible control solutions. Both control schemes work on the theory that if a secondary flow is pulsed at the frequency, amplitude, and opposite phase, 180 out of phase, of the primary flow’s vortex-excited state, interference occurs reducing the acoustic disturbance to zero. The challenge for this system is controlling the phase of the secondary injection which is used for actuation.

8.1 Open-Loop Control

Active open-loop control was the first control method implemented to test how effective the secondary injection system was in eliminating the acoustic pressure disturbances. An open-loop controller is advantageous because of its simplicity, however, it doesn’t account for any changes in the system over time. The input variable is set to a value chosen on the basis of past knowledge of the system prior to the controllers implementation.

Figure 8.1: Open-loop control model.
The open-loop control scheme’s logic is shown in Fig. 8.1. The actuator’s input for this controller can be found in Chapter 5. The desired input voltage characterized in Chapter 4 shows that a voltage of 770 mVDC must be supplied to the Minark MM23201 motor controller to produce an analogous response to the response produced by the vortex-driven pressure disturbances.

![Figure 8.2: Dynamic pressure response using open-loop control.](image)

Following from Fig. 8.1 the desired input of 770 mV is supplied to the motor controller which powers the motor. The motor turns the intake valve via a gear reduction of 7.2 : 1 which produces an output pulsed flow frequency response of 93 Hz. The 93 Hz pulsed flow is then injected into the system, the cold-flow simulation chamber.

Figure 8.2 shows the dynamic pressure trace for using the described open-loop control. Since the phasing is not controlled, the secondary injection can either add or subtract out the acoustic disturbance. In this case the FFT plot, shown in Fig. 8.3, shows that they
are adding. The result in Fig. 8.2 is an increase in pressure fluctuations at 93 Hz. The conclusion that is drawn from this result is that the secondary and primary acoustic signals are very close in phase. The resulting signal has an amplitude very close to the sum of the amplitudes of the two independent signals of the primary and secondary flows as seen in Fig. 8.3 showing 0.15 RMS amplitude for the FFT of the pressure signal. It is probable that the two systems become synchronized because of the coupling between the primary and the secondary flow systems. The facet where the secondary injection is hooked into the chamber has no way to prevent back flow from the primary injection. Because of this significant pressure back flow into the secondary injection system it eventually syncs with the primary flow.
The effectiveness of the open-loop control in the elimination of the pressure disturbances caused by vortex shedding was completely insufficient. The open-loop controller produced the opposite of the desired result, it added instead of subtracting. From this result the author’s conclude that a control scheme must be designed that will directly modify the phase of the secondary injection’s pulsed flow to allow for destructive interference and the eliminating of the pressure disturbances.

8.2 Open-Loop Control With PWM

A Pulse Width Modulated (PWM) signal was multiplied by the open-loop control scheme that was used previously as seen in Fig. 8.4. The theory behind this control scheme is to allow the ideal input value of 770 mV to be On for a certain amount of time and then the input value to be at a maximum, 1200 mV for a certain amount of time. Figure 8.5 shows the actual input voltage value that was used for this particular control scheme.

This is essentially speeding up the motor approximately every 0.0025 seconds. This is an open-loop response that blindly adjusts the motor for proper phasing of the secondary injections system.
Figure 8.5: Open-loop control with a PWM input signal which varies from 770 mV to 1200 mV.

Figure 8.6 shows the pressure response resulting from that application of the open-loop control with an additional PWM. Figure 8.7 shows the FFT response obtained from the pressure response seen in Fig. 8.6.

Using the open-loop control with an additional PWM signal proves to be effective. It does effect the phase of the secondary injection to the point of reducing the pressure signal to 0.035 RMS amplitude from 0.065 RMS amplitude.
Figure 8.6: Dynamic pressure response using open-loop control with a PWM signal which varies from 770 mV to 1200 mV.

Figure 8.7: Open-loop with PWM signal frequency spectrum of the system response, of Fig. 8.2.
8.3 Closed-Loop Control

A Proportional-Derivative (PD) controller was tested for control of the proper phasing of the secondary injection. Note that the setup does not allow direct changes to the phase of the secondary injection flow. This was overcome by indirectly changing such a phase by increasing the motor RPM for a very short time and then bringing the RPM to the desired 93 Hz value. This resulted in a Pulse Width Modulation (PWM) control action which, to the author’s knowledge, has never been implemented to control vortex-driven oscillations. A proportional controller allows for the feedback error to be modified by a proportionality constant, where the derivative controller modifies the difference in error over time. The proportional and derivative terms summed together makeup the new control input to the system.

\[
PD_{\text{cont}}(t) = k_p \ e(t) + k_d \frac{de(t)}{dt} \tag{8.1}
\]

where \( PD_{\text{cont}} \) is the Proportional-Derivative controller term, \( k_p \) is the proportional constant, \( k_d \) is the derivative constant, and \( e(t) \) is the error. The parameters \( k_p \) and \( k_d \) were tuned for optimal results.

Table 8.1 shows different proportional and derivative controllers that were tested. Figure 8.8 graphically verifies the data reported in Table 8.1. As the proportional gain is increased there is a greater reduction in the overall FFT amplitude at 93 Hz. It was found that the derivative term did little to effect the controllers effectiveness in reducing the pressure, therefore, a proportional controller gain was used to modify the feedback signal. It was
concluded that a $k_p$ equal to 9.0 and $k_d$ equal to 0.0 resulted in the optimal closed-loop controller without causing the input voltage to top out above 1250 mV.

A Labview Virtual Instrument (VI) was written to perform the required data acquisition and to compute the control input needed to control the phase of the secondary injection. Figure 8.9 shows how the control input for control was calculated.

Figure 8.8: Closed-loop frequency spectrum of the system response.

Figure 8.9: Closed-loop control model.
Table 8.1: Closed-loop frequency spectrum response and control gains.

<table>
<thead>
<tr>
<th>Amplitude (RMS) volts</th>
<th>Proportional Gain</th>
<th>Derivative Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.07549</td>
<td>1.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.05562</td>
<td>2.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.05199</td>
<td>3.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.04956</td>
<td>4.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.0509</td>
<td>5.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.04267</td>
<td>6.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.03852</td>
<td>7.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.05707</td>
<td>7.0</td>
<td>0.005</td>
</tr>
<tr>
<td>0.03545</td>
<td>8.0</td>
<td>0.0</td>
</tr>
<tr>
<td>0.03285</td>
<td>9.0</td>
<td>0.0</td>
</tr>
</tbody>
</table>

The error is calculated as the difference of the amplitude of the 93 Hz-component of the signal measured by the PCB pressure transducer from zero. Feedback from the system is measured by the PCB pressure transducer shown in Fig. 4.4. The error was calculated as

\[ e(t) = 0 - Amp(t) \]  \hspace{1cm} (8.2)

where \( Amp(t) \) is the amplitude of the FFT at 93 Hz. Ideally the error would reduce to zero.

This controller was unique in that it needed to change the speed of the motor from its set value to the corrected value due to an error for a particular amount of time. After that period of time was completed, it returned to its conditioned value. A PWM signal with a configured frequency of 200 Hz and a duty cycle of 50% was multiplied by the controller to accomplish this type of control

\[ PD_{PWM}(t) = PD_{cont}(t) \cdot S_{PWM}(t) \]  \hspace{1cm} (8.3)
where $PD_{PWM}(t)$ is the Proportional-Derivative controller modulated by the PWM signal $S_{PWM}(t)$. Figure 8.10 shows the closed-loop control input obtained when using this kind of control technique. The input signal shows that the signal does decrease towards the ideal input voltage value of 770 mV.

As shown in Fig. 3.8, the control input is sent as a voltage input to the motor controller. The motor turns at a RPM which is related to the input to the motor controller through the motor dynamics. This RPM will open and close the intake valve at a specific frequency. To produce a secondary frequency of 93 Hz, which is needed for acoustic instability cancellation in the duct, a motor input voltage of 770 mVDC is needed as previously discussed. The control voltage input is calculated as the difference between the desired voltage and PD-controller value modulated with the PWM signal. The PWM signal is used so that by
accelerating and slowing power to the motor, the secondary flow phase is effectively modified. Experiments utilizing the PD control scheme described above were performed.

When operated in a closed-loop mode, a clear reduction in the amplitude of the pressure oscillations at the dominant acoustic mode was observed. The use of a closed-loop control scheme was an improvement from the open-loop control scheme which utilized the PWM signal. This proves that control over how much the motor speeds-up or slows-down during its controlled On/Off PWM phase adjustment is important to the secondary injections phase control.

Figure 8.11 shows a dynamic pressure oscillation after the controller phasing was optimized. The peak-peak pressure oscillations are seen to be only 1.5% of the $P_c$. The reduction in RMS amplitude at 93 Hz is also seen in the FFT of Fig. 8.12. Figure 8.12 shows that the
dominant frequency, 93 Hz, was reduced to 0.032 RMS amplitude from the excited state of 0.065 RMS amplitude. This shows a 50% reduction in the pressure induced compared to the FFT in Fig. 4.4. Acoustic instabilities caused by the vortex shedding phenomenon in the duct took approximately 12 seconds for the maximum reduction to result. This result shows that the control method employed and the secondary injection that was used for actuation is feasible. Although the RMS amplitude at the dominant mode, 93 Hz, was significantly reduced, Fig. 8.11 and Fig. 8.12 show that other frequencies become excited, albeit at reduced magnitudes. Further experiments are suggested to explore this effect and to decrease the time response of the control scheme.
As a final result the cold flow simulation that was designed exhibited the vortex shedding phenomenon that was expected. The frequency of 93 Hz was ideal for implementing the designed secondary injection system. The secondary injection system alone was not sufficient.
to cancel the vortex shedding pressure oscillation. But with a feedback control scheme a satisfactory result was achieved.

As seen in Fig. 9.1 the open-loop control scheme by itself was not effective in reducing the pressure disturbance which instead amplified it because of the synchronization of the primary and secondary flows. The open-loop response did prove however that there is a certain need for phase control in the secondary injection system by highlighting the system’s strong tendency to want to add and not subtract responses and proved the need for a more stringent controller.

![Figure 9.2: Primary flow’s characterization signal compared with the open-loop with PWM control result.](image)

The additional of the PWM signal in the open-loop control scheme did prove to be effective as shown in Fig. 9.2 and Fig. 9.3. It effected the phase of the secondary injection which resulted in a decrease in the pressure inside the chamber. A second peak near 120 Hz
Figure 9.3: Frequency spectrum of the primary flow’s characterization signal compared with the open-loop with PWM control signal shown in Fig. 9.2.

This second peak is a result of the excitation of the secondary injection system when the controller is ran at 1200 mV.

As shown in Fig. 9.4 and Fig. 9.5 the overall success of disturbance reduction in the cold flow chamber was excellent. Nearly a 50% reduction in RMS amplitude was realized with the use of the active closed-loop controller. The closed-loop response also gives rise to an additional peak at approximately 110 Hz. This additional peak also arises from the applied controller. When the controller speeds up to higher RMS additional frequencies are excited by the secondary injection system. Therefore, as the voltage being applied decreases, the adding peak will continue decrease. If the controller value ever reached the ideal value of
Figure 9.4: Primary flow's characterization signal compared with the closed-loop control result.

Figure 9.5: Frequency spectrum of the primary flow’s characterization signal compared with the closed-loop control signal shown in Fig. 9.4.
770 mV then just the peak at 93 Hz would exist. Since the controller is turned On and Off at such a high rate the additional frequency is excited and remains.

## 9.1 Comments

When a closed-loop controller was implemented on the cold flow simulation system, a reduction in the 93 Hz signal was realized. While the 93 Hz mode was reduced others were excited. The excited modes aren’t related to the natural acoustic modes of the chamber so it isn’t thought that the stray excited frequencies are results of vortex shedding. The theory is that these excited states are coming from the inherent natural frequencies of the motor used for the secondary injection system. The PCB piezoelectric pressure transducers are extremely sensitive, therefore, it is possible that the excited states are noise from the motor itself.

It is also interesting that when the open-loop controller was implemented on the cold flow system the primary and secondary flow almost perfectly added. This could result from a large pressure difference at the chamber’s entrance, allowing over time for the secondary injection system’s flow intake to adjust to the primary flow’s pressure fluctuation.
9.2 Difficulties

Various difficulties occurred throughout the experimental process. Finding the correct spacing and exit diameter to give rise to the vortex shedding disturbance was difficult. While adjusting the spacing between the orifice plates was relatively easy, the distance that the plates could be moved was limited. Also, the spacing precision was limited as well. All the spacers were 0.125 inches thick, therefore, the spacing was adjusted in multiples of 0.125 inches.

The designed chamber was easier to excite at 300 Hz. A stronger signal could nearly always be found at 300 Hz. This made it difficult to obtain the 98 Hz frequency response that was desired due to the secondary injections limitations.

The flow from the secondary injection was also somewhat of a problem because it exhibited higher RMS amplitude when viewing the corresponding FFT response. The author solved this problem by inserting additions needle valves to inhibit the flow and decreasing the strength of the added perturbation.

9.3 Future Work

Several aspects of this project need to be examined more closely. A more in-depth analysis needs to be done on the vortex shedding itself.
Many new layers need to be added to the control software. A means of locking onto the frequency that is being excited in the system as well as the amplitude needs to be developed. In SRMs, this is particularly important because as the solid propellant burns away during usage, the diameter and chamber characteristic change and therefore different modes can be excited as time progress during the life of the SRM. Therefore, a means of recognizing these two major parameters is essential.

A more sophisticated controller would also aid in better control of this phenomenon. There has been some exploration in this area using adaptive control methods, however, this type of controller has never been applied to a secondary injection means of suppression.

A better means of actuation is important for allowing the system to work over a wider range of frequencies. A fast acting valve that can open and close at a range of 50 Hz to 500 Hz would be necessary for further investigation into this problem. Another hardware modification would be a way to minimize the flow so that the amplitude could be controlled easier. This system needs to be adaptive as well since it needs to be able to match the excited frequencies amplitude as close as possible in order to obtain cancellation.

There are various modifications to the experimental setup as well as the functionality of the software that would allow for more precise control. The secondary injection system should be redesigned to allow for the elimination of multiple frequencies.

The addition of a check valve could also be very useful. This would allow the decoupling of the secondary flow from the primary flow. Using a check valve would prove if the back...
flow from the primary flow actually has a profound effect on the secondary injections ability to do its job.

The software should be programmed to be sensitive to differences in frequency and amplitude. The experiment that was performed in this work dealt with a set frequency and amplitude of interest. However, for real-world applications, the software and hardware should be able to configure to any frequency and amplitude within a certain range, and be able to eliminate a frequency and amplitude that vary with time.

Additional work that needs to be done is to develop another means of calculating the error. This should be found for a more precise method of understanding the real time system dynamics. The time response for the controller should be increased for a higher level of precision in the controller itself. And lastly, hot-fire experiments should be performed so as to better match SRM conditions.
LIST OF REFERENCES


