AN EXPLORATION IN ATMOSPHERIC GAS BURNERS

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ABSTRACT

The objective of this paper is to explore the significant factors for primary aeration in atmospheric gas burners. Historically linear relationships have been used to estimate primary aeration with a design's geometry and features. With a given heating output a burners geometry is designed to meet safety, efficiency, reliability and customers' expectations. This paper explores the significant factors and how they interact with primary aeration. Experimentally exploring, port area, port loading, injector axial position, injection angle and the throat diameter interaction with port area. The structure creates an outline for atmospheric burner design.

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LIST OF ABBREVIATIONS

DOE, Design of Experimentation

ASGE, American Society of Gas Engineers

MIE, Minimum Ignition Energy

Btu, British Thermal Unit

FRD, Factor Relationship Diagram

LIST OF SYMBOLS

α, Thermal diffusivity

$\bar{\rho}$, Average density

A, Area of orifice opening (in^2)

A_f, The conical surface area

A_t, Tube cross-sectional area

B, Beta function

C, Correction factor

C_P, Specific heat capacity at constant volume

D, Specific gravity of gas

d_g, The gas density

d_q, Quenching diameter

D, Port diameter

h_c, High heating value

K, Discharge coefficient

L_f, Flame height/length

$\dot{m}_f$, Mass flow rate

N, Number of ports

P_f, Pressure difference across the injector ("WC)

PA_t, Air-Gas mixture at burner temperature

PA_r, Air-Gas mixture at room temperature

Q, Rate of flow (ft^3/hr.)

Q, Volumetric flow rate

Q_f, Volumetric flow rate of fuel

S, Stoichiometric ratio

S_L, Flame speed

S_L^0, Laminar burner velocity

$S_{L,0}$, Stoichiometric laminar flame speed

T_t, Burner temperature

T_r, Room temperature

T_f, Temperature of flame

T_∞, Ambient temperature

V_f, Volume flow

V_g, Orifice velocity

$V_t, Average\ flow\ velocity$

CHAPTER I

INTRODUCTION

In the commercial and gas appliance industry, burner design optimization has become crucial in the past 5 years. This need has come from appliance standard regulations and consumers. As emission standards, efficiencies standards and consumer expectations have risen the pool of acceptable burner designs has dropped significantly. The dangers of breathing carbon monoxide are now understood in greater detail. This is driving a greater expectation from public agencies and consumers.

Since commercial heating efficiencies and emissions standard are now higher, more is expected from product development to maximize burner performance. This is causing manufactures to consider new materials and new methods.

With the increase in efficiency standards condensing and modulating units are becoming more and more common. With the condensing unit the flue gas is cooled until the water vapor that is formed from combustion is condensed, allowing the unit to extract even more energy. The water vapor condensing can create corrosion in the unit and a requirement for less corrosive materials is now needed. With a modulating unit at high turn down, the temperature of the burner is elevated. The elevated temperature will cause deformation of a standard stainless-steel burner. A current solution that has been created is a medium for combustion, such as metal fiber or ceramics is a current solution.

For appliances, cooking speed, cooking control, efficiency and emissions have played a large role in the advancement of new burners. There are 3 burners typically used for an appliance, top burners, boil burner and bake burner. Top burners must perform and look appealing to a customer. The boil burner is the hardest to design. This is because the broil is at the top of the oven cavity and it has the least amount of access air. The bake burner is typically the highest input. The top burners typically slightly less than the bake. Historically top burners were cast iron or stamped steel. Currently, Die-Cast aluminum or brass is commonly used. Brass is considered more aesthetically pleasing as die-cast aluminum allows for flexible shapes and port designs.

Cooking speed, specifically time to boil water, has become an interest recently. The driver for this is the desire to reduce cook times. This can be accomplished by increasing the energy output of a burner. However, the customer desire of a high turn down ratio and agency testing restrict this as a solution. A gas valve designed around a specific burner can assist, as the flow curve is designed for that specific burner. Duel outlet valves are typically used so that retention flames can help keep the main flames stay stable with high turn down ratios.

Cooking control is an interest in foods that require finesse when cooking. An example is cooking chocolate or spaghetti. This requires an even distribution of heat and a low heating output. If too much heat is transferred to the pot these the food will burn. It is common to see a very thick cap on a burner that is designed for cooking chocolate. The thick cap reduces the efficiency of the burner by becoming a heat sink, reducing the energy transferred to the pot.

Emissions control is possibly the most important factor in both cooking and commercial applications. The goal is to have carbon monoxide emissions as close to zero as possible. This is because of health issues associated with carbon monoxide and the global impact from its emissions. There is also a desire to reduce NO_x because it is a greenhouse gas. This can be difficult because the formation of NO_x occurs when the flame temperature is at its highest, a desired state for efficiency.

Cost efficient units tend to emit more emissions. Part of this is funding for development and another part is fewer engineering solutions can be employed. In recent years lower cost gas appliance have become a concern in large cities, such as Chicago or New York. The concern comes from the realization that many small apartments get less air changes than what are recommended. Even a unit performing within the standards can raise concerns when a room receives low levels of air changes. Issues described above have nudged testing agencies to be diligent in their review of units.

Statement of the Problem

With emission standards stringency increasing, an atmospheric gas burner has become harder to get past agency testing. One factors, that will be explored in this paper, is maximizing primary aeration. Primary aeration is the amount of air injected before the flame front. A higher value will increase flame speed and flame temperature until the access air reaches about 1.1 to 1.3.

There are two classifications of burners that are typically used in commercial the appliance industry, full premixed and partially premixed. Full premixed relies on a blower to force air and gas into a burner. The equivalency ratio is around 1.3 for these types. Partially premixed relies on an orifice and a venturi to pull in primary air. Primary aeration for these burners typically has an excess air of .4 and .65.

Atmospheric burners do not have the ability to run at such high primary aeration percentages. They rely on a good design for primary air and the root of the flame to have good access to secondary air. The significant factors for primary air are critical in burner design.

Objective of the Study

The focus of study will be partially premixed burners also referred to as atmospheric burners. This thesis will explore possible factors that affect primary air in a burner. The study will explore 3 main features of a burner that are the foundation for acceptable performance with a given firing rate (Btu/Hr). These features are orifice position, port area and the throat area. They are thought to be the main factors that control primary aeration in a burner.

The study will look to confirm or deny the significant factors. The study will also look to confirm the range set for these factors in previous literature. The study will look to review the interactions between some of the factors.

Scope and Limitations of the Study

The scope of the study will only be atmospheric burners. Forced air burners will not be studied in this paper. This study will only review primary aeration. Secondary air will not be reviewed. To review secondary air the study would require a unit to review.

Some limitations that the study faces come when lab results are compared with computer models. The exact composition of species in natural gas varies. This makes computer modeling more difficult. For this study methane was used. Methane has a slightly lower specific gravity than natural gas.

Significance of Study

This studies significance comes from the need to understand the significant factors of burner design. When a given input is set for a design it is important to understand how to maximize performance.

How the burner pulls air into the burner is a critical factor for burner performance. If the burner is pulling too much air in, an air-shutter can choke off the air-supply. A unit's performance is a mix of primary air and how secondary air penetrates the flame front. Understanding the factors allows an understanding of how to balance primary air and secondary air at the flame front.

CHAPTER II

LITERATURE REVIEW

The focus of this study is an atmospheric gas burner. An atmospheric gas burner uses an orifice to inject gas. The gas supply, if natural gas, is typically delivered to the orifice at a range of 2.5 to 7 inches of water column. It is impossible for the supply pressure, for a typical atmospheric gas burner, to supply an adequate amount of air to the burner for it to reach stoichiometry [20]. Because of this secondary air is required for complete combustion. The burner is reliant on the orifice to inject primary air from the burner's surroundings.

Primary air, in industry, is known as the combustible air that enters the burner upstream from the burner ports [20]. For an atmospheric burner using town gas with a typical orifice pressure the primary aeration is typically from 40% to 60% of stoichiometric [27]. For many atmospheric burners using a gas jet primary aeration is 50% to 70% for natural gas [27]. For stove top burners using natural gas 50 to 60% primary aeration is typical [5]. There are many features of a burner and its components that control primary aeration. The orifice geometry, the burner tube diameter, the venturi geometry, the port loading, and interactions between these factors. For water heating applications, higher an efficiency is required than cooking. Some of these burners require a larger amount of primary air and almost always require a fan. For the ones that do not have a fan a require at the least 60 to 85% primary air [5].

It is important that the source of primary air is not contaminated with exhaust gas that has been recirculated for the burner or another burner on a unit. Broil burners in an oven cavity often have this problem. Some units will have the burner bent where the inlet is near the bottom of the oven cavity searching for a clean source of air. While some will have the broil burner sticking out of the back of the unit to find clean air. For air from the bottom top burners, burners on the top of the oven that pull air from below the sheet metal, recirculation can occur if the oven cavity allows spillage into the top burner cavity. Not all top burners pull air from below the sheet metal. Top burners that pull air from above the sheet metal, typically called air from top, provide a solution for spillage but typically suffer from transport losses in trying to inject primary air.

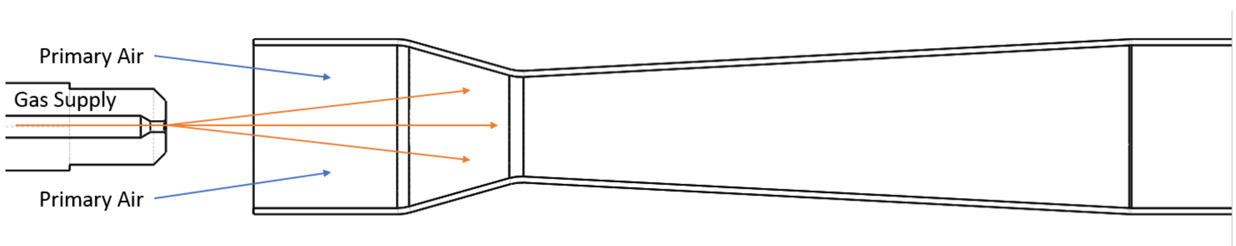

Figure 2.1 Primary Aeration and Gas Supply at Burner Inlet

Secondary air is known as the combustible air that enters downstream from the burner ports [20]. Secondary air heavily reliant on the geometry of the unit or burner box and how it restricts or allows air flow. Too much secondary air and heat will be lost due to the heating of the access air. Too little secondary air and heat will be lost due to incomplete combustion. An ideal state, for a flame front to give off maximum heat, is to have total air from 120 to 130% of stoichiometric [20]. This range can be seen in other texts as well. Handbook of Industrial Gas Utilization shows the best range for maximum output to be from 110 to 125%. The reason these values are greater than stoichiometric is because it is impossible to have perfect mixing.

Figure 2.2 Primary and Secondary Aeration Visual Explanation

Top burners are significantly more difficult to design than other types of burners. One reason comes from all the different geometries of pots that are typically used in households. The pot not only directly affects the air flow to the root of the flame, but the pot is also the heat exchanger to the food. A bake burner is slightly less complicated as the secondary air is a little more predictable when it is in use. The slots in the unit that allow for flow around the bake burner are geometrically controlled, pulling air from below the unit. A broil burner can prove to difficult, this is because the flow field is very complex and dirty combustion products can be found at the top of the oven cavity with the secondary air that is required for complete combustion.

Burner Injection

The orifice controls the flow rate of gas to the burner at a given pressure. The pressure is regulated to the orifice upstream generally from a regulator. It is important to measure the pressure as close to the orifice as possible in order to estimate the flow rate. As the gas passes through bends and fittings a pressure drop is expected.

In an atmospheric burner the orifice is one of the main components whose geometry directly affects primary aeration. The main affects are the approach angle, main hole diameter, orifice channel and a counter bore. Orifices are rated by something called the discharge coefficient. Most orifices have a discharge coefficient between .8 and .85 [29]. For orifices with an approach angle of 90° the coefficient of discharge is .65 [20]. The channel length has been determined experimentally. At 2”WC the maximum discharge rate can be found with a channel of around .11 inches in length. While at 4”WC it lowers to .1 inches in length, and 6’’WC is reduces to .095 inches in length [3].

To estimate the velocity of gas from the injector at low pressure with a low Reynolds number the following can be used [4].

$$V_g = \left(\frac{2P_f}{C d_g}\right)^{1/2} \tag{2.1}$$

Where:

P_f= Pressure difference across the injector (“WC)

d_g= The gas density

V_g= The orifice velocity

C= Correction factor

Where C is given as $C = 1 + {}^{4}/_{Re}$ within the range of 10<Re<400 [Briggs].

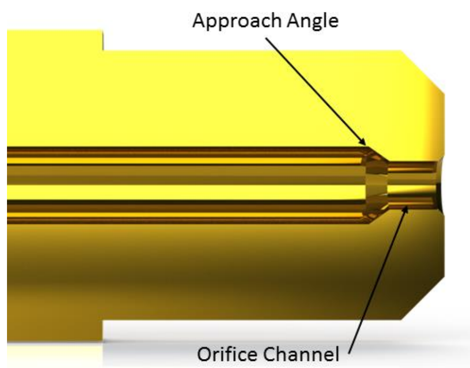

Figure 2.3 Orifice Internal Geometry

There is a relationship for a gas orifice discharge that is accepted by almost all burner texts. This equation can be found in the "Gas Engineers' Handbook". The discharge rate of gas flow from an orifice can be defined as:

$$Q = AV_g = 1658.5\ A\ K\sqrt{\frac{P_f}{d}} \tag{2.2}$$

Where Q = Rate of Flow (ft^3/hr.)

A = Area of orifice opening (in^2)

K = Discharge Coefficient

d = Specific gravity of gas

The momentum of the gas stream from an orifice is responsible for creating the pull of primary air. The greater the velocity of the gas stream the higher the momentum. To a certain point a greater velocity means greater the pull of primary air. There is a point where an increase in velocity will not increase primary aeration. The orifice with the highest overall exit velocity has a sharp edge channel [3]. There needs to be some channel at the exit of the orifice to develop the fluid flow field. A developed flow field is necessary for burner performance for a consistent injection of primary air. As the length of the channel increases so does the decrease in primary aeration, because of a reduction in velocity from friction. The position of the orifice from the burner throat also is a significant factor for the pull of primary air. It is recommended the axial distance from the tip of the orifice to the venturi throat is 2 to 3 throat diameters [27].

The momentum of the gas stream is then converted into the momentum of the mixture. The momentum of the mixture is always less than the momentum of the gas stream due to friction and other transport losses. The amount of momentum that is converted depends on the components of the burner. The components and features of a burner that are considered a significant factor for the momentum of the mixture are, the air-shutter, tube diameter, throat diameter, venturi geometry and port area.

Burner Introduction

When burners are designed for an appliance, the first design constraint that is generally established is the firing rate. To maximize primary aeration, it is important to have the correct port area for the desired firing rate. For a steel atmospheric tube burner with punched circular ports, it is recommended by ASGE that the port loading is between 20,000 and 25,000 BTUs/in^2-hour. This value is the power per area.

For a simple burner with circular ports the port area can be calculated. The letter N is used to represent the number of ports. The letter D is the diameter of the port.

$$A_{Total} = \frac{N * pi * D^2}{4} \tag{2.3}$$

Typically, the power of the burner is calculated using a volumetric flow rate meter, then assuming a given calorific value or by using a calorimeter. The calorific value is given as the energy per unit volume.

$$\dot{Q} = Heat\ Power = V_f * Cal \tag{2.4}$$

The power can also be given in the following format. This is using the mass flow rate and the high heating value [7].

$$\dot{Q} = Heating\ Power = \dot{m}_f * \Delta h_c \tag{2.5}$$

Once the heating power of the burner is known and the port area is defined, to get the volumetric flow rate at the ports the primary aeration must be found. This is typically done by tapping into the inside of a burner and using an oxygen sensor to determine the primary aeration. If the value is given in percent oxygen, the following can be used. It should be noted, Air% and O_2% are not the same.

$$Fuel\% = 20.95 - O_2\% + \left(\frac{78.09}{20.95}\right) * (20.95 - O_2\%) \tag{2.6}$$

$$Primary\ Aeration = \left(\frac{\frac{Air\%}{Fuel\%}}{\frac{Air\%_{Stoichiometric}}{Fuel\%_{Stoichiometric}}}\right) * 100 \tag{2.7}$$

The volumetric flow rate at the ports can be found with the following (Couto 3). It should be noted, assuming a constant volumetric flow rate from port to port is not accurate but in most cases is acceptable. To review the true volumetric flow rate at each port, the burners pressure curve must be reviewed.

$$\dot{m}_{air,premixed} = \eta * Air\ to\ Fuel_{Stoichiometric} * \dot{m}_f \tag{2.8}$$

$$Q_{Total} = \frac{\dot{m}_{air,premixed} + \dot{m}_f}{\bar{\rho}} \tag{2.9}$$

$$Q_{port,average} = \frac{Q_{Total}}{N} \tag{2.10}$$

If the loading is too low the port mixture exit velocity will slow. This will make the flame front move closer to the burner's ports. Because of the reduction of the volume flow rate out of the port and the change is flame front geometry the temperature of the steel burner will increase. This will create access wear on the burner and in extreme cases can allow for flash back. Flash back is when the flame front moves through the burner ports and into the burner.

If the loading is too high the port mixture exit velocity will be high and will increase pressure which will reduce primary aeration. The reduction in primary air will eventually cause yellow tipping and the formation of soot. When the exit velocity is increased, because the port loading is too high, it is common for the flame to lift from the rim of the burner. This is because the exit velocity becomes greater than the flame speed.

The tube diameter is a significant feature for primary aeration. As the desired input of a burner is increased the need for a larger tube will be apparent. If the tube diameter is too small the burners internal pressure will be increased and will reduce the momentum of the mixture. The momentum reduction will reduce the pull of primary air. It is recommended that the burner tube cross-sectional area should not be less than 1.5 of the total burner port area [27].

Not all recommendations are as conservative, Walters recommends the tube cross-sectional area to be greater than 1.2 of the total burner port area. If the burner tube diameter is too large the slope of the venturi will be too great. This will increase momentum losses due to eddy currents in the mixture stream. The recommended burner tube diameter should be between 1.25 and 1.55 throat diameters [27]. The length of the burner should be at least 6 times the inner diameter of the burner mouth [22].

A feature where the recommended dimensions from the literature varies the most is the venturi throat cross-sectional area to the total port area. However, the literature does agree that the throat area should be less than the total port area. It is noted that the burner should never have a throat area less than 40% of the total burner port area [5]. That said, one claim is that the optimum throat area should be 43 percent of the total burner port area [3]. A similar optimum ratio of 45% is also given [29]. Many authors choose to give a range. One recommendation is to keep the range from 45 to 65% [27]. A slightly higher range was given from experimental data at 60 to 70% [1]. A less conservative recommendation, that might not be best to test the limits with, of 22 to 91% could also be a solution [18]. Ranges are typically given for this feature because of tooling constraints. Companies have developed ways to make porting flexible. This allows for a large assortment of total port area at a low cost. However, the venturi is typically formed in a die. Creating a die for each burner design is typically not practical.

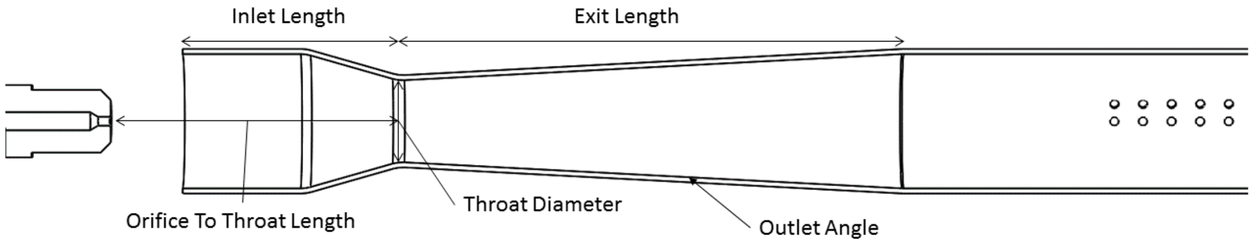

Figure 2.4 Venturi Geometry

The inlet and exit lengths of the burner also affects primary aeration. The main restriction in industry is the space that a burner can occupy. The inlet lengths are recommended to be at 2.25 to 2.5 venturi throat diameter and the exit lengths are recommended to be from 4 to 8.5 throat diameters [27]. One claim is made that the inlet length should be 2 times the throat diameter and the exit length should be 6 times the throat diameter [5]. It is recommended that the outlet angle of the venturi is about 2° [3]. If the outlet angle approaches 3.5° transport losses can be seen [18].

The orifice position relative to the throat diameter is also a significant factor and influences primary aeration. Not only in axial direction of the burner but also in an angular direction. This would be seen if the orifices gas steam was not pointed down the center of the venturi. This is referred to as misalignment. The orifice to throat length is not considered to be critical but it is recommended to be between 2 to 3 throat diameters [27]. It has been shown the reduction in primary aeration is small from .7 to 3.3 throat diameters [18]. Typically, the orifice position can easily be set in a manufacturing setting. That makes the axial distance easy to control. The misalignment of the orifice can be harder to avoid in a manufacturing setting. A single degree of misalignment is said to reduce primary aeration by 2.1% [21]. Misalignment could be seen when an orifice is cross-threaded, fixturing is misaligned or the orifice channel was drilled at an angle.

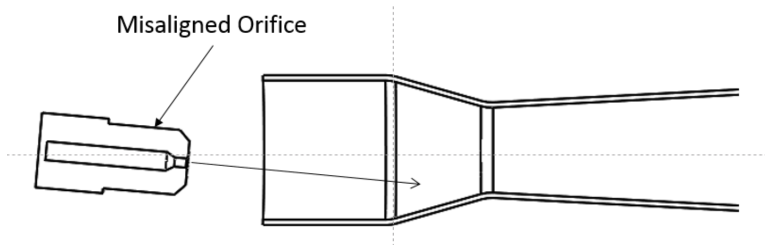

Figure 2.5 Orifice Misalignment from Centerline of Burner

Orifice displacement will also cause a reduction in primary aeration. In an appliance this is typically caused by two mounting surfaces being misaligned. If the misalignment occurs the reduction of primary aeration loss is estimated to be 3.8% loss for every millimeter will be seen [21].

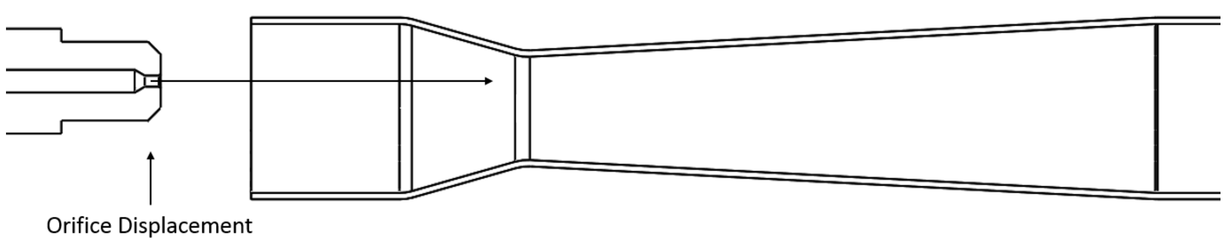

Figure 2.6 Orifice Displacement from Center of Burner

The source for primary aeration is often controlled by an air shutter. An air shutter is designed to control the amount of primary air that is allowed into a burner. Because propane has more carbon than natural gas per molecule it requires more air to be entrained for a given rate. It is recommended that the inlet for primary air is greater than twice the area of the ports [27]. This recommendation assumes flame lift is not an issue. For a well-designed burner the primary air opening should be bounded with a maximum area. A recommendation for the primary air opening, that bounds an upper and lower limit, is given from 1.25 to 2.25 the total port area [19]. Another way to look at an ideal air-shutter position is to view the velocity of the air as it is entering. From the Department of Commerce experiment, to maximize primary aeration, the inlet velocity of the primary air should not exceed 4 to 5 feet per second [3]. However, this is also a recommendation that is not insuring flame lift will not be an issue.

After considering the primary aeration of a burner it is important to consider secondary air and how the two affect the flame. A flame that has the proper amount of air and fuel in the flame front should have a visible primary and secondary cone. If the flame is receiving too much air the flame will lift from the burner port. If the flame has too little air the cones will become very long, and a soft flame might form or even a sheet flame. The primary cone outline represents a region where primary air creates a mixture that is stoichiometric, sometimes referred to as the premix flame front [12].

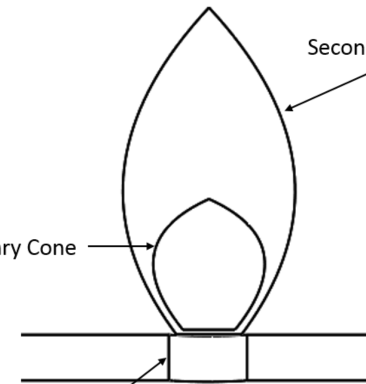

Figure 2.7 Flame Features

If the port loading is higher the exit velocity will be higher as well. This will keep the burner cooler than a burner with a low exit velocity. When a burner has a low exit velocity the flame can sit on the burner creating excess heat. The hot burner will preheat the gas and affect primary aeration. An estimation of the change of primary aeration as the burner heats up can be given by the following [29].

$$PA_t = PA_r\left(\frac{2T_r}{T_t + T_r}\right)^{.5} \tag{2.11}$$

Where PA_t, PA_r, T_t, T_r are defined as air-gas mixture at burner temperature, air-gas mixture at room temperature, burner temperature, burner temperature at room temperature, respectively of their order.

It should be noted that the above theoretical equations are assuming the flames are not coalescing. This is when the secondary combustion zones combine to form one long flame. When a flame coalesces the amount of oxygen available is reduce. With atmospheric gas burner, downstream from the flame front the combustion process relies on the carbon monoxide to oxidize with secondary air. For slotted ports it is recommended the slot width is kept no greater than .045'' to avoid a coalescing flame [5].

Burner Flame

The flame speed is sometime referred to as the burner rate. The flame speed is affected by multiple factors. The type of gas, the equivalence ratio of the mixture, the temperature of the flame front and the direction of the burn are all factors that affect the flame speed. The shape of the duct affects the flame speed for ducts that are narrower than 7 millimeters [14].

The laminar flame speed of a premixed gas is given as the velocity of the unburnt gas just before the gas enters the flame front. This is given as,

$$S_L = V_t \frac{A_t}{A_f} \tag{2.12}$$

Where,

A_f= The conical surface area

$V_t = Average\ flow\ velocity$

$A_t = Tube\ cross\ sectional\ area$

The flame temperature of many gases reaches their highest value with an equivalence ratio between 1 and 1.1 [13]. There are two chemical reaction zones in a flame front. The first is called Zone I, it is referred to as the Preheat Zone. The second is Zone II, it is referred to as the Reaction Zone [12].

When the burners conditions have reached steady state the flame front will stabilize. The burner temperature is significant in flame stabilization. Heat flows from the flame front to the burners wall at the port. The loss in energy from heat reduces the flame speed. The flame speed will be reduced from heat loss until the flame speed is equal to the local flow velocity [35]. This is of course assuming flash back does not occur.

If the flame speed is faster than the air-gas mixture velocity coming out of the burner port the flame front will move into the burner, creating what is called flash back. This is of course dangerous and will cause burner failure. There are a couple of ways to eliminate flash back and extinction pop. One way is quenching the flame with the port size. The rate of heat loss generally comes from the loss of heat to the burner port wall. If the reaction rate is lower than the heat loss rate, a flame will not be possible [30]. A relationship with the quenching diameter and the flammable mixture is given by the Peclet number. The relationship is created by connecting the convective transport and the heat transport. The critical Peclet number is used in combustion to characterize the flammability limits [24]. The relationship is given,

$$Pe = \frac{Convection\ Transport}{Heat\ Transport} = \frac{S_L^0 d_q}{\alpha} \tag{2.13}$$

Where,

S_L^0= Laminar Burner Velocity

d_q= Quenching Diameter

α= Thermal Diffusivity

With a fully premixed burner a fan is used. Typically, the fully premixed system is design so that the mixture velocity coming out of the port is very high. In an atmospheric burner the mixture exit velocity is not as fast. For these types burners the port must be designed to quench the flame. As the flame front comes closer to the burner, the burner takes more heat from the flame front. This reduces the flame speed. There is a distance from the burner, where the flame speed will equalize with the mixture exit velocity.

Burner Ports

If the mixture velocity is greater than the flame speed the flame front will push away from the burner. This will continue until the two-reach equilibrium or the flame is extinguished. A flame is said to lift when the root of the flame is no longer touching the rim of the port. A lifted flame will often have the appearance of floating. A flame that is lifted is considered unstable. The concern is that the flame will extinguish, allowing unburnt gas to flow into the combustion chamber.

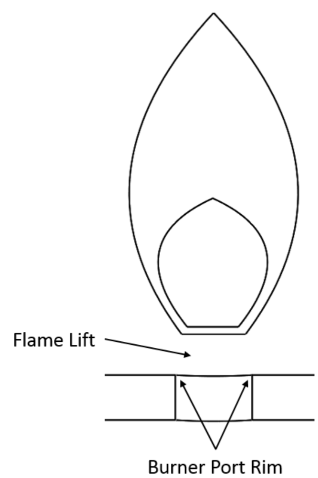

Figure 2.8 Flame Lift Example

A common practice is to use smaller retention ports to avoid lifting. This technique can be seen on top burners when walking through the local appliance store. Unlike other applications top burners work with a gas valve where the input is controlled by the consumer. A typical turn down ratio for a top burner is 10:1. However, due to testing such as the chocolate test there is a push to increase the turn down that a top burner can perform at. With such a large operating range the geometry of the burner cannot be designed to be optimum for the entire span of the gas valve. To help keep the flame from lifting when the gas valve is at a position when the burner is lean retention ports are very helpful. It is recommended the retention ports are 20% of the total flow and that they are no more than 6.5 mm from the main ports [27].

Port depth is another characteristic that influences air that is pulled into the burner. A deep port will resist fluid flow. The transport losses will decrease primary aeration upstream. As port depth decrease from 1.5 inches to 1/16 of an inch primary aeration will increase, reaching a maximum at 1/16 of an inch [28].

A well-designed port pattern can create an environment that can give the burner the best chance for complete combustion of the gas air mixture as it leaves the burner ports. Ports that are too far apart might not carry the flame down the burner. This will allow for unburnt gas to escape into the combustion chamber. To help insure all ports have a flame, it is typical for a flame sense rod to be on the other side of the burner from the ignitor. This will help insure the flame completely carried around the burner. If there is a visible small gap between ports where flame does not exist, this is referred to a hard flame [20]. Ports that are too close together will coalesce. Flames the coalesce typically soot. Because the flame cones come together, it becomes hard for oxygen to penetrate the flame front.

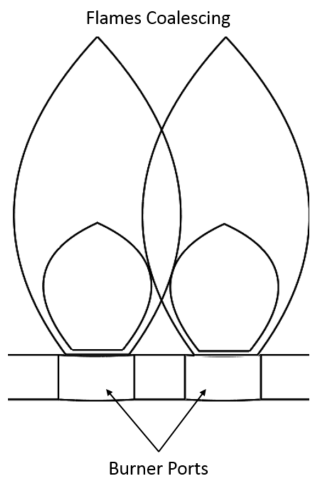

Figure 2.9 Flame Coalescing

There are many styles of burner ports in industry. Circular punched ports, lanced ports and slotted punched ports are the most common. The lanced port gives the mixture a direction as it leaves the burner. Typically, lanced ports do not have as high port loading as circular ports. Port spacing is less trivial than the other features of a burner. This is because the amount of secondary air required is dependent on the primary air. Also, the flame characteristics such as size are dependent. Even the size of the port plays an important role. For most domestic applications keeping the port spacing, center to center, less than 9/32’’ is a good idea [29]. When the port spacing from edge to edge is at .25’’, the burner is less likely to lift and less likely to yellow tip. As the spacing approaches .125’’, the burner will require 3% more primary aeration to eliminate yellow tipping [18].

Each style of port will have a different velocity profile, the most common in industry are circular ports, lanced ports and square ports. The main factors that control how the mixture exits the ports are the port area, gas rate per port area, primary aeration and the burner temperature. For a circular port, the flame height can be estimated with the below [7] [31].

$$L_{f,experimental} = 1330 \frac{Q_f * (\frac{T_\infty}{T_f})}{\ln(1+\frac{1}{S})} \tag{2.14}$$

To estimate the flame height from a theoretical approach the following can be used, if the burner is well ventilated with excess oxygen [31]. For a circular port the following can be given.

$$L_{f,theorectical} = \frac{Q_f \left({T_\infty}/{T_F}\right)}{4\pi D_\infty \ln\left(1+{1}/{S}\right)} \left(\frac{T_\infty}{T_f}\right)^{.67} \tag{2.15}$$

Circular ports are often used in combination with square ports or even rectangular ports for flame stability. Because the discharge of the gas from different style ports are different this makes a flame front that is less susceptible to resonate frequencies causing flame lift or even flame disruption. The flame height of a square port can be estimated with the below [7] [31].

$$L_{f,experimental} = 1045 \frac{Q_f \left({T_\infty}/{T_f}\right)}{[inverf[1+S]^{-.5}]^2} \tag{2.16}$$

$$L_{f,theorectical} = \frac{Q_f \left(\frac{T_\infty}{T_f}\right)}{16 D_\infty [inverf(1+S)^{-.5}]^2} \left(\frac{T_\infty}{T_f}\right)^{.67} \tag{2.17}$$

Slotted punched ports are different than lanced ports. Lanced ports pierce the material and give the gas direction when exiting the ports. A slotted punched port is simply a square cut-out in the material. The flame height for a square punched port can be estimated with the below [7] [31].

$$L_{f,experimental} = 2 * 10^3 \left(\frac{\beta^4 Q_f^4 T_\infty^4}{a h^4 T_f^4}\right)^{1/3} \tag{2.18}$$

$$L_{f,experimental} = 2 * 10^3 \left(\frac{\beta^4 Q_f^4 T_\infty^4}{ah^4 T_f^4}\right)^{1/3} \tag{2.19}$$

$$L_{f,theorectical} = \left[\frac{9\beta^4 Q_f^4 T_\infty^4}{8D_\infty^4 ah^4 T_f^4}\right]^{1/3} \left(\frac{T_f}{T_\infty}\right)^{2/9} \tag{2.20}$$

Where L_f = Flame Height

Q_f = Volumetric Flow Rate of Fuel

T_∞ = Ambient Temperature

T_f = Fuel Temperature

S = Stochiometric Ratio

β = Beta Function

Aluminum die cast burner design has seen more focus in burner design. Die casting production allows the parts to be produced in all types of shapes and sizes. Aluminum die cast burners are typically found on the top of a stove top. A common design across industry is to slant the ports. This sends the flame upward towards the cooking pot and towards a secondary air source. Some more radical design is known as swirl ports. This is where the ports are curved. A swirl burner extends the residence time when the flame and exhaust gas is below the pot [16]. A swirl burner has been attempted in production and in a laboratory setting. The swirl burner tends to see an increase in CO. It also has a higher maintenance cost. Because of the swirled channels it is hard to taper the channels. When the die pulls apart, a part without a taper see's more wear and a shorter tool life.

Ignition

Once the burner is designed an ignition sources should be considered. Two commonly used ignition sources in modern appliances are a hot surface ignitor and a direct spark ignitor. The hot surface ignitor heats up to around 2,500 Fahrenheit. It creates ignition by heating the mixture up above the gas's ignition temperature. Direct spark is when an ignitor creates a spark in the mixture causing ignition. An advantage to the hot surface ignitor is the contact area with the gas mixture stream. A disadvantage is the hot surface ignitor costing more than direct spark. Standing pilots were very common in the past. They are now considered an older technology and are being used less and less.

Minimum ignition energy is the lowest amount of energy required to ignite the fuel-air mixture. From Glassman the Minimum ignition energy is given as

$$MIE \sim \left({}^{D_{th}}\!/_{S_{L,0}} \right)^3 \rho C_p (T_{ab} - T_0) \tag{2.21}$$

Ignition is begun when current between two electrodes create a plasma channel. In this phase the flame kernel growth has started. After the plasma channel is created the arc phase begins. From the arc phase the glow phase follows. In the glow phase there is a significant energy loss to the electrode itself. Ignition is often described in terms of probability for ignition at a given location [2].

One of the most referenced experiments for ignition was conducted by Lewis and Von Elbe. The minimum ignition energies from the Lewis and Von Elbe experiment are to an ignition probability of .01 [2]. However, this claim is in question because the experiment data and number of experiments ran by Lewis and Von Elbe is unknown. Their experimentation showed for methane at 1 atm and 25° C, ignition can occur when there is between 60 to 130% primary aeration. The minimum energy occurs, around 85% primary aeration, close to .3 mJ. As the aeration increases or decreases the minimum spark ignition energy increases. Their experiments also reviewed spark gap. They found that shape and size of the electrode do not affect the minimum ignition energy when the spark gap is greater than the quench distance. At the stoichiometric mixture of methane and air the quench distance is around .08 inches. Greater than .1 inches the energy required is increased until around .2 inches. With a spark gap greater than .2 the amount of energy required for ignition is not practical for an appliance.

Hot surface ignition relies on the hot surface ignitor to heat the air-fuel mixture until ignition occurs. The lowest temperature at which methane will ignite is found when the equivalency ratio is at .7 which is a leaner mixture [32].

CHAPTER III

METHODOLOGY LABORATORY EXPERIMENTATION

Introduction

To properly review a burner a couple of tools are very useful. A gas analyzer is a must, for this experiment a SERVOMEX MiniMP 5200 was used. This machine can take oxygen readings in percent of gas at room temperature. Natural gas can be supplied to consumers in the United States at 1,035 ±50 Btu per standard cubic foot at 1 Atm. and 60 °F [33]. To understand the gas better a Cutler Hammer Controls calorimeter was used in this study as well. A calorimeter is used to measure the heat from the combustion process. The calorimeter used in the study displayed the calorific value in British Thermal Units per standard cubic foot.

Figure 3.1 Oxygen Meter and Calorimeter

To capture the volumetric flowrate an American Meter Dry Test Meter was used. A stop watch was used to record the time it took to complete a revolution. A single revolution represented one tenth of a cubic foot. These two meters the calorimeter and the dry meter are used together. With the calorific value and the volumetric discharge of gas it is possible to calculate the port loading.

Figure 3.2 Dry Meter

A hole was drilled into the burner where a hose barb was threaded in the hole. The position was halfway down the burner, this will allow time for mixing of air and fuel. The machine was set to measure oxygen. It is important to note that the samples were taken from within the burner, this is to determine the primary aeration.

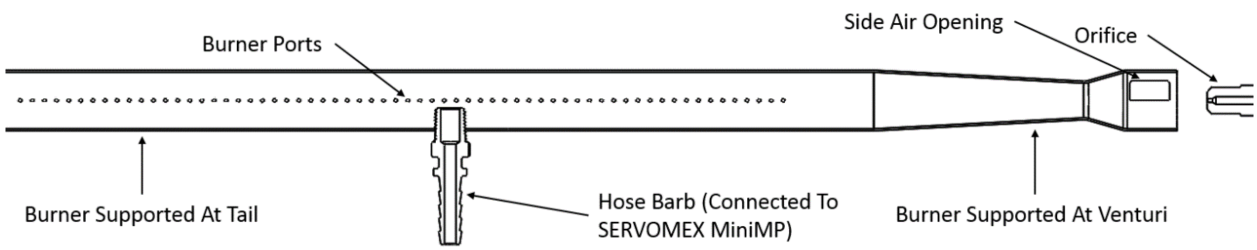

Figure 3.3 Experiment Testing Oxygen Tapping

The method above was used to pull the mixture from inside the burner. This was done instead of pulling from the flue gas. The flue gas would also have included secondary air. The center of the burner was chosen. This location was far enough from the exit of the venturi where the mixture has had time to mix. It is also not at the end of the burner, depending on the burner length and port there is sometimes a pressure spike at the end of the burner.

Another tool that is important when analyzing a burner is a calorimeter. The laboratory where the experiments were ran had one and the values were able to be recorded. The last is a volumetric flow meter or a mass flow meter. Most labs have dry meter where a dial is used, for these a stop watch is typically used to take the rate.

With these tools, analyzing the design of the burner and how well the burner entrains primary air is possible. It is also possible to review how deviating from a specified design can create losses in primary aeration.

For the experiment a replacement burner for a grill was ordered online. The burner had circular punched ports, a typical venturi, and had an air opening that was restricted by a piece of aluminum tape. The burner had a hole drilled where a hose barb was placed so that the primary air could be analyzed.

The burner was positioned with two magnetic v-blocks. The air-shutter was removed, and tape was placed covering the lower half of the air-opening on both sides. A red hose was run from the burner to the SERVOMEX.

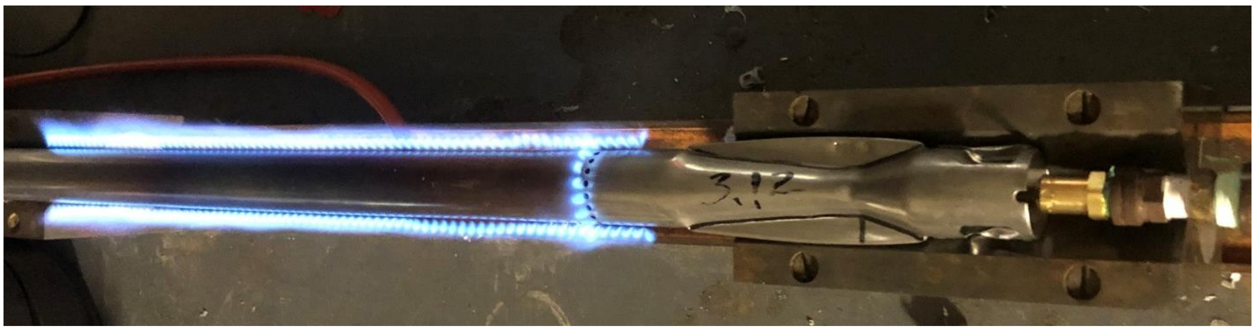

Figure 3.4 Fired Burner on Test Bench

For all lab experiments the burner was fired on a test bench as seen above. The test bench allowed the burners alignment to be controlled. The burner was always fired into atmosphere in a temperature-controlled room that stayed around 72 (°F).

Lab Experiments

Lab Experiment 1 – Orifice Concentricity with Burner

To review how concentricity effects the primary aeration of the burner, an adjustable head was moved to move the floating orifice up and down inside of the burner. The movement was measured using a caliper connected to the rail.

Figure 3.5 Test Bench Height Adjustment

The orifice was moved down in the burner. Two adjustments were made from zero. The picture below shows how the orifice is free floating. It also shows how the air-shutter has been removed.

Figure 3.6 Orifice Adjustment for Off-Center

Study Population – Orifice Concentricity with Burner

Three positions were observed in the study. The first was a baseline at zero. From the baseline an off-center of .125 inches and .25 inches was observed. The orifice was drilled to a diameter of 1.92 millimeters.

Lab Experiment 2 – Orifice Axial Depth

To review primary aeration change depending on the orifices axial depth in the burner throat the burner was fired in open atmosphere. Because the focus of the experiment was to review primary aeration and how it changes with the axial depth of the orifice, the burner was not fired inside of a unit.

Pressure at the orifice was measured with a digital manometer. The pressure was pulled just up-stream from the orifice. The height of the burner was centered in the burner by the adjustable table, as seen below.

Figure 3.7 Orifice Depth Origin

The picture below shows how the orifice was moved towards and away from the burner. The position pictured is the negative 0.25-inch position. Negative was chosen because the typical position for the orifice is inside of the burner.

Figure 3.8 Orifice Depth Experiment

Study Population – Orifice Axial Depth

Seven positions were chosen for the orifice inside of the burner. The zero point was at the beginning of the burner. This point was defined as when the tip of the orifice is flush with the inlet of the burner. The negative positions were .75, .5 and .25 inches. The positions inside of the burner or positive positions were .25, .5 and .75 inches. The orifice diameter was drilled out to 1.60 millimeters. The day of the Orifice Axial Depth study, the Calorimeter showed the calorific value of the natural gas was 1,055 Btu per cubic foot. The volumetric flow rate at the meter was measured at 0.176 cubic feet per minute.

Lab Experiment 3 – D.O.E. – Port Area, Orifice Diameter and Orifice Depth

The factor relationship diagram from the design of experimentation can be seen below. There are 3 factors with the D.O.E. set-up as a full factorial, giving the study 7 degrees of freedom.

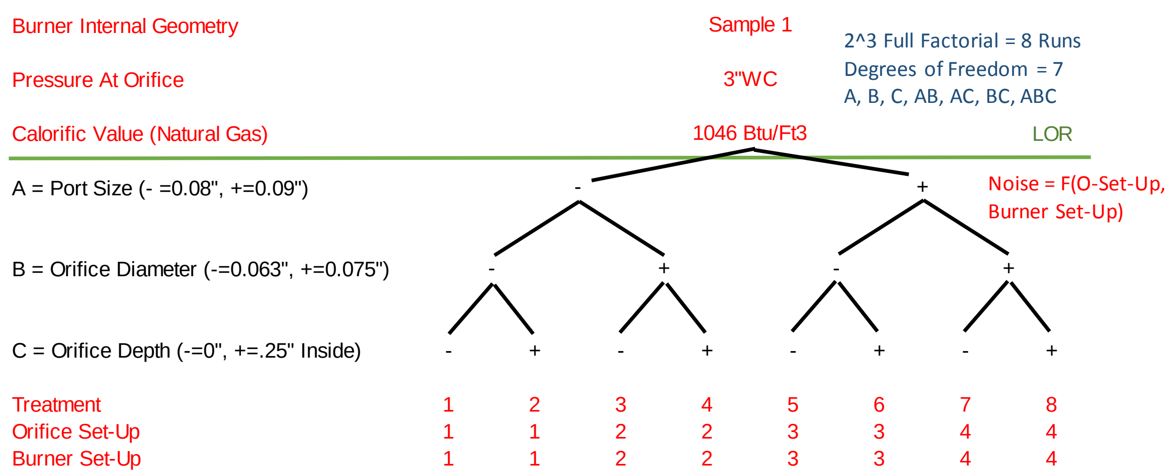

Figure 3.9 Factor Relationship Diagram for DOE, Lab Experiment 3

The orifice depth was changed in the same way as Lab Experiment 2. In this experiment the orifice was only moved into the burner. Position zero is displayed below.

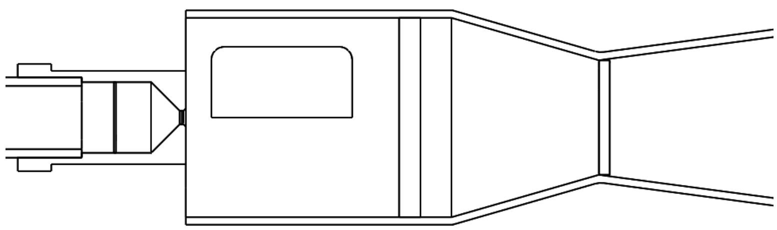

Figure 3.10 Orifice Depth, Position Zero

Study Population – Port Area, Orifice Diameter and Orifice Depth

The study used one burner, the burner ports were drilled out to increase the size from 0.08 to 0.09''. The study used two orifices, one with a diameter of 0.063'' and the other with .075''. The two locations for depth were the zero point and 0.25'' inside the burner. The locations used the tip of the orifice to align with the inlet end of the burner.

Computer Modeling Experimentation

Computer Modeling Fluid Flow – Orifice Axial Depth

The platform used for the computer modeling was SolidWorks. The simulation package used was the SolidWorks add-in Flow Simulation. The was set-up as an internal flow simulation. This was done by creating an air-tube surrounding the burner. At the surface of two air-tubes a Total Pressure Boundary Condition was established. The values were air at standard temperature and pressure.

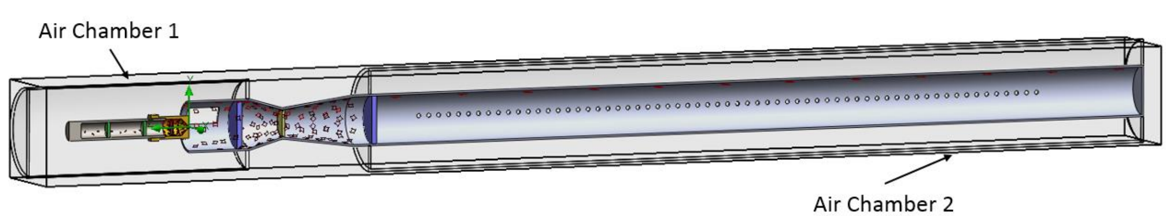

Figure 3.11 Flow Simulation Set-Up

The inlet boundary condition was also set as pressure, to replicate lab testing. The gas in the simulation was pure methane. The methane is mixing with air when leaving the orifice.

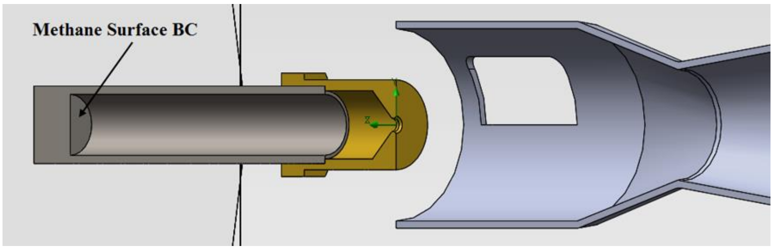

Figure 3.12 Flow Simulation Gas Inlet Boundary Condition

Top replicate transport frictional losses, surface goals were created at the material surfaces. The below shows the locations of the boundary conditions. The red and back squares represent where these are being applied.

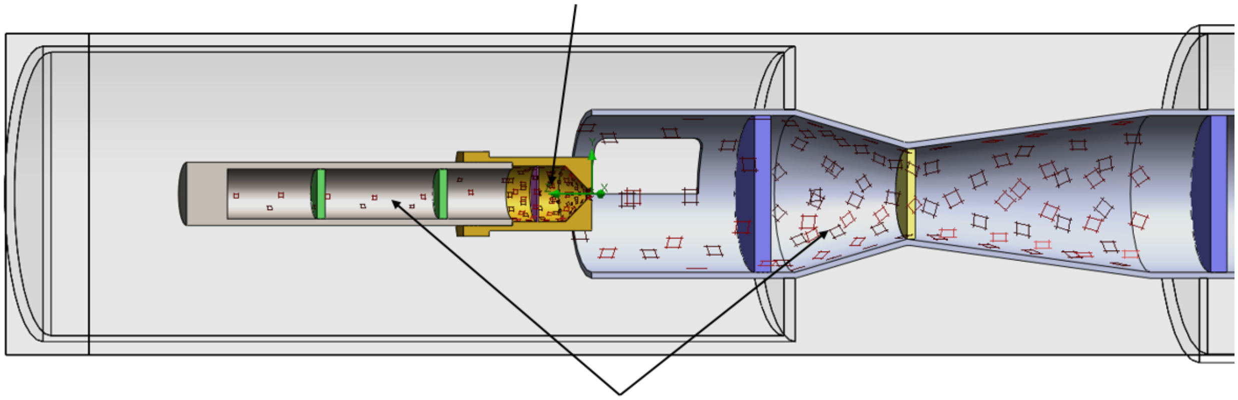

Figure 3.13 Inlet Boundary Conditions

Gates were used as surface goals. The first step was to make sure the cylinders were not involved in the calculation in component control. The next was to create surface goals at the faces of the sensors. The goals used, Total Pressure, Mass Flow Rate, Mass Fraction of Air, Mass Fraction of Methane, Average Volumetric Flow Rate of Air and Methane. These goals were used where applicable by the model.

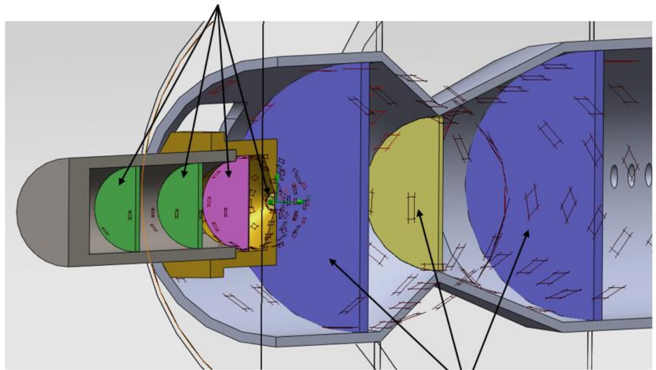

Figure 3.14 Inlet Boundary Goals Gates for Convergence

The last step to be noted is critical to replicate the results of the experiment. A first guess was used for the initial condition concentration of Methane and Air. The simulation was run multiple times, each time slightly tweaking the initial concentration condition until the solution converged. EdgarDavids#1

Study Population – Computer Modeling – Orifice Axial Depth

The inlet pressure for the pure methane was 3 inches of water column above standard pressure. The flow field was defined as fully-developed leaving the methane surface boundary condition.

Limitations of Study – Orifice Axial Depth

When comparing the laboratory study and the computer model there is an expected slight discrepancy. Because the gas supplied in the lab was not pure methane and varies day to day and given the equipment, it is not possible for the species in the simulation to match the species that make of the composition of natural gas in the lab.

There is also a slight deviation expected in the solid model with regards to roughness of the material as well as geometry. Geometrically, the ports diameter, total port area and venturi geometry are all expected to be slightly different. The orifice is also expected to be slightly different. The exact geometry of the orifice would require special equipment for measurement.

Computer Modeling Fluid Flow – Orifice Yaw Angle

The study was conducted with the same goals and sensors as the orifice depth experiment. The mesh quality was also the same. There was a difference in firing rate. The orifice was made slightly larger at 0.075''.

Study Population – Computer Modeling – Orifice Yaw Angle

The orifice yaw angle study was set up with the tip of the orifice 0.25'' inside of the burner. The orifice hole was slightly larger at 0.75''. There was an increase in flow rate from the orifice depth experiment. The orifice was started at 0°. The orifice was then inclined at 1° and the inclined at 2°. 3 experiments were run in total.

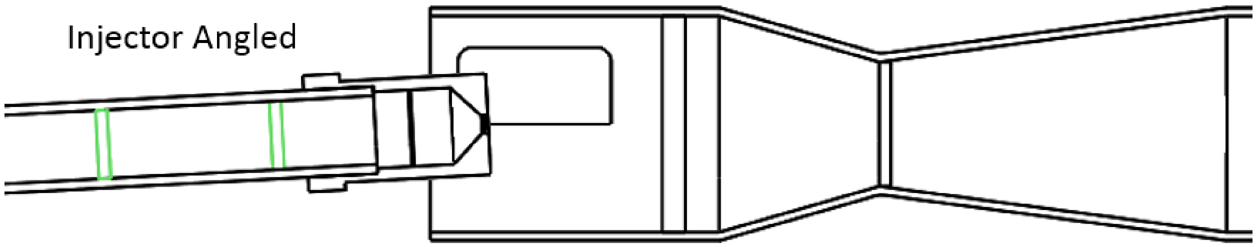

Figure 3.15 Injector Angle Experiment

Computer Modeling Fluid Flow – D.O.E. – Throat Diameter and Port Area

The experiment was conducted as a full factorial. Factor A was designated for port diameter and factor B was for throat diameter. The experiment was conducted with the same goals and mesh density as the other computer modeling studies.

Larger throat is picture below.

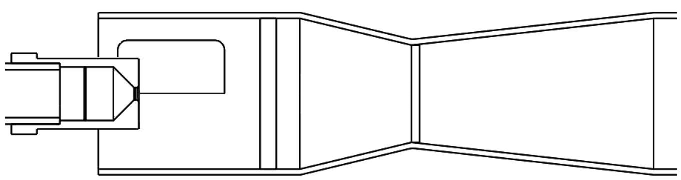

Figure 3.16 Experiment Set-Up Larger Throat Diameter

The smaller throat is pictured below.

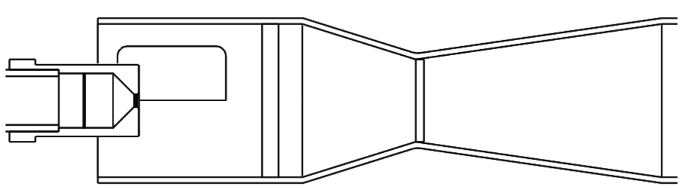

Figure 3.17 Experiment Set-Up Smaller Throat Diameter

Study Population – D.O.E. – Throat Diameter and Port Area

The factor relationship is given below. The line of restriction shows the same boundary conditions were set for all four treatments. The three degrees of freedom are given as, A, B and the interaction between A and B.

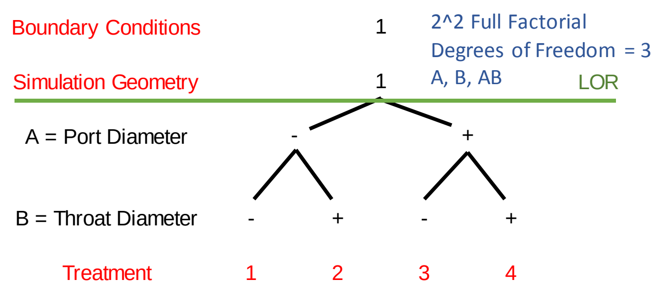

Figure 3.18 Factor Relationship Diagram Computer Modeling DOE

CHAPTER IV

RESULTS

Laboratory Experimentation

Laboratory Experiment 1 – Orifice Concentricity with Burner

The data when reviewing the off-centered orifice is given as the below. H.R. Jones uses 3.8% reduction in primary aeration per millimeter. The data from the laboratory experiment showed similar results. The laboratory collected data does not show a perfect linear function.

Orifice Position Off-Centered (Inches)	O2%	Fuel %	Air/Fuel	Primary Aeration
0	17.82	14.8	5.758	59.36
0.125	17.63	15.7	5.371	55.38
0.25	17.55	16.07	5.221	53.83

Table 4.1 Lab Experiment 1 Results

The expectation of a linear function when reviewing how an orifice being off-centered will affect primary aeration should not be expected. As the mixture stream approaches the venturi wall drag will be increased. When the mixture stream hit the walls of the throat on the venturi the gas will have to change directions to enter the burner.

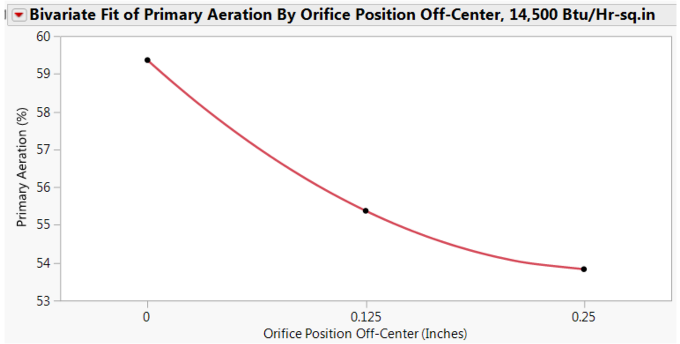

Figure 4.1 Orifice Position Off-Center vs. Primary Aeration

The data points for primary aeration shows data points that do not appear to be completely linear. From a fit model, an estimated function for the primary aeration bounded between centered and 0.25 inches out of centers is given by the following.

$$Primary\ Aeration = 58.142 - 22.13 * P + 78.14 * P^2 \tag{3.1}$$

The equation above is for this burner at this firing rate. There shouldn't be an expectation for this function to be relevant with a different burner with a different firing rate or geometry. We can however, expect the shape of the function to be similar.

Laboratory Experiment 2 – Orifice Axial Depth

The axial depth study had two parts. A computer modeling part and a laboratory part. The output of the experiment in both parts of the study was primary aeration. The two studies were expected to have slightly different outcomes. This comes from the differences in the properties of pure methane and the natural gas. The simulation used methane to simplify the computations. The exact species of the natural gas that was supplied was unknown, the noise control strategy was to capture the calorific value using a calorimeter.

The best position to maximize primary aeration was found to be in the range of 0 inches to around positive .5''. If there was an air-shutter on this burner, this range would be close to the allowable movement on the orifice. The raw data from the experiment can be seen below.

The calorific value on the test day was $1055\ {}^{Btu}/_{Ft^3}$. The firing rate of the injector is calculated to be $11{,}170\ {}^{Btu}/_{Hr}$. The port loading was calculated to be approximately $17{,}000\ {}^{Btu}/_{Hr \times in^2}$.

Orifice Position	O2%	Fuel %	Air/Fuel Ratio	Primary Aeration
-1	17.61	15.79	5.333	54.98
-0.75	17.7	15.36	5.509	56.79
-0.5	17.86	14.61	5.846	60.26
-0.25	17.98	14.04	6.122	63.12
0	18.29	12.58	6.952	71.67
0.25	18.32	12.43	7.043	72.61
0.5	18.47	11.72	7.529	77.62
0.75	18.16	13.19	6.582	67.85
1	18	13.95	6.171	63.61

Table 4.2 Lab Experiment 2 Orifice Position Results

Plotted out the data from the lab experiment can be seen below. The throat of the burner to the zero point was estimated to be just over 3 throat diameters. The trend is like what is expected from the literature. Prichard, Guy and Connor estimated the ideal range to be 2 to 3 throat diameters away from the venturi neck. This is as expected as the maximum location for primary aeration is with the orifice just inside of the burner.

The burners flames were lifting in most positions. From Prichard a typical atmospheric burner is aerated from 40% to 60% primary. From the outputs of oygen taken after two minutes, a burner with lifted ports is expected. The ports will start to stick to the burner as the burner temperature rises.

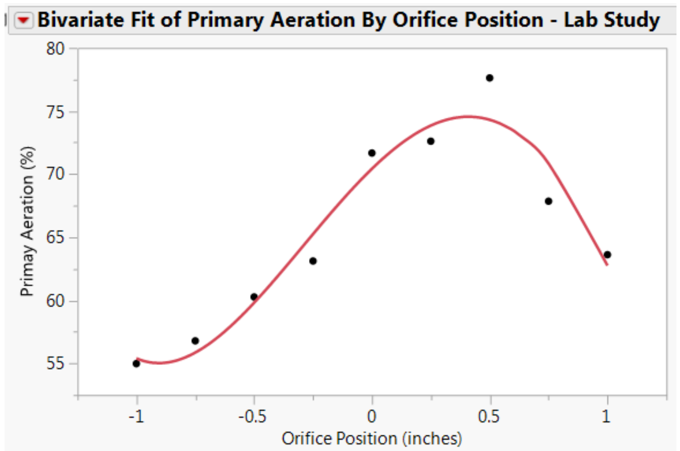

Figure 4.2 Orifice Position Depth vs. Primary Aeration

Plotted in the graph above is an estimated of how the burner will respond between the boundaries of negative 1 inch and positive 1 inch. It should be noted, this equation is only valid for this specific burner, with the amount of primary air opening in this experiment, in this particular lab set-up. The estimation within the boundaries can be seen below.

$$Primary\ Aeration = 70.425 + 18.097 * P - 14.16 * P^2 - 14.42 * P^3 + 2.81 * P^4 \quad (3.2)$$

For a burner that is exposed to a normal amount of secondary air the amount of primary aeration is too high. During the experiment some flame lift was observed until the burner reached temperature. This could be controlled by reducing primary aeration with the air-shutter that was removed from the burner.

Laboratory Experiment 3 – D.O.E. – Port Area, Orifice Diameter and Orifice Depth

From the normal probability plot the experiment showed the factor that is considered significant with the given inputs for the experiment was factor B, the orifice diameter. The next largest factor was shown to be the interaction between A and B, port area and orifice diameter respectively. The interaction between the orifice diameter and port area confirms why most texts move the port area based on the desired input for the burner.

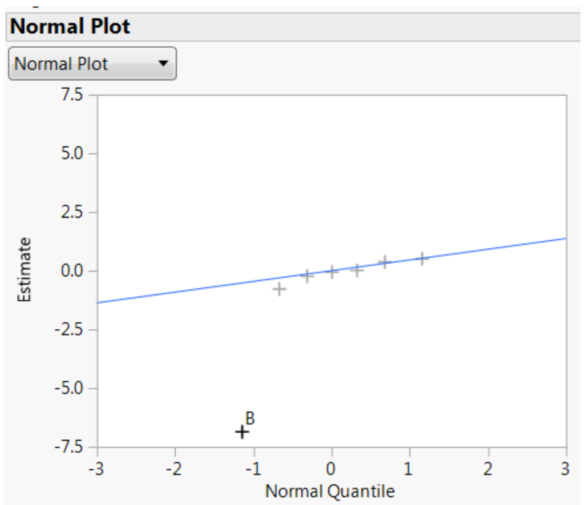

Figure 4.3 Normal Probability Plot Lab Experiment 3

Pareto estimates are seen below with the standardized effects. Orifice diameter, B, is shown to have the highest magnitude with a negative direction. The negative directions show that as the orifice diameter is increased the primary aeration percent decreases.

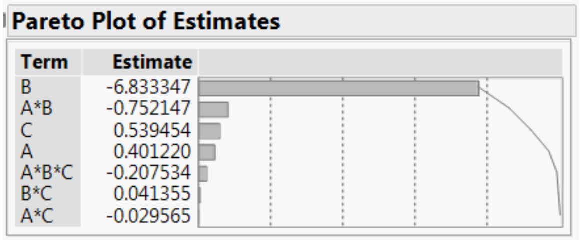

Figure 4.4 Pareto Estimates Lab Experiment 3

The interaction plot from the DOE can be seen below. There is an interaction between Port Area and Orifice Diameter, as stated above. There does not appear to be an interaction with the orifice diameter and the orifice position, as A and C appear to be parallel.

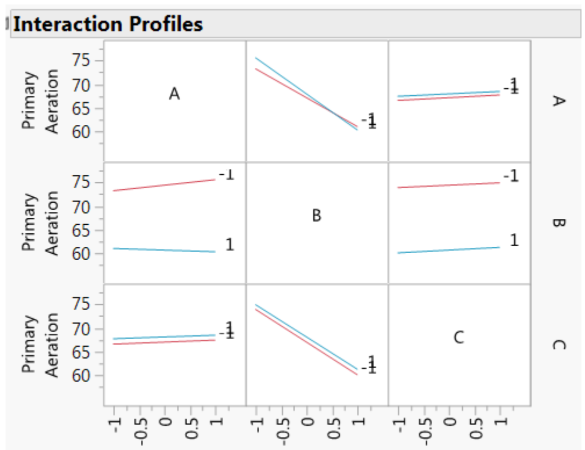

Figure 4.5 Interaction Plots Lab Experiment 3

A variability chart can be seen below. On the chart maximum normal primary aeration is displayed at 60%. To compete combustion 130% aeration is typically required from the combination of primary and secondary. Some estimates are higher than 60%, for most burners primary aeration rate slightly higher than 60% should be acceptable.

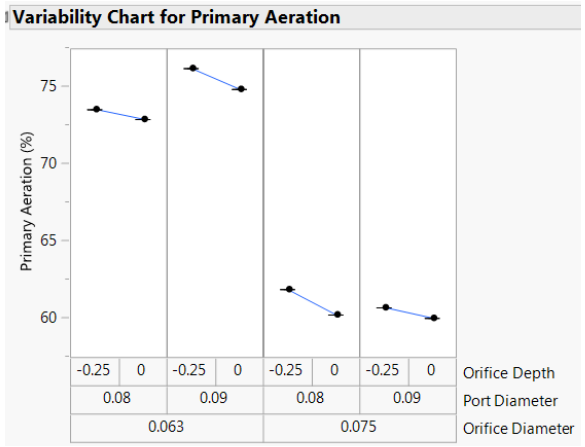

Figure 4.6 Variability Chart, Level Settings vs. Primary Aeration, Lab Experiment 3

The two charts below are control charts, sometimes called Shewhart Charts. The charts show the 3 sigma limits in the upper chart, X-Bar. These limits are calculated from the range and the sample size using a coefficient d_2. The limits show 3 standard deviations from the mean [Wheeler].

The chart on the left subgroups on the factor relationship diagram at the port diameter level. The crosses relationship gives the 4 subgroups. The leverage of variation is shown to be between subgroups because the points are outside of the 3 standard deviation limits on the x-bar chart.

The chart on the right has averaged the one and two to create subgroup 1, then averaged subgroup 3 and 4 to create subgroup 2. The chart on the right shows the leverage of variation is within subgroup.

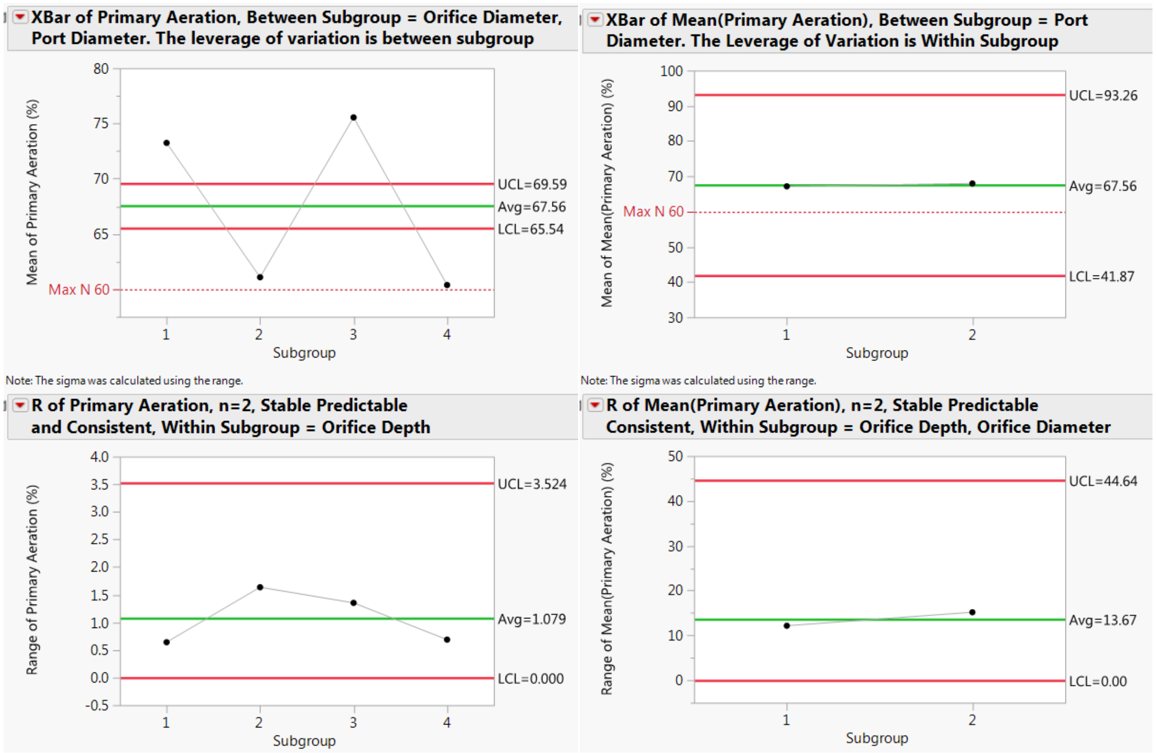

Figure 4.7 Control Chart with Subgroup, Lab Experiment 3

The two charts are pointing to the leverage of variation being orifice diameter to orifice diameter. The variance is given as,

$$\hat{\sigma}^2_{ODep} = \left(\frac{\bar{R}}{d_2}\right)^2 = \left(\frac{1.079}{1.128}\right)^2 = .915 \tag{3.3}$$

$$\hat{\sigma}^2_{ODia} = \left(\frac{\bar{R}}{d_2}\right)^2 - \hat{\sigma}^2_{OD} = \left(\frac{13.67}{1.128}\right)^2 - \left(\frac{.915}{2}\right) = 146.4 \tag{3.4}$$

The above displays numerically how much more orifice diameter plays a role in changing primary aeration. This is expected, orifice diameter controls firing rate. Firing rate is the input that drives recommended burner geometry.

Computer Modeling Simulation

Computer Modeling 1 – Orifice Axial Depth

The estimated orifice position with the highest primary aeration in the computer model is between the zero position and the negative 0.25-inch position. This is slightly shifted from the laboratory experiment. The primary aeration values were also slightly lower.

Position	Volume Fraction	Primary Aeration
-0.75	0.841	55.443
-0.5	0.842	55.924
-0.25	0.843	56.483
0	0.844	56.895
0.25	0.842	56.187
0.5	0.837	54.085
0.75	0.833	52.390

Table 4.3 Computer Simulation 1, Orifice Axial Depth

The review conducted from the computer model is expected to be slightly different than the laboratory experiment. Two differences to be noted. The simulation was run with methane, the experiment was conducted with natural gas. The simulation was run without moisture in the model, we expect there was moisture in the gas injected into the burner at the lab.

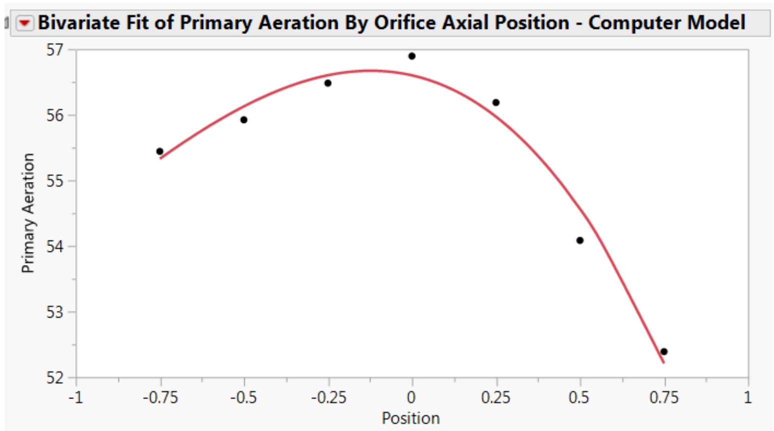

Figure 4.8 Computer Model, Orifice Depth vs. Primary Aeration

The trend from the chart above's trend follows the same form as the lab experiment. The estimated maximum orifice position for primary aeration is just past three throat diameters. An equation estimating the primary aeration based on the orifice position with the flow simulation is given as,

$$Primary\ Aeration = 56.6 - 1.16 * P - 5.03 * P^2 - 1.63 * P^3 \qquad (3.5)$$

The simulation shows the point of maximum primary aeration is right at 3 throat diameters from the venturi throat. In the flow simulation just as the lab, Prichard's recommendation of 2 to 3 throat diameters is a good range. As the burner approaches 2 throat diameters it starts to lose a source for air. It is expected the lack of a source is the reason of the losses in aeration, not the transport losses from the throat distance.

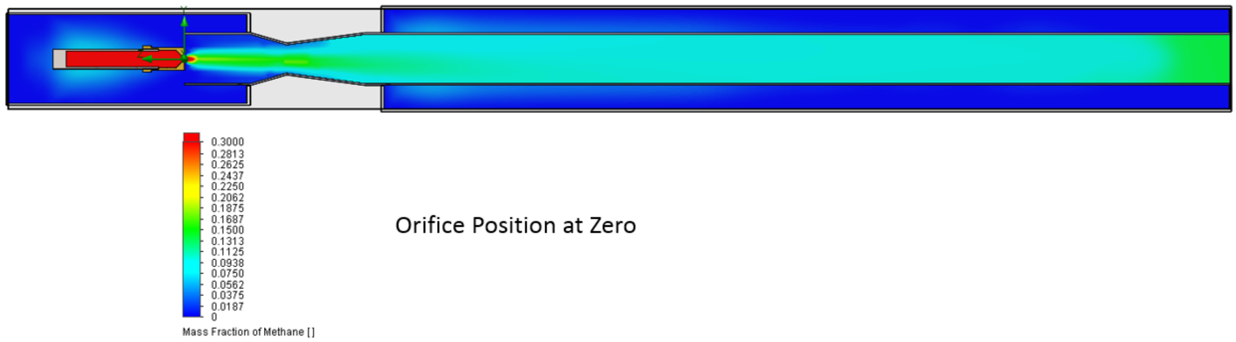

Figure 4.9 Cut-Plot, Computer Model with Orifice Depth at Zero

A screen shot of the right plane cross-section in the flow simulation gives a visual estimation of how the injector pulls air into the burner. Two points to note in the visual display. At the end of the burner, there is a point of high primary aeration. The simulation solution was not set to converge in that location. The initial condition was set above the actual primary aeration rate. The solution never refined this area of the burner. To eliminate this in the simulation, another iteration could be run with an initial condition closer to the observed mass fraction of methane. The second point to note, is the methane around the top and bottom of the ignitor, seen in white. This is also left over from the initial condition. This is as expected, there was not a significant amount of fluid flow in this region.

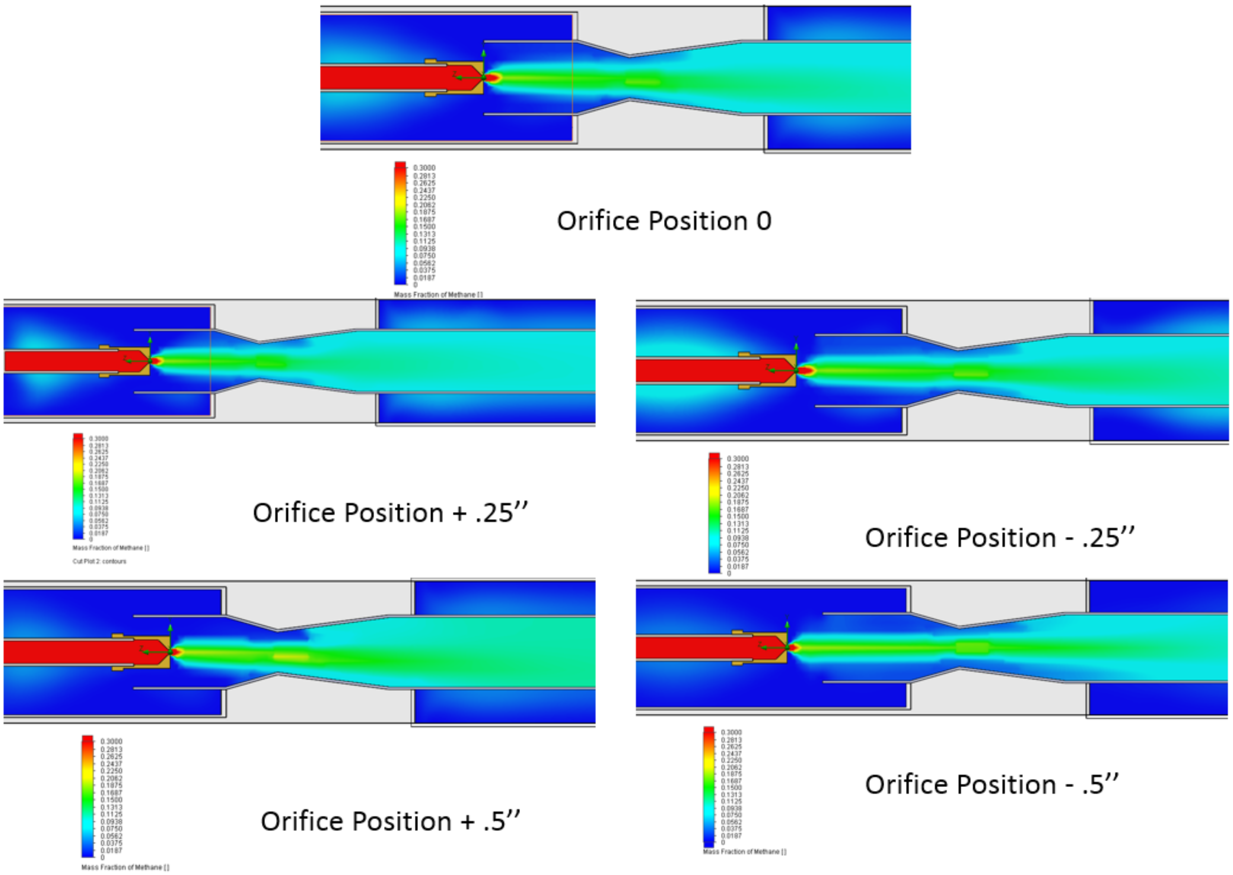

Figure 4.10 Cut-Plot, Computer Model at Venturi for all Depths

The figures above show how the flow field alters as the orifice is moved into the burner on the left side and away from the burner on the right side. As the injector is moved away from the burner, there is a lighter blue color around the walls of the inlet.

Computer Modeling 2 – Orifice Yaw

To review how the injector/orifice angular position affects primary aeration a computer model was used. The visual results from the simulation can be seen below. The top chart has the orifice perfectly centered. The second has the orifice angled at 1° and the third at 2°. The simulation set-up goals and convergent conditions were set the same as the orifice depth model. The orifice was set to a larger diameter at 1.92 millimeters. The depth of the orifice was at the positive 0.25-inch position.

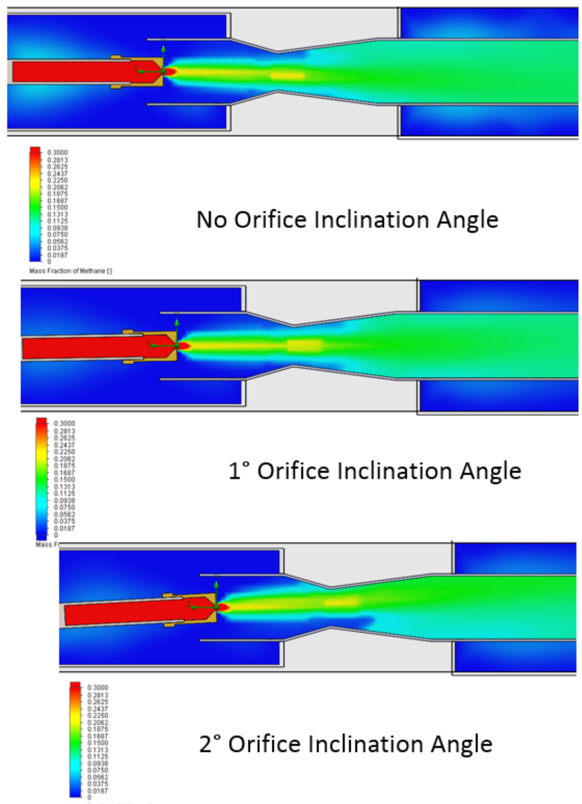

Figure 4.11 Cut-Plot, Computer Model Orifice Inclination Angle Study

The change in primary aeration from injector yaw is estimated at 2.1% per degree [H.R.N. Jones]. From the simulation, the first degree only showed a reduction of around .4 in primary air. The second degree showed a reduction of 1.471 from the first degree. This is slightly lower and not linear.

Orifice Angle (°)	Volume Fraction	Primary Aeration
0	0.809	44.413
1	0.808	44.233
2	0.803	42.762

Table 4.4 Computer Simulation 2 Results

The chart below shows the simulation did not give linear results. Linear results should not be expected. There should be an interaction with orifice position from venturi throat and the inclination angle.

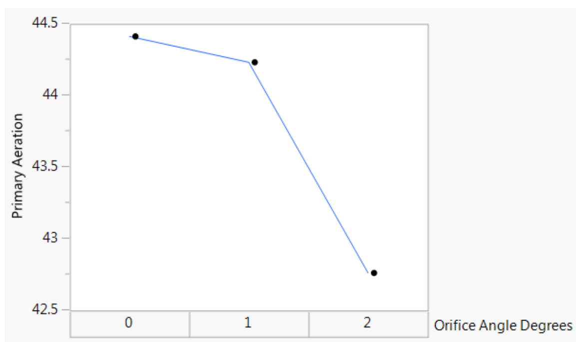

Figure 4.12 Computer Simulation, Orifice Angle vs Primary Aeration

In an appliance the injector is positioned off the unit and not the burner. The burner is generally positioned off a different wall in the unit than the injector. The stack up between these two datums make angular injector misalignment a common source for reduction in aeration losses. The drop off show above explains why the injectors angular position is a significant factor is appliance assembly.

Computer Modeling 3 – Port Diameter

The experiments injector firing rate calculated out to approximately $15{,}000\,\frac{Btu}{Hr}$. The results from the experiment can be seen below based on the given input of port size for the experiment.

Ports Size (in)	Total Port Area (sq.in)	Port Loading (Btu/hr-in)	Volume Fraction Air	Primary Aeration
0.06	0.362	41,447	0.766	34.448
0.07	0.493	30,451	0.772	35.604
0.08	0.643	23,314	0.801	42.200
0.09	0.814	18,421	0.814	46.031
0.1	1.005	14,921	0.823	48.778

Table 4.5 Computer Simulation 3, Port Diameter Results

American society of gas engineers study manual recommends circular port to be loaded from 20,000 to $25{,}000\ \frac{Btu}{Hr*in^2}$. It should be noted that the recommendation for square ports are from 15,000 to $20{,}000\ \frac{Btu}{Hr*in^2}$. It could be inferred that primary aeration be higher a lower value for circular ports. Circular ports often have more issues with lifting. This could why ASGE recommends higher port loading. The higher port loading will reduce aeration as seen below.

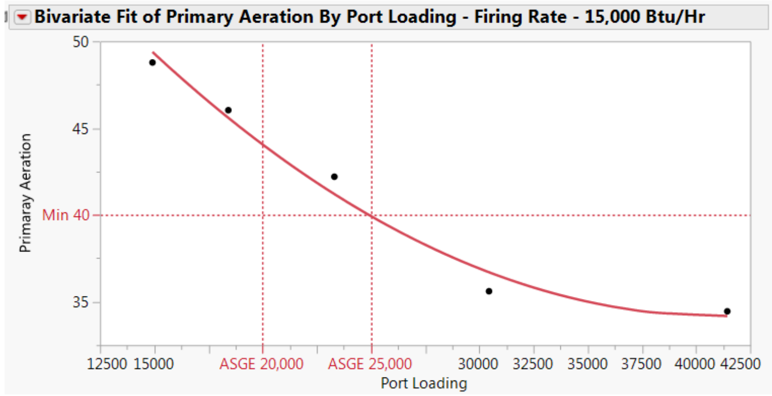

Figure 4.13 Computer Simulation, Port Loading vs. Primary Aeration

From the simulation an estimation for primary aeration can be seen below. Depending on the type of burner would determine the loading. If this was a broil burner, which does not get as much secondary air, the higher aeration with less port loading would be better. If this was a bake burner, either the 0.9-inch or the 0.8-inch port diameter would be acceptable.

$$Primary\ Aeration = 56.995 - .0006829 * PL + 2.158e^{-8} * (PL - 25710.5)^2 \qquad (3.6)$$

The relationship given from the equation above follows the expected trend, where it is bound by the experiment. If the port area was increased further with the given fire rate, the flame front would move closer to the burner. Heating the burner and eventually reduce primary aeration. The relationship above should only be followed from the bounded 15,000 to 41,500 $\frac{Btu}{Hr-in^2}$.

Computer Modeling 4 – D.O.E. – Port Area and Throat Diameter

A design of experiment was conducted using two factors. Factor A was the port diameter and factor B was the throat diameter. The normal probability plots show all factors and the interaction are significant.

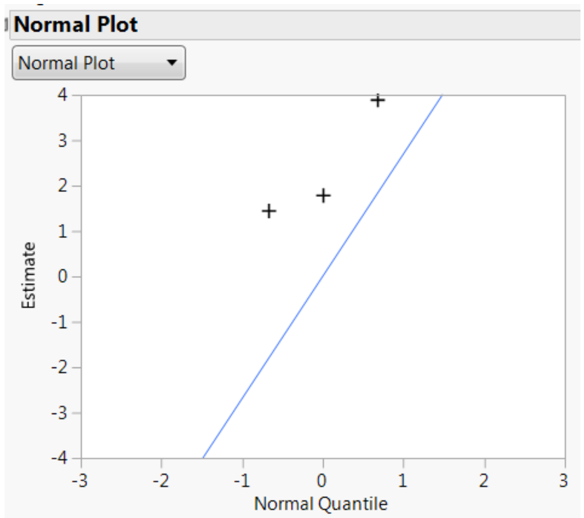

Figure 4.14 Normal Probability Plot Computer Modeling Experiment 4

The largest estimate comes from port diameter. The estimate is positive. The positive value shows as the throat diameter is increased from diameter 1 to diameter 2 primary aeration increases. The port area estimate is positive as well. Direction for the next DOE could be to pick a level setting higher for both port area and throat area if an increase in primary aeration was desired. There should be a point where increasing port area and throat area will reduce primary aeration. The port area might be delayed, the simulation does not consider burner temperature. As the port area increase the flame height will decrease. This will increase the temperature of the burner and reduce primary aeration.

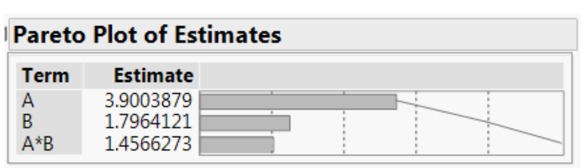

Figure 4.15 Pareto Estimates Computer Modeling Experiment 4

The interactions can be seen below. The highest primary aeration is shown to be at the level setting, A positive and B positive. The interaction between A and B is significant and should be retained in future studies. If a fractional factorial was conducted it would be important to make sure the interaction was not aliased with another factor.

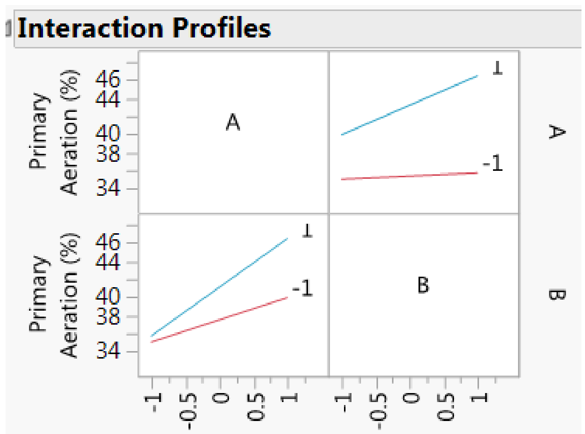

Figure 4.16 Interaction Plot Computer Modeling Experiment 4

A variability chart as seen below displays the magnitude of the level settings. For this burner with this port loading, the level setting of A at the positive level setting and B at the positive level setting could be the best design. This of course is dependent on the amount of secondary air that the burner has access to.

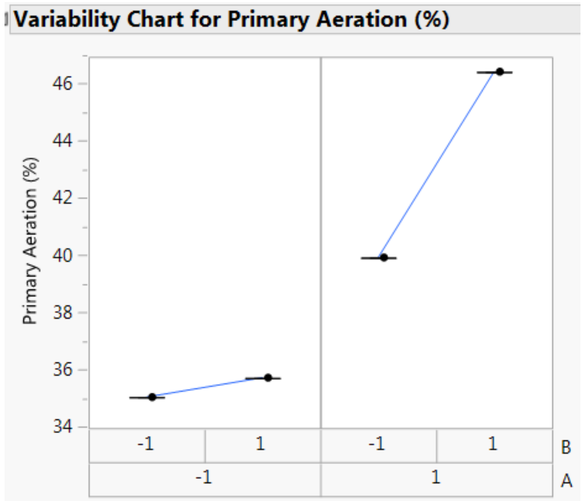

Figure 4.17 Variability Plot, Factors vs. Primary Aeration Modeling Experiment 4

CHAPTER V

CONCLUSION

The design boundaries given for a burner with a specific firing rate, unit pressure, and fuel type requirements are good estimates for performance for a given burner geometry. Using computer modeling software is an acceptable method for determining significant factors in the design of a burner. There are slight discrepancies between lab testing and computer modeling in this study. Some of the discrepancies come from unknown factors, theories are the exact species natural gas and moisture in the gas. Some of the others come from the expected difference in densities of natural gas and methane.

Estimates for loss in primary aeration are often displayed as linear in texts. That is often not the case. The losses might be linear over a small range. The linear range is dependent on factors related to the geometry of the burner. This should be understood, as some features such as tube diameter could make the range tighter or loser. An example of how primary aeration does not follow a linear path is orifice injection angle. In the lab experiment number one the decay of primary aeration is expected to be linear. The results show a curved decay. As the jet stream approaches the venturi throat there is a drop off in primary aeration that is not linear. This shows that the burner geometry is important to consideration.

Some burners such as broil burners might require more aeration than most burners. The requirement for more primary aeration is from a restriction in secondary air at the flame front. The amount of required primary and secondary air at the root of the flame requires balance. It is possible to achieve the required increase primary aeration. One method is to increase the port area, as seen in computer modeling experiment number 3.

A product such as an air-shutter allows the engineer to balance a burner. The burner could be designed with high aeration and reduced to the required aeration by choking down the inlet area. If the burner is restricted on primary air this method will not work.

Factors that are shown to be significant in burner design with a given firing rate are, port area, throat diameter, orifice position and the interaction involving throat diameter with port area. These factors are described as significant in texts and reviewed in this paper for confirmation.

Injector position is critical for burners a burners primary aeration. Any off-set of the injector relative to the burner will reduce performance. During the assembly process it is critical to insure the burner assembly is correctly orientated relative to the burner.

When designing a burner, the engineering literature provides a good starting point for the design. From the starting point, strategic experimentation must be employed with an understanding of how the design levers changes burner performance. These levers control primary aeration. The unit must also be considered, the airflow to the root of the flame with secondary aeration.

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VITA

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