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Thermal Design of a Fire Grate

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Abstract

Conventional wood burning fireplaces make inefficient space heating elements for residential use. The addition of some commercial fire grates claim to increase the heating efficiency. The purpose of this project is to test the validity of these claims using theoretical models as well as experimental analysis of heat transfer. Heat transfer and fluid dynamic equations were used to model the theoretical air flow through the fire grate. A test structure was fabricated and instrumented to gather experimental data. This data was used to compute the heat added by the fire grate and the results were compared to the theoretical model. These calculations yielded roughly 625 Watts added to the room from one, four-piped insert. This is equivalent to a 40% increase in heat production from a similar fireplace without an insert. From this it is concluded that the addition of a fireplace insert similar to the one used in this project with a blower attachment is beneficial for space heating.

Acknowledgments

I would like to thank Professor John M. Sullivan for all of the guidance and patience he provided throughout this project. I would also like to extend my thanks to Neil Whitehouse for use of his lab as well as his guidance with the design. Finally I would like to thank Randall K. Harris, and Toby Bergstrom without whom the testing could not have been completed.

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Introduction

Fireplaces are a common commodity in residential American homes, and are used both for aesthetic purposes and as a source of space heating. Unfortunately, a fireplace can be a very inefficient source of heat since large quantities of heat and residential air exit via the chimney. While fireplace inserts are readily available to the general public, the claims that they provide a significant increase in the heat output of the fire are not supported with quantitative, publicly available data. The purpose of this project is to model the heat and fluid flow of a fireplace, construct and instrument a fireplace grate, and analyze the data collected to measure the quantity of mass flow and heat being exchanged. These results are compared to simplified theoretical estimates of the fireplace system.

The background research included benchmarking research of fireplace inserts, typical dimensions of fireplaces, and typical costs for space heating in New England. The results are summarized subsequently.

Fireplaces are a common commodity in American homes. In 1997 it was estimated that there were 27 million fireplaces nationwide that were used as a primary space heating source (Kochera, 1997). More than 7 million of these installations utilized a fireplace insert to increase the efficiency of heat output (Bucklin Smith & Associates, Inc., 1989-1996). This is due to the fact that a fireplace alone is a very inefficient source of heat since all or most of the heat generated by the burning of the wood flows out via the chimney. Fire grates are designed to mitigate this loss in heat using various techniques.

Fire grates are readily available for purchase to the general public. A patent review was conducted to better understand their function as well as their advantages and disadvantages. A fire grate is typically a series of metallic tubes that lie within close proximity to a fire's logs and have inlets and outlets that allow for air exchange with the room containing the fireplace.

The two main types of fire grates consist of vertical and horizontal grates. Horizontal grates, like the one described in U.S. Patent 4,010,729 places the inlet and outlet of the grate horizontal with respect to each other. A schematic of this patent is provided in Figure 1: Schematic of U.S. Patent 4,010,729 - Example of a Horizontal Fireplace Grate with BlowerFigure 1. The logs sit atop the piping, heating the air which, in this case, is being pumped through via a small squirrel cage blower.

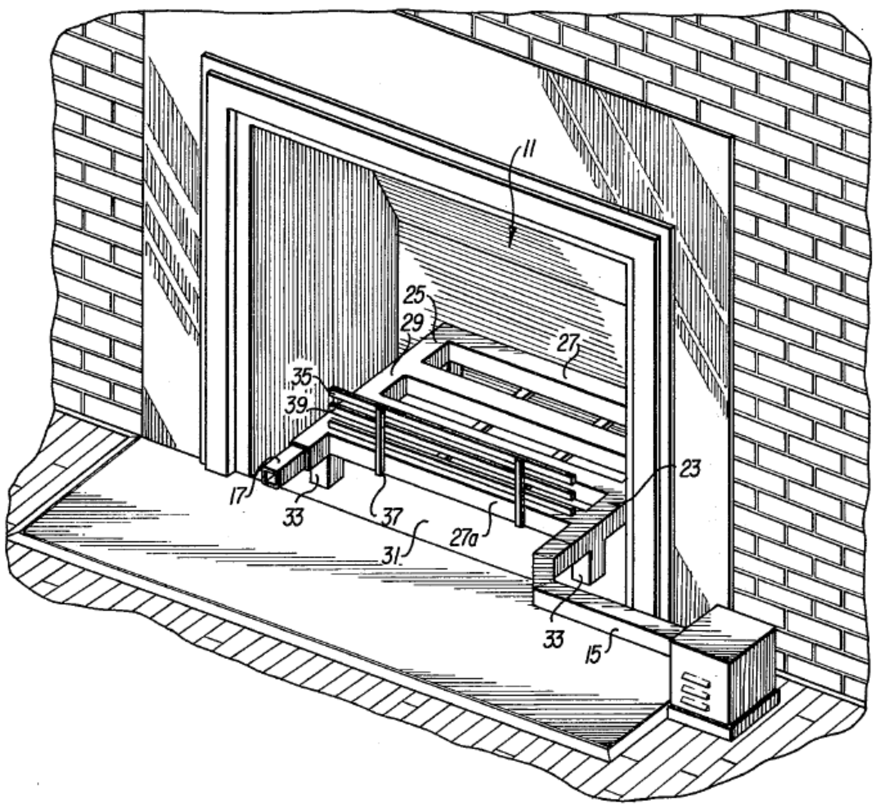


Figure 1: Schematic of U.S. Patent 4,010,729 - Example of a Horizontal Fireplace Grate with Blower (Egli, 1975)

One advantage of a horizontal blower is its shape does not need to be changed for various fireplaces. By adding variable length piping for inlet and outlet sections, as done in patent 4,010,729, the grate compensates for fireplaces that have varying depths. However, horizontal grates receive all or most of their heat transfer from the conduction between the logs and the piping. A disadvantage to a horizontal grate is the requirement of a blower to induce air flow through the grate. While squirrel cage blowers are small, they are not as aesthetically pleasing due to noise generation. This design also requires electricity to operate which is counterproductive to increasing heating efficiencies.

The second common fireplace insert orients the inlet and outlets vertically with respect to one another. These vertical fire grates tend to be shaped roughly like a "C" and typically consist of several pipes configured parallel to each other. They are arranged in such a way that they extend back into the fireplace then climb quasi-parallel to the back wall, then angle back toward the room. Patent 4,129,113 clearly displays the typical layout of a vertical fireplace grate in Figure 2.

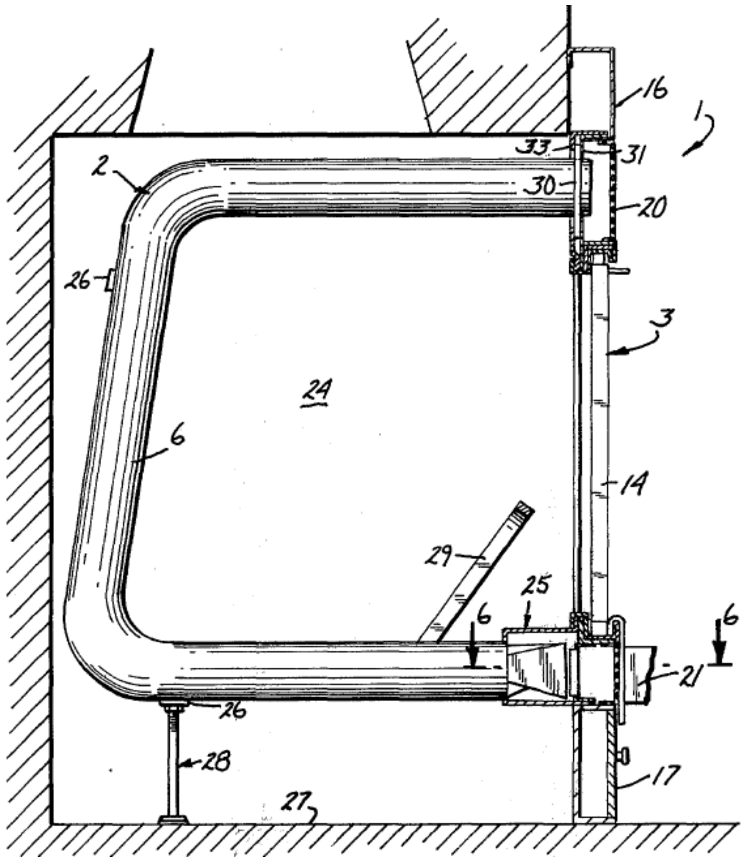


Figure 2: Side View Schematic of U.S. Patent 4,129,113 - Example of a Vertical Fireplace Grate with Glass Door Faceplate (Bergstrom, 1977)

The grate seen in Figure 2 features a glass door facade for aesthetic and heat conservation purposes. Vertical grates offer many advantages over horizontal inserts. One advantage is that vertical grates receive heating not only from conduction between the logs and the piping but also significant radiation heating since the piping surrounds the fire more completely. Another advantage vertical grates have over horizontal ones is that natural convection helps drive the air flow through the piping eliminating the need for a blower. A blower can still be attached to attain greater flow velocities if desired. The major disadvantage of vertical fireplace grates is their restricting shape. The back wall of most fireplaces are not strictly vertical but rather angle out toward the room halfway up the wall. This requires the piping to be angled similarly in order to fit within the fireplace. This can cause the shape to vary slightly for each fireplace.

Masonry fireplaces come in a variety of sizes from about 0.64 meters to as much as 1.2m wide, by roughly 0.61m to 0.91m high, and 0.20m to 0.56m deep. For our considerations, the firebox dimensions are assumed to be 0.61 m wide, by 0.76m tall, by 0.45m deep. Fire wood that is burned in such fireplaces is typically sold in 16 inch long pieces. However, while a fire is burning, the outer portion of the logs is not burning as hot as the center of the fire. Therefore, for calculation purposes, the heat source dimensions will be assumed 0.23 m long by 0.15 m tall.

For this project, a vertical fireplace grate was designed similar to the one depicted in Figure 2. This was chosen because of the natural air flow produced removed the requirement for a blower to produce flow through the pipes. This fire grate consists of several metallic pipes fitted together such that they fit within a fireplace and will function as a stand for logs to sit upon. These pipes are open to the room in such a manner that the air within them can be heated via the burning logs. This heated air, through natural or forced convection, flows out into the adjacent room supplying additional heating to the room.

This grate was fabricated out of black galvanized steel malleable iron piping purchased at a local Home Depot store. The ends of the individual pieces are threaded in such a way that assembly simply consists of screwing together adjacent pieces together. This allows for easy access to the interior of the piping for placement of temperature reading devices.

A Computer Aided Design (CAD) model was built to visually represent the design for the fire grate. This was done in SolidWorks Education Edition. The piping that makes up the C shape is shown in Figure 3. The lengths of each segment are suitable such that it would fit within a standard fireplace. The bottom portion of the tubes is divided into segments to provide access to the interior for test instrumentation. The 90 degree elbow would likely be placed such that is touching the back of the firebox of the fireplace. A 45 degree elbow is placed above the 90 degree elbow to accommodate for the slanted back walls of fireplaces. A second 45 elbow is introduced to redirect the air flow back out into the room. The design is built such that the entrance and exit pipes extrude out into the room roughly 0.05 m so as to avoid mixing of the desired heated air with toxic fumes from the fire. The overall assembly would consist of four of these pipes placed next to each other.

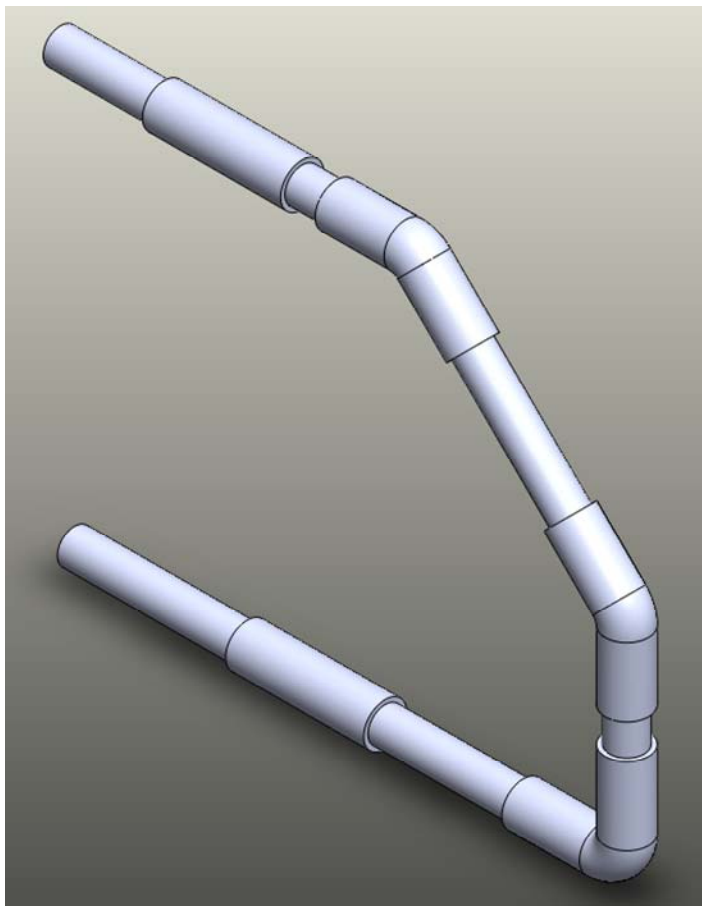


Figure 3 : SolidWorks Rendering of Fireplace Grate "C Pipe"

These pipes were designed to be held up by a bracket device that is bolted together using bolts similar to those shown in Figure 4. These clamps will be bolted into a plate of metal to connect adjacent tubes together. The end clamps in turn will be bolted onto legs as can be seen in Figure 5. These legs would keep the logs elevated above the floor to promote increased air flow as well as keep the unburned logs from being smothered by the ash that collects below the fire.

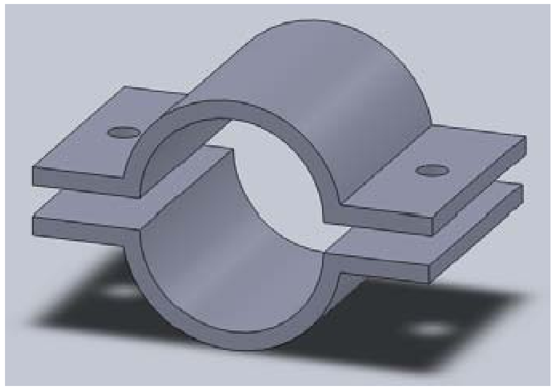


Figure 4 : SolidWorks Rendering of Pipe Securing Clamp

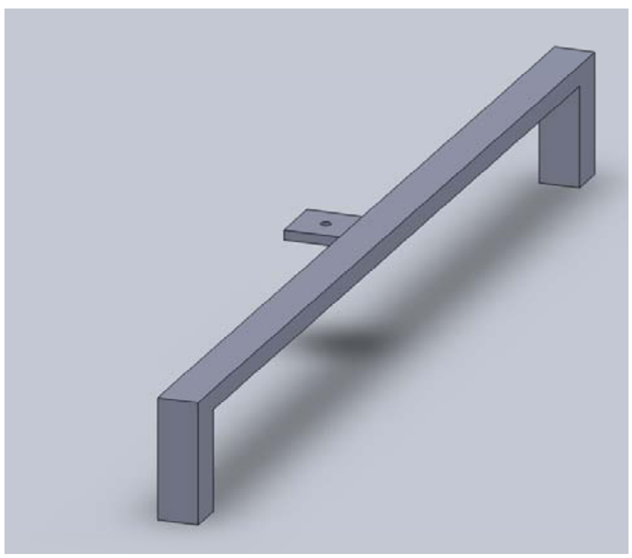


Figure 5 : SolidWorks Rendering of Fire Grate Support Legs Concept

This assembly is meant to be inserted into the fireplace such that the logs to be burnt would be placed on top of the C-shaped pipes. The burning of the logs provides heating to the air contained within the tubes thus producing a flow.

There are three basic methods of heat transfer: convection, conduction, and radiation. Convection heat transfer occurs whenever fluids flow over a solid of a different temperature than that of the fluid. In all cases heat will transfer from the hotter entity to the cooler one according to the second law of thermodynamics. Similarly, conduction heat transfer refers to the flow of heat from a hot surface to that of a cooler surface that is touching the heated surface. In this way conduction only applies when there is physical contact between two surfaces and a temperature gradient exists. Radiation, the third mode of heat transfer, occurs at all times between all objects that are of differing temperatures. Any hot surface radiates heat based on the temperature gradient between the hot surface and its surroundings.

There is no air flow from an open fire into a room and those enjoying its warmth so we can reason that the heat transfer due to convection is negligible. Indeed, if there were significant flow from the fire into the room, it would be filled with toxic chemicals contain within the smoke and as such air usually flows from the room and into the fireplace. Similarly, open fires have negligible effects on heating a room via conduction since the on physical contact is between molecules of air which are so fleeting and random they are practically zero. It can thus be reasoned that the only significant heat transfer that occurs between an open fire and the adjacent room is due to radiation.

With this knowledge of heat transfer and its corresponding equations we can predict the behavior of a fireplace grate quantitatively. This allowed us to quantitatively estimate the heat transfer of a grate system using well established governing physics.

A fire puts out a significant amount of heat per second when burning at full force. It can be assumed that the amount of heat felt by a human standing three feet away from a fire is comparable to the amount of heat felt if that person was standing three feet away from a 1500 watt resistance heater. If we take this comparison to be true, we can assume that a fire radiates roughly 1500 watts of heat out into a room. The fire radiates heat in all directions at all times, however we assume that the portion of heat radiated felt by the human is within a 45 degree angle. With these assumptions we can calculate the expected temperature at which the fire burns within the fireplace if the ambient temperature of the room is 21 degrees Celsius.

Equation 1 : Rate of Net Radiation Heat Transfer Per Unit Area Between Two Surfaces (Cengel, Turner, & Cimbala, 2008)

Let

T_f = Temperature of the fire

Q = Emittance felt in front of fire

$$Q := 1500W$$

σ=Stefan-Boltzmann constant

$$\sigma := 5.671 \cdot 10^{-8} \frac{W}{K^4}$$

α=Angle of Radiation

$$\alpha := 45$$

$$T_{room} := 21\ °C = 294.15 \cdot K$$

f = Estimated Shape Factor

$$f := \frac{\alpha}{360} = 0.125$$

Q1 = Total emmittance from fire

$$Q_1 := Q \cdot \frac{1}{f} = 1.2 \cdot 10^4 W$$

$$Q_1 = \sigma \cdot \left(T_f^4 - T_{room}^4\right)$$

$$12000 = \sigma \cdot \left(T_f^4 - T_{room}^4\right)$$

$$T_f^4 = \frac{12000}{\sigma} + T_{room}^4$$

$$T_f := \sqrt[4]{2.191 \cdot 10^{11}}\ K = 684.087K$$

$$684.087K = 410.937\ °C$$

$$684.087K = 771.687\ °F$$

This value of how hot the fire burned gave a temperature to aim for when performing tests. This value is also backed by findings of masonry fireplace studies (Peacock, 1987). It also allowed for a thermocouple to be chosen for temperature data collection when testing the grate system.

Thermocouples are one of the most of common means of measuring the temperature of specific points in a system because they are compact, durable, repeatable and have a fast temperature response time. They utilize the Seebeck effect which converts thermal energy to electrical energy. The thermal energy is in the form of a change in temperature at one node which creates a voltage within the loop. The setup used in my project consists of a thermocouple made of two different metals that are joined by a third metal to create a junction. This junction is placed in the environment where temperature is to be measured. The two metals are joined at a second junction, typically copper which connects the thermocouple to a digital data reader. As long as the two metals making up the thermocouple remain at the same temperature at the copper junction, there will be no interference with the reading. This setup is illustrated in Figure 6 obtained from the Maxim website on thermocouple basics (Maxim Integrated Products, 2007).

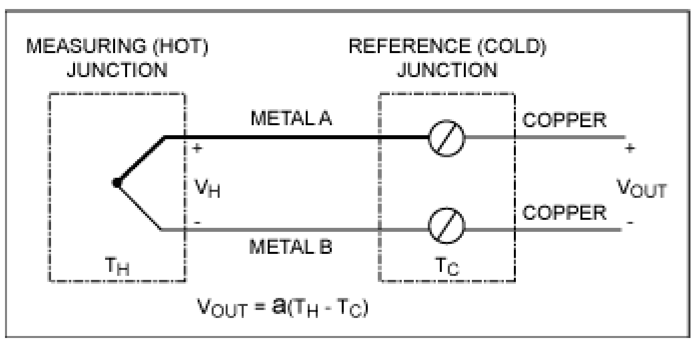


Figure 6 : Thermocouple Schematic

The copper junction is known as the cold junction while the measured junction is called the hot junction. The voltage read by the digital data reader is found by multiplying the temperature difference between the two junctions by a scaling factor α, called the Seebeck coefficient. This value can be looked up using reference tables. According to the equation at the bottom of Figure 6, the only way of finding the voltage change is to know the temperature at the cold junction. An easy way to know this reference temperature is to create an ice bath. However since ice baths are not practical in most environments, one can use the ambient temperature at the cold junction. Unfortunately this introduces another voltage into the loop which must be compensated. This compensation is called the cold junction compensation or CJC. Without this compensation, there would be an inherent error in all readings because the voltage being read consists not only of the voltage change from the hot junction but from the cold junction as well.

The two different metals that make up the thermocouple wires are typically chosen based on the temperatures they will be subjected to. Thermocouples come in a variety of types based on their temperature range of operation as well as their sensitivity. Based on the predicted temperature of a fire, a type K thermocouple was chosen. A type K thermocouple is one of the most commonly used thermocouple types because of its wide temperature range and low cost. Type K thermocouples are made of chromel (90% Ni, 10% Cr) and alumel (95% Ni, 2% Mn, 2% Al, 1% Si), can operate in environments ranging from about -200°C - +1200°C, and have a sensitivity of about 41 $\mu V/^{\circ}C$ (Pico Technology).

Using basic assumptions and simplified fluids equations, the expected velocity of the air flow exiting the room via the chimney flue can be estimated. The flow of a fluid can be classified as either laminar, turbulent, or a mixture of the two. The flow type is determined by the value of the Reynolds number (Re). Laminar flow is generally a very smooth flow with no disruption or mixing motions. On the other hand, turbulent flow is very choppy with very random motion of individual particles within the fluid. The Reynolds number is a function of the velocity of the fluid flow, the geometry of the pipe/duct through which the fluid is flowing, and the viscosity of the fluid, in such a way as subsequently defined. If the fluid velocity is assumed to have a constant profile throughout the pipe (or chimney flue) then the velocity can be depicted by the constant “u”.

Equation 2 : Relationship between Reynolds Number and Flow Velocity (Cengel, Turner, & Cimbala, 2008)

$$Re := \frac{\delta\ u\ d_h}{\mu}$$

In Equation 2 dh is the hydraulic diameter of the flue, μ is the absolute dynamic viscosity of the air, and δ is the density of the air at an assumed temperature. Natural convection flows are assumed to be both turbulent and laminar in nature which corresponds to a Reynolds number of around 2300 (Cengel, Turner, & Cimbala, 2008). This is the point at which most viscous flows enter the transition stage between laminar and turbulent. Therefore, by assuming a value of 2300 for the Reynolds number, we can calculate the expected flow velocities of heated air up the chimney. The chimney flue diameter was estimated to be about 0.254m (10 inches) (The Engineering Toolbox, 2005). In a study performed by Richard Peacock on masonry fireplaces, log fires were burned at steady state around 400°C with the use of a fireplace insert. The temperature of the air flowing through the chimney was found to be roughly 100 degrees C. Using this as an assumed temperature of the flue gasses and the Reynolds number, we can estimate the amount of heat flowing up the chimney over a period of time.

Equation 3 : Estimated Heat Loss up Chimney Flue (Part 1)

Assume:
Consider dry air at 400°C

$$\rho_{400} := 0.508 \quad \frac{kg}{m^3}$$

$$\mu_{400} := 32.754 \cdot 10^{-6} \quad \frac{N \cdot s}{m^2}$$

For now we will assume the Reynold's Number to be 2300

$$Re := 2300$$

From the definition of the Reynold's Number:

$$Re = \frac{\rho \cdot u \cdot d_h}{\mu}$$

where u is the flow velocity and d_h is the hydraulic diameter

Assume a circular flue of radius 0.203 meters or 8 inches
and a square flue of equal cross sectional area.

$$d_h = 2 \cdot r_{flue} \quad \text{where} \quad r_{flue} := 0.203 \quad m$$

$$d_h := 2 \cdot r_{flue} = 0.406$$

$$A_1 := \pi \, r_{flue}^{\;2} = 0.129 \quad m^2$$

$$l_{flue} := \sqrt{A_1} = 0.36 \quad m$$

$$d_{h2} := l_{flue} = 0.36 \quad m$$

$$A_2 := A_1$$

We can solve for velocity to predict expected flow velocities up the chimney flue.

$$u = \frac{Re \cdot \mu}{d_h \cdot \rho}$$

$$u_{400} := \frac{Re \cdot \mu_{400}}{d_h \cdot \rho_{400}} = 0.365 \quad \frac{m}{s}$$

$$u_{400sq} := \frac{Re \cdot \mu_{400}}{d_{h2} \cdot \rho_{400}} = 0.412 \quad \frac{m}{s}$$

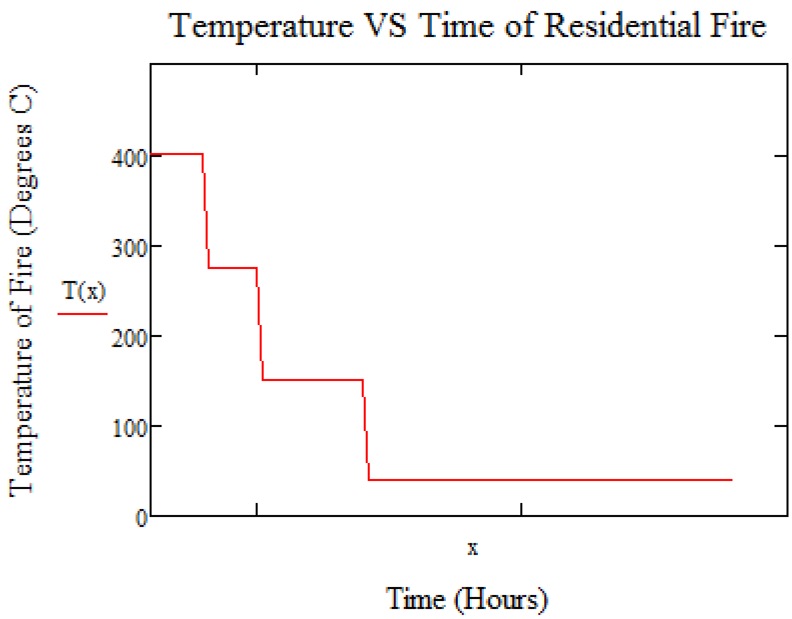
The volumetric flowrate of air up the chimney can be calculated by multiplying the flow velocity by the cross-sectional area of the flue.

$$Vol_{rate1} := u_{400} \cdot A_1 = 0.047 \quad \frac{m^3}{s}$$

$$Vol_{rate1sq} := u_{400sq} \cdot A_2 = 0.053 \quad \frac{m^3}{s}$$

Equation 4 : Estimated Heat Loss up Chimney Flue (Part 2)

We can make assumptions of the fire's behavior over time using intuition. Consider a typical cold night in New England. After dinner a fire is made such that the fire attains its maximum temperature of 400^{0}C at 8:00 PM. The fire burns around 400 degrees for an hour after which we can consider the temperature to be 275 degrees C. The fire again holds this temperature for an hour due to the residents tending to it and adding logs as necessary. At 10:00 PM the fire is starting to die down and holds a temperature of 150^{0}C. The residents grow tired and go to bed, however the fire is still too hot to safely close the flue, and consequently it gets left open all night. From midnight until 7:00 AM the fire remains at 100 degrees C. The graph below depicts the temperature of the fire as a function of time.



We can now find the volume of air that needs replacing during the time at which the fire burned at 400^{0}C using:

$$Vol_1 = Vol_{rate1} \cdot t_{400}$$

$$Vol_{1sq} = Vol_{rate1sq} \cdot t_{400}$$

$$Vol_1 := Vol_{rate1} \cdot 3600 = 170.234 \quad m^3$$

$$Vol_{1sq} := Vol_{rate1sq} \cdot 3600 = 192.089 \quad m^3$$

Similarly we can find the volume of air that needs replacement for every temperature interval.

Equation 5 : Estimated Heat Loss up Chimney Flue (Part 3)

For T= 275 C:

$$\mu_{275} := 2.849 \cdot 10^{-5} \quad \frac{N \cdot s}{m^2}$$

$$\rho_{275} := 0.6418 \quad \frac{kg}{m^3}$$

$$u_{275} := \frac{Re \cdot \mu_{275}}{d_h \cdot \rho_{275}} = 0.251 \quad \frac{m}{s}$$

$$u_{275sq} := \frac{Re \cdot \mu_{275}}{d_{h2} \cdot \rho_{275}} = 0.284 \quad \frac{m}{s}$$

$$Vol_{rate2} := A_1 \cdot u_{275} = 0.033 \quad \frac{m^3}{s}$$

$$Vol_{rate2sq} := A_2 \cdot u_{275sq} = 0.037 \quad \frac{m^3}{s}$$

$$Vol_2 := Vol_{rate2} \cdot 3600 = 117.203 \quad m^3$$

$$Vol_{2sq} := Vol_{rate2sq} \cdot 3600 = 132.25 \quad m^3$$

For T= 150 C:

$$\mu_{150} := 2.393 \cdot 10^{-5} \quad \frac{N \cdot s}{m^2}$$

$$\rho_{150} := 0.8345 \quad \frac{kg}{m^3}$$

$$u_{150} := \frac{Re \cdot \mu_{150}}{d_h \cdot \rho_{150}} = 0.162 \quad \frac{m}{s}$$

$$u_{150sq} := \frac{Re \cdot \mu_{150}}{d_{h2} \cdot \rho_{150}} = 0.183 \quad \frac{m}{s}$$

$$Vol_{rate3} := A_1 \cdot u_{150} = 0.021 \quad \frac{m^3}{s}$$

$$Vol_{rate3sq} := A_2 \cdot u_{150sq} = 0.024 \quad \frac{m^3}{s}$$

$$Vol_3 := Vol_{rate3} \cdot 3600 \cdot 2 = 151.423 \quad m^3$$

$$Vol_{3sq} := Vol_{rate3sq} \cdot 2 \cdot 3600 = 170.863 \quad m^3$$

Note: The length of time the fire is held at 150 C changes and thus there is a multiplier in the Volume 3 equation.

Equation 6 : Estimated Heat Loss up Chimney Flue (Part 4)

For T=40 C:

$\mu_{40} := 1.913 \cdot 10^{-5} \quad \frac{N \cdot s}{m^2}$

$\rho_{40} := 1.127 \quad \frac{kg}{m^3}$

$u_{40} := \frac{Re \cdot \mu_{40}}{d_h \cdot \rho_{40}} = 0.096 \quad \frac{m}{s}$

$u_{40sq} := \frac{Re \cdot \mu_{40}}{d_{h2} \cdot \rho_{40}} = 0.109 \quad \frac{m}{s}$

$Vol_{rate4} := A_1 \cdot u_{40} = 0.012 \quad \frac{m^3}{s}$

$Vol_{rate4sq} := A_2 \cdot u_{40sq} = 0.014 \quad \frac{m^3}{s}$

$Vol_4 := Vol_{rate4} \cdot 3600 \cdot 7 = 313.715 \quad m^3$

$Vol_{4sq} := Vol_{rate4sq} \cdot 7 \cdot 3600 = 353.99 \quad m^3$

These volumes can be converted to kilograms of air using the simple relationship: Volume*density=mass

$m_{400} := Vol_1 \cdot \rho_{400} = 86.479 \quad kg$

$m_{400sq} := Vol_{1sq} \cdot \rho_{400} = 97.581 \quad kg$

$m_{275} := Vol_2 \cdot \rho_{275} = 75.221 \quad kg$

$m_{275sq} := Vol_{2sq} \cdot \rho_{275} = 84.878 \quad kg$

$m_{150} := Vol_3 \cdot \rho_{150} = 126.363 \quad kg$

$m_{150sq} := Vol_{3sq} \cdot \rho_{150} = 142.585 \quad kg$

$m_{40} := Vol_4 \cdot \rho_{40} = 353.557 \quad kg$

$m_{40sq} := Vol_{4sq} \cdot \rho_{40} = 398.946 \quad kg$

$m_{total} := m_{400} + m_{275} + m_{150} + m_{40} = 641.619 \ kg$

$m_{totalsq} := m_{400sq} + m_{275sq} + m_{150sq} + m_{40sq} = 723.99 \quad kg$

This is the amount of air that must be replaced and subsequently heated during the time in which the flue is left open. This air must come in from the outside, which is assumed to be at 0 degrees C and thus must be heated. The energy required to heat this replacement air to room temperature (25 C) is found using:

$$E_{total} = m_{total} \cdot c_{p0} \cdot \Delta T = m_{total} \cdot c_{p0} \cdot (T_{25} - T_0)$$

$c_{p0} := 1006 \quad \frac{J}{kg \cdot K}$

$E_{total} := m_{total} \cdot c_{p0} \cdot 25 = 1.614 \times 10^7 \ J$

$E_{total\_sq} := m_{totalsq} \cdot c_{p0} \cdot 25 = 1.821 \times 10^7 \quad J$

$E_{total\_kwh} := 16.14 \cdot 10^6 J = 4.483 \cdot kW \cdot hr$

$E_{total\_sq\_kwh} := 1.027 \cdot 10^7 J = 2.853 \cdot kW \cdot hr$

$$\frac{E_{total\_sq}}{E_{total}} = 1.128$$

Equation 7 : Estimated Heat Loss up Chimney Flue (Part 5)

Therefore it takes around 4.5 kWh's of energy to heat the air that is replacing what is lost up the chimney for a circular flue. With a square flue of equal area, it would take roughly 13% more energy to heat that escaping air. For this reason we can dismiss the square flue in future calculations. We assumed earlier that the fire provided 1500 Joules/second of heat to the room. Thus the amount of energy the fire provides toward the 4.5 kWh is found using:

$$E_{fire} := 1500 \cdot 3600 = 5.4 \times 10^{6} \quad J$$

$$E_{fire\_kWh} := 5.4 \cdot 10^{6} J = 1.5 \cdot kW \cdot hr$$

According to these assumptions, the fire would be able to produce the required energy to heat the replacing air in the first three hours of operation. Since the fire is used for only three hours the net heating gains are negligible, adding only to the aesthetics of the room. It is for this reason that many families use glass doors in front of their fireplaces.

We will now examine the differences had glass doors been shut when the family was finished using the fire.

If the glass doors are shut at midnight we can assume that the velocity of the air traveling up the chimney drops essentially to zero. Therefore we can drop the mass of air needing replacement at 40 C from m_total. The energy to heat the air then changes as follows:

$$m_{total\_glass} := m_{400} + m_{275} + m_{150} = 288.063 \quad kg$$

$$E_{total\_glass} := m_{total\_glass} \cdot c_{p0} \cdot 25 = 7.245 \times 10^{6} \quad \text{J or 2.013 kWh}$$

The fire still provides the same amount of heat to the room and so there is a net gain in heat into the house from the fire of about 2.5 kW's.

According to the previous calculations a fireplace alone might add heat to a house or even loose heat based on damper positioning. Instead of constantly adjusting the glass doors to increase fireplace efficiency the use of fireplace inserts can be used.

Methodology

In order to estimate the performance of commercial, convection based fire grates, physical tests were run in addition to the theoretical calculations. To this end, an individual vertical grate component was constructed and tested. For time and efficiency only one C pipe was built since the results could simply be multiplied by the number of pipes. A section view of the C pipe built is shown in Figure 7. This figure also shows the placement of type K thermocouples (TCs) used in testing.

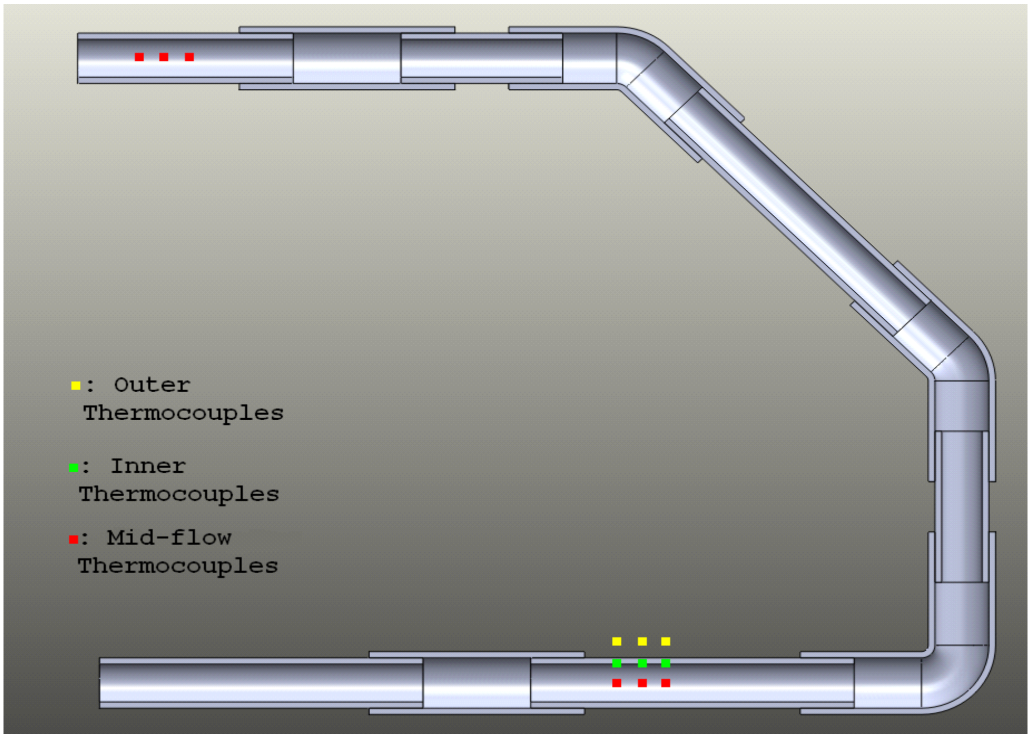


Figure 7 : Cross Section View of Test Rig with Thermocouple Placement

These thermocouples were placed such that temperature readings could be taken at key locations within the rig. Three thermocouples were attached to the outside of the piping using IDEAL stainless steel SAE size #12 pipe clamps purchased at Home Depot. These clamps had a range of 0.5 inches to 1.25 inches. The three thermocouples were secured by using two clamps in tandem. The thermocouples were placed on the top of the pipe where the logs would sit and were used to measure the temperature of the fire. A photo of this is available in Figure 8.

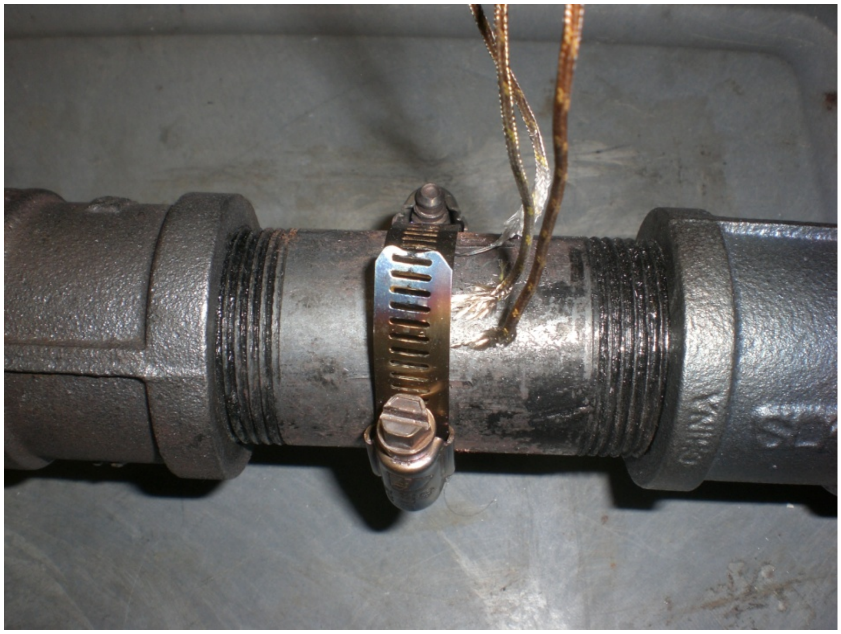


Figure 8 : Outer Thermocouple Configuration Close Up

Three more thermocouples were secured inside the piping by spot welding the hot junctions to a smaller diameter pipe than the C pipe. This smaller pipe was then inserted until the thermocouples welded to it were approximately below the outer TCs. Additionally three TCs were secured such that their hot junction lied approximately in the center of the inner pipe. There were three small holes drilled into the outside of the inner pipe to provide access to the center. These TCs were used to take the temperature of the air midflow within the pipe. These midflow thermocouples were secured to the inner pipe by use of Hercules® Regular Body High-Heat Furnace Cement purchased at Home Depot. Figure 9 and Figure 10 illustrate the thermocouple placement on the inner pipe.

Figure 9 : Inner Pipe Thermocouple Layout

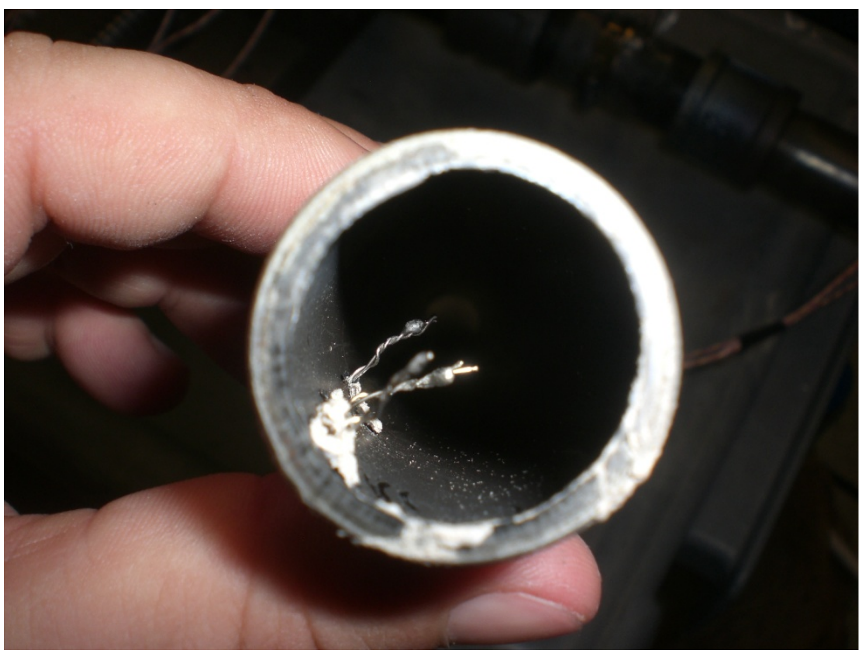


Figure 10 : Inner Pipe Midflow Thermocouples Located Near Pipe Entrance

Finally, three similar holes were drilled into the top of the rig and thermocouples were placed such that they could record the midflow exit temperature of the air. These TCs were also secured using the furnace paste. The following two figures display this setup.

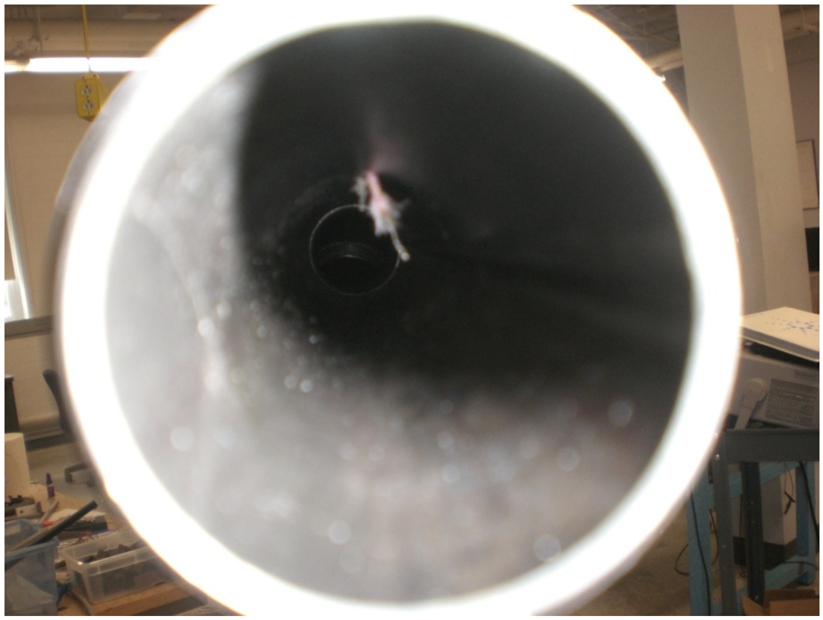


Figure 11 : Midflow Thermocouples Located Near Pipe Exit

Figure 12 : Upper Midflow Thermocouples Secured via Furnace Paste

This arrangement of thermocouples allowed for data to be gathered in several key locations: the fire's temperature (outer TCs), the approximate inner surface temperature of the rig, and the entrance and exit air flow temperatures. With these temperatures known, the heat transferred to the air from the fire could be calculated.

Since only one C pipe was fabricated, a propane torch was used as a heat source instead of burning logs when running tests. The C pipe was constrained by using a table vice. The overall setup can be seen in Figure 13.

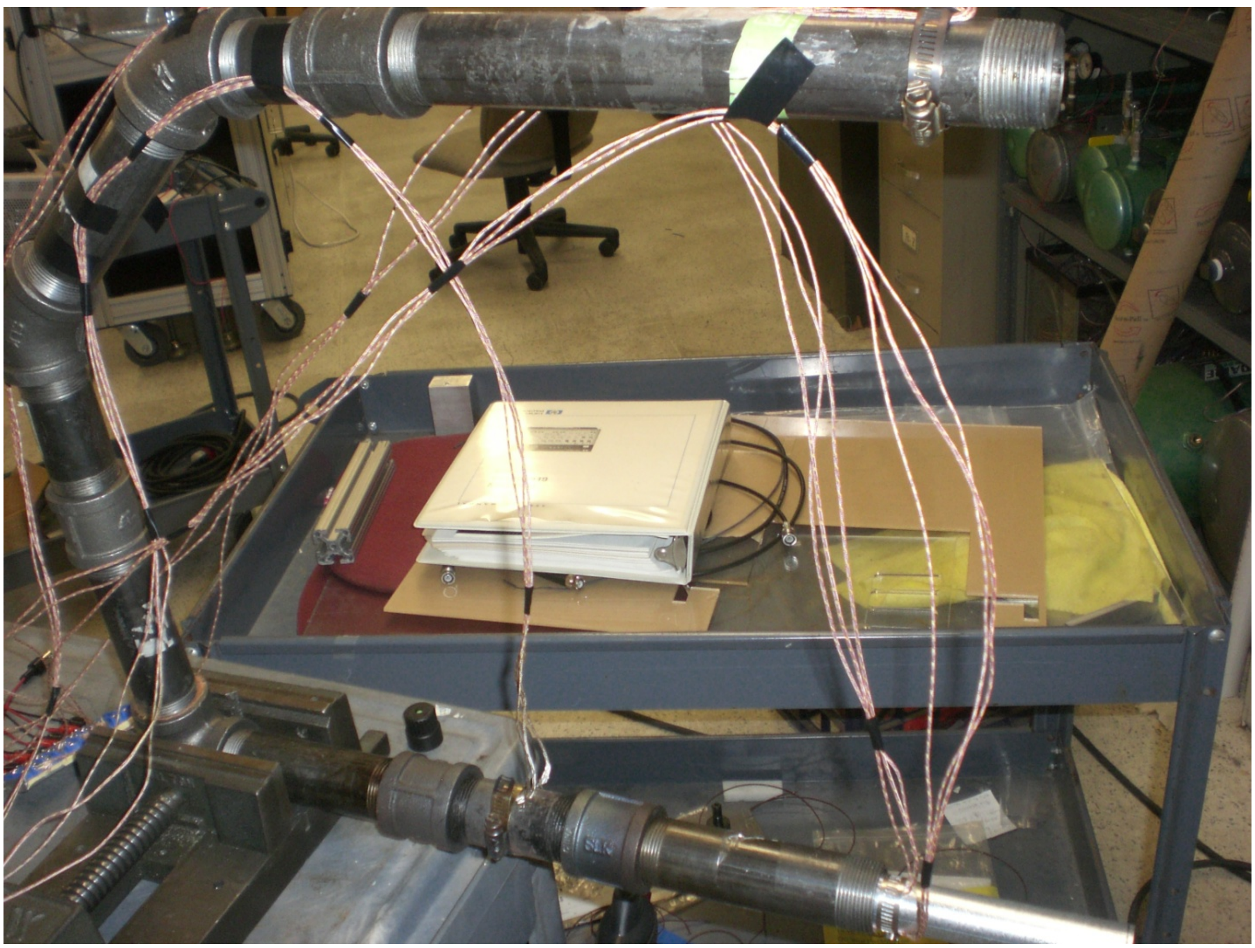


Figure 13 : Entire Test Rig

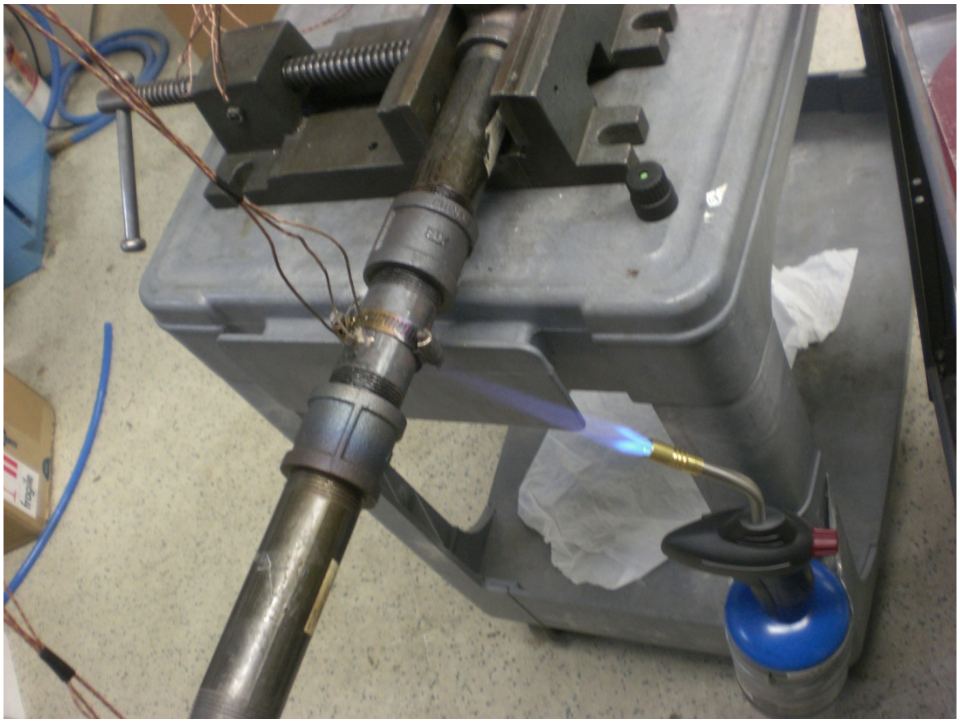


Figure 14 : Testing Configuration

Data was gathered using a National Instruments NI USB-6229 Data Acquisition Board (DAQ). This board can be seen in Figure 15.

Figure 15 : Data Acquisition Board Used During Testing

This board is capable of reading 16 independent sources of data at 16 bit resolution and 250,000 samples per second. The thermocouples were fed into this board using BNC cables.

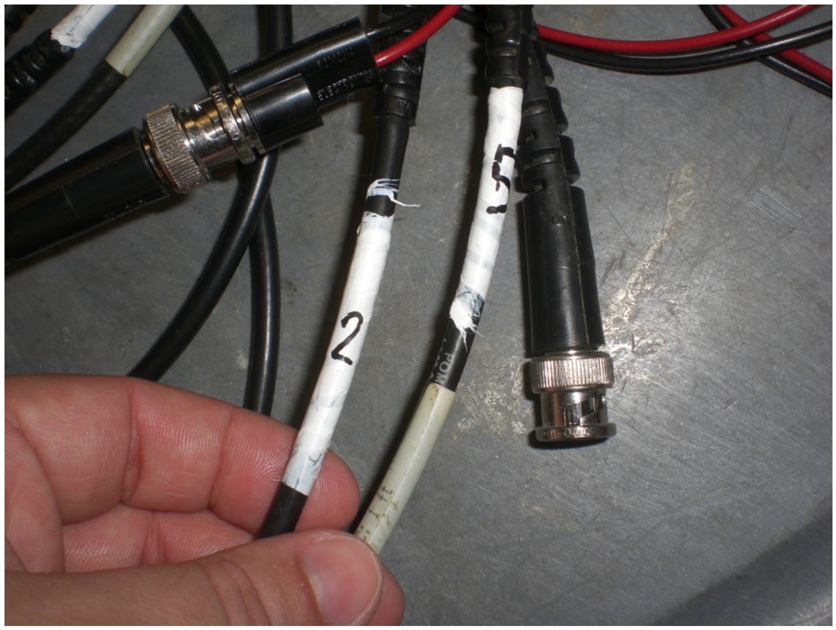


Figure 16 : Labeled BNC Cables Attached to DAQ Board

These cables were labeled from 1-12 according to which DAQ Board channel they used during testing as determined by the VI's programming. These BNC cables were attached to the thermocouples via screw terminals and alligator clips as seen in Figure 17.

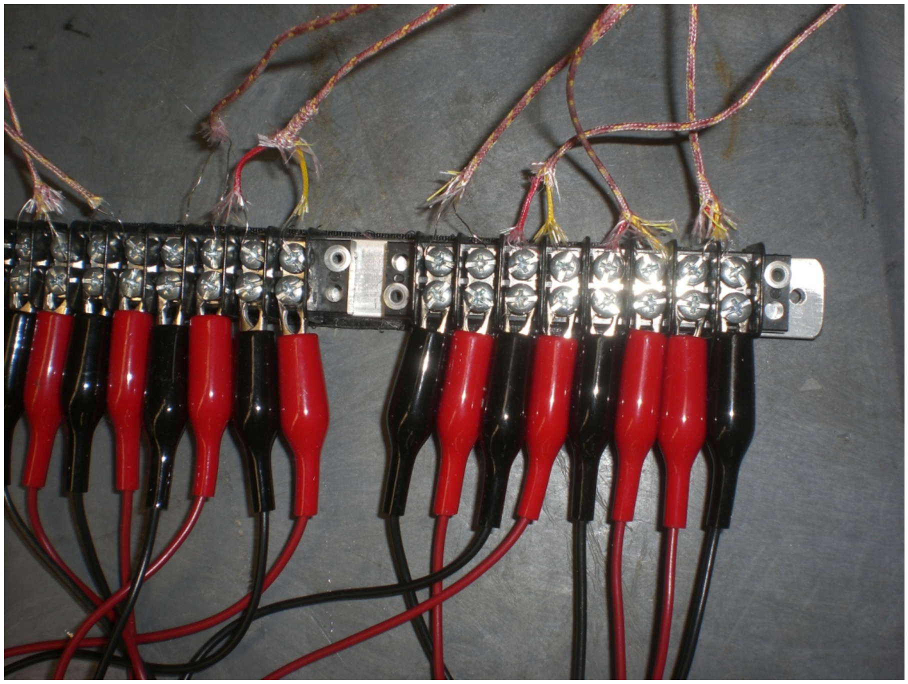


Figure 17 : Screw Terminal Thermocouple Wiring

A virtual instrument (VI) was created using Lab View 8.6 to read and record the TC readings to a Microsoft Office Excel 2007 spreadsheet. This VI was also programmed to use the TC readings and user supplied constants to calculate the amount of heat being added to the air flowing through the pipe. A screen shot of the front panel of the VI is shown below in Figure 18.

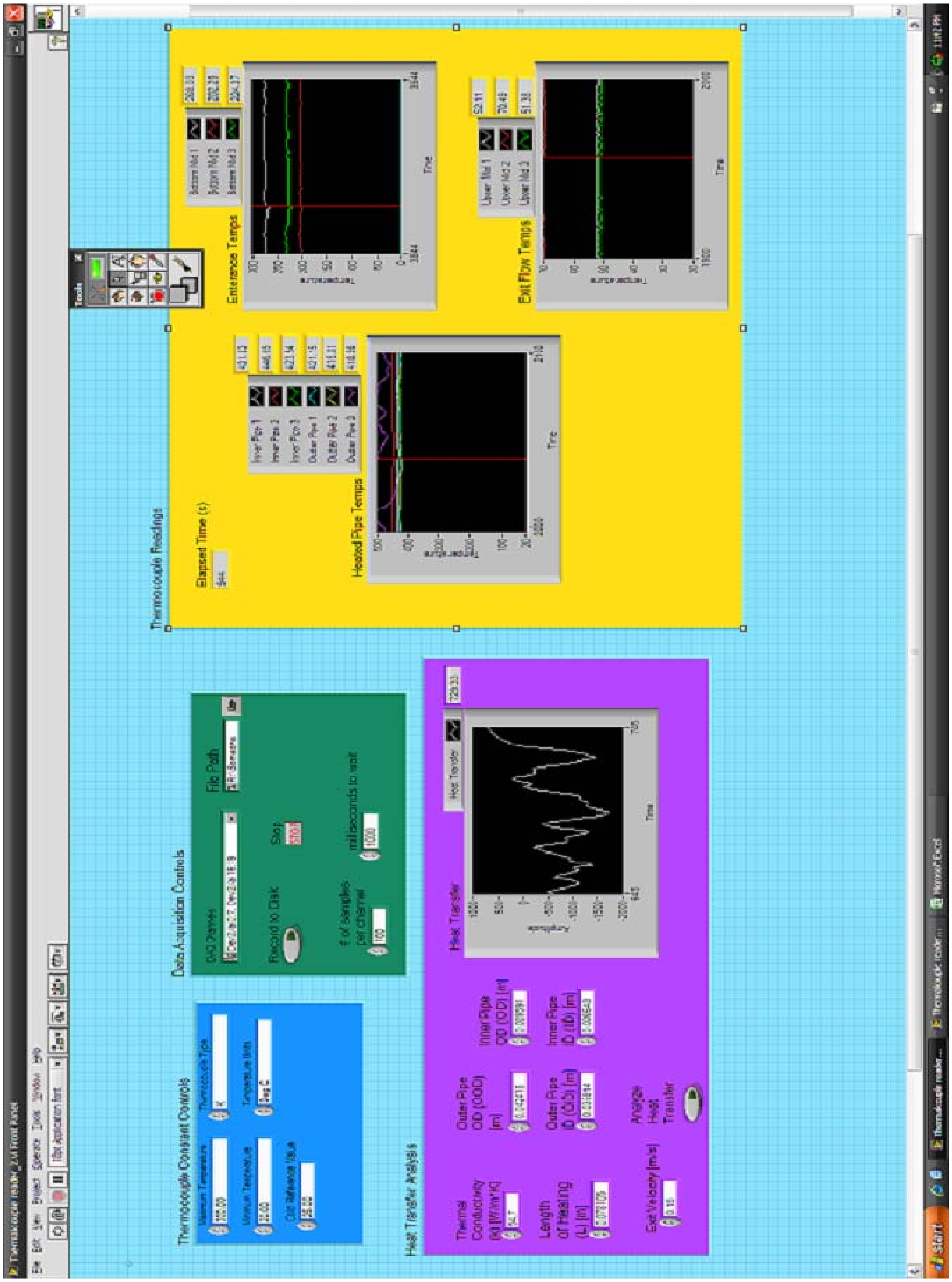


Figure 18 : VI Front Panel

The Thermocouple Constant Controls located in the blue section of the front panel include user input controls such as the minimum and maximum expected temperature values, the cold junction reference, the thermocouple type and the units of the TC readings. The green section contains Data Acquisition Controls such as the record to disk button, the milliseconds to wait button, the STOP button as well as the file path controls. The yellow section displays the TC readings graphically and are grouped together according to the TC's location on the testing rig. For example, the left graph plots the outer TC readings as a function of time as well as the inner TC readings. Finally, the purple section contains the constants and dimensions controls required to calculate the heat added to the air during testing. The accompanying graph plots the calculated heat transfer versus time. This portion of the VI is contained within its own triggered loop so the computer does not waste time and resources computing the heat transfer while the system is still heating to steady state.

The block diagram in Figure 19 shows how the VI was programmed. The instrument is set up to read 12 separate channels from the data acquisition board. These readings are then averaged before being plotted on the front panel. These averages are also fed to the record to disk node which writes the data to an Excel spreadsheet when active. Some of these averages are diverted again to a formula loop where the heat transfer is calculated when active. The results of these calculations are fed into a graph which is displayed on the front panel.

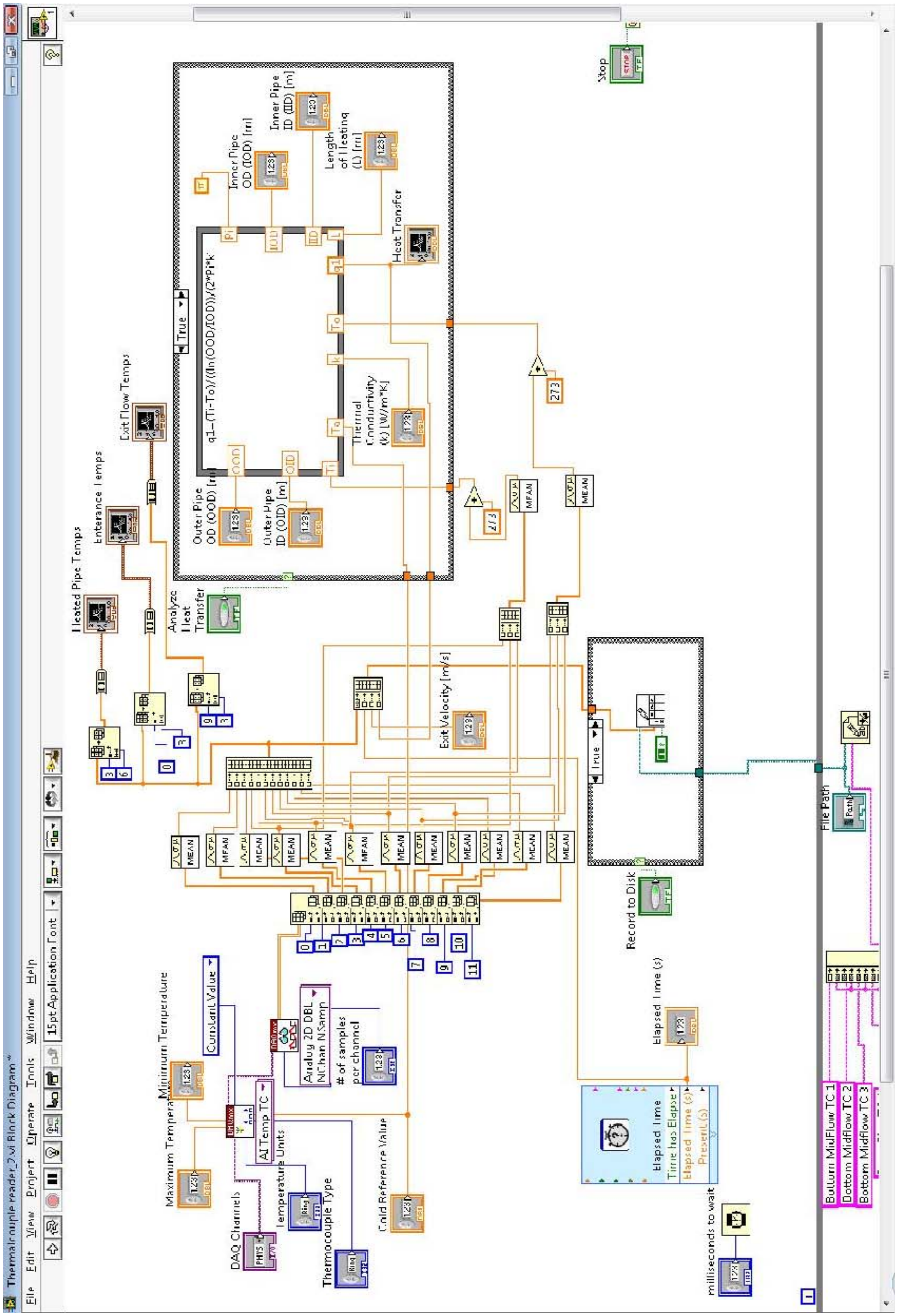


Figure 19 : VI Block Panel

Utilizing this VI, several tests were run at near steady state conditions and TC readings were recorded when the outer TC readings were at 250 °C, 300 °C, and 400 °C. Each test was conducted by aiming the propane torch at the desired heating area and letting the system heat for roughly 20 minutes until the outer thermocouples read a constant temperature of 250°C, 300°C and 400°C. This constant temperature reading ensured that the system was near steady state conditions. This is important because the heat transfer principles used to analyze the system are only valid for steady state conditions. Steady state conditions in this test meant the heat added to the system was no longer raising the temperature of the system but rather maintaining the temperature.

Results

The data gathered during tests was analyzed using simplified heat transfer equations as well as graphically. Tests were run for roughly twenty minutes to ensure steady state conditions before data was collected. Once at steady state, data was collected with a sampling rate of one point per second. For each region of the rig the three TC readings were averaged. These averages were plotted against elapsed time to produce graphs summarizing the test conditions. An example of this graph for the 250°C test can be seen in Figure 20.

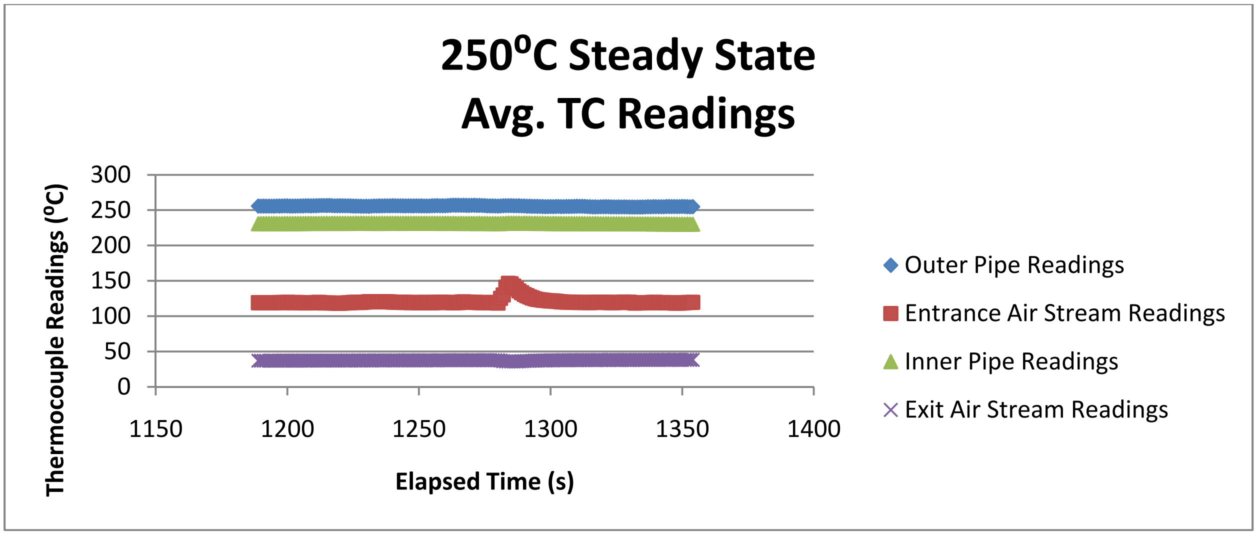


Figure 20 : 250C Test Temperature vs. Time

It should be noted that the x-axis values do not start at zero because the VI had been recording elapsed time since the heat was first applied. Data was not recorded however, until the variance in temperature readings were small enough to assume steady state conditions. The spike in the inlet temperature observed in the middle of Figure 20 was likely due to the air flow out of the piping being temporarily interrupted. With the exit flow blocked the air within the piping would be heated to a higher temperature before dropping back down once flow is restored.

Similar graphs were generated for the subsequent 300 and 400 C steady state tests and can be seen below in Figure 21 and Figure 22. To attain the elevated temperatures from 250 to 300 degrees C, the test rig was simply moved closer to the nozzle of the propane torch. This propane torch was secured such that it aimed at a fixed point throughout the tests. This ensured that the amount of heat added over time was constant. To elevate the temperatures above 300 C however, a second torch was required. This torch was hand held and manually aimed at the rig. Since the second torch was not secured a constant distance away from the piping, the temperatures in the 400 C test were not as steady as previous tests. However, the variations were minor. Data was collected once the readings were semi steady. However, a slight overall increase in the temperatures can be observed when plotting the TC readings as a function of time.

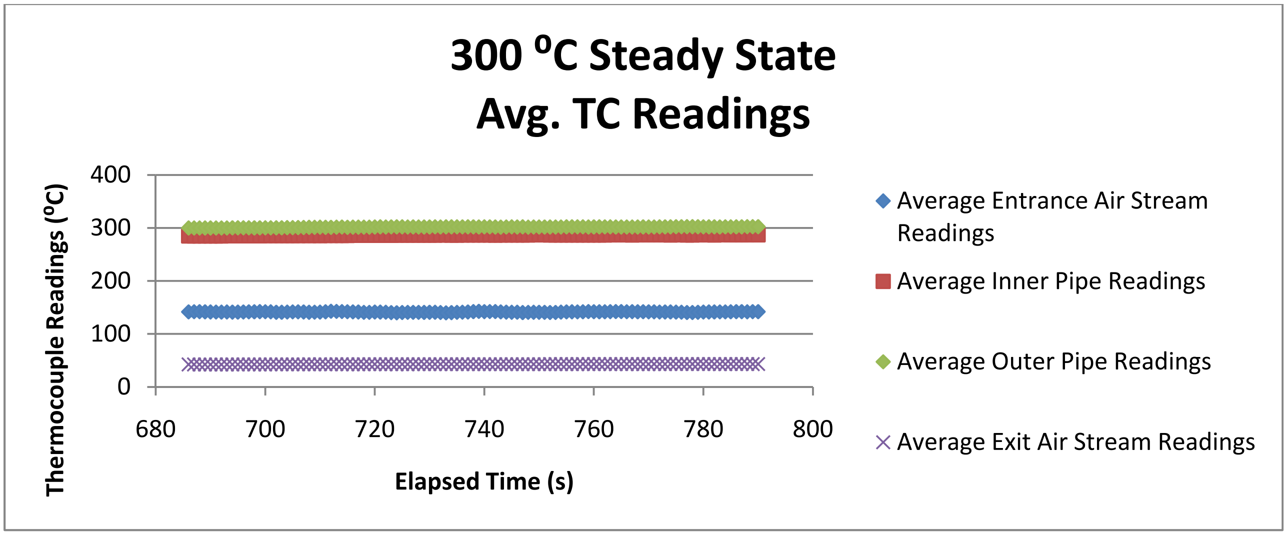


Figure 21 : 300C Test Temperature vs. Time

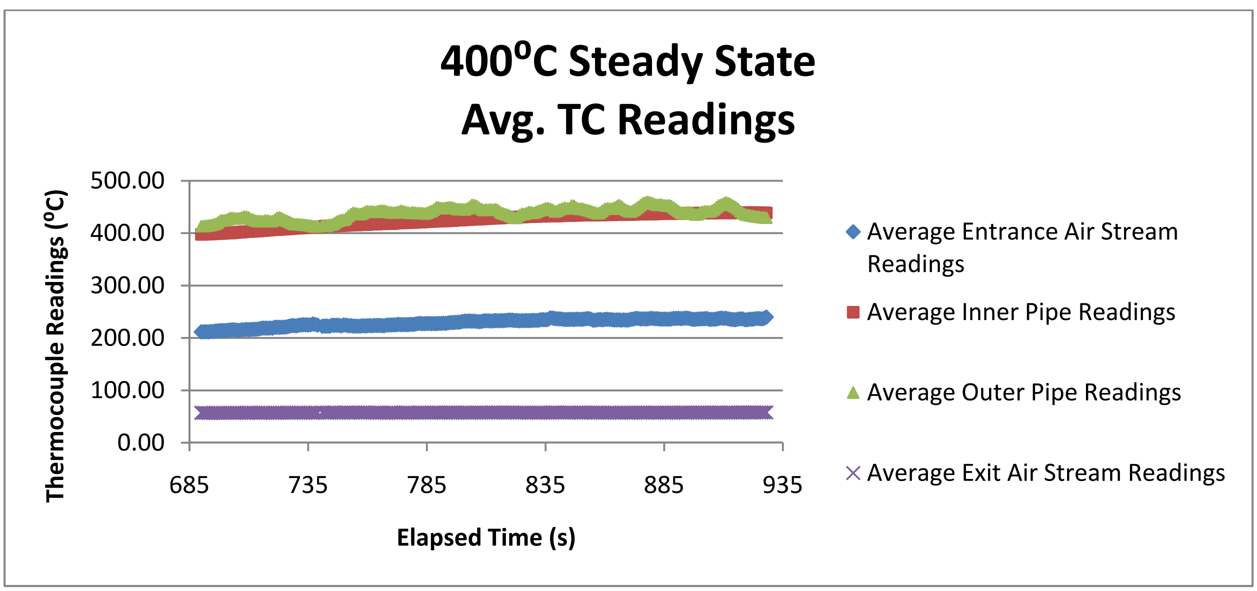


Figure 22 : 400C Test Temperature vs. Time

The VI was programmed to write the data to a spreadsheet where it could be analyzed further within Microsoft Excel 2007.

The amount of heat added to the air traveled through two forms of thermal resistance. The heat from the fire must first travel through the outer piping wall by means of conduction. Then the heat is added to the air gap via convection heat transfer. Finally the heat is applied to the inner thermocouples after being transferred via convection to the surface of the inner piping. Though this air gap was significantly small compared to the inner diameter of the piping, studies have shown that even small air gaps can produce significant drops in temperature. An example is given in a study on the thermal performance of chimneys using different chimney liners. They found that with an air space of only 20 mm, the measured temperature on a surface was almost half as when the gap was not present (Peacock, 1987).

Calculations were performed in MathCAD 14 which included a film coefficient in the resistance path of heat. This analysis assumes a fire temperature of roughly 400°C based on previous calculations as well as previous studies. The vertical and upper sections of piping can be assumed to be roughly 200°C based on tests performed by Peacock. In his tests, Peacock reported temperatures located at chimney's dampers of 188°C (Peacock, 1987). These damper temperatures correspond to a log fire with a fireplace insert. Based on test data gathered in this project, the air temperature within the rig when heated to 400°C was roughly 200°C. If the vertical and upper sections of the test rig used in this project are assumed to have surface temperatures near 200°C, then it can be assumed that the pipes are insulated. This infers little to no heat transfer between the air traveling through the piping and the pipe walls since both items are at similar temperatures. Calculations using these assumptions are as follows.

Equation 8 : Heat Transfer Equations (Kreith & Black, 1980)

$$Q = (UA)\cdot(T_o - T_i)$$

Where U is defined as one over the total thermal resistance of the system. For this setup, the total resistance consisted of a conduction component and a convection component.

$$U = \frac{1}{R_{total}}$$

$$UA = \frac{1}{\dfrac{1}{2\cdot\pi\cdot L\cdot\left(\dfrac{ID}{2}\right)\cdot h} + \dfrac{\ln\left(\dfrac{OD}{ID}\right)}{2\cdot\pi\cdot L\cdot k}}$$

where:
Ti = Inner Temperature
To = Outer Temperature
k = Thermal Conductivity of Piping
L = Length of Heating
OD = Outer Diameter of the Outer Pipe
ID = Inner Diameter of the Inner Pipe
h = Convective Heat Transfer Coefficient

Reference:(Kreith & Black, 1980)

The thermal conductivity of the piping used below is the thermal conductivity of Iron and was obtained in Fundamentals of Thermal-Fluid Sciences properties table.

$k := 54.7 \quad \frac{W}{m\cdot K}$ $\qquad$ $h := 20 \quad \frac{W}{m^2\cdot K}$

$OD := 0.042418 \quad m$

$ID := 0.026543 \quad m$

$L_{tot\_400} := 0.4699 \quad m$

$T_o := 400$ deg C

$T_i := 200$ deg C

$$UA_{400} := \frac{1}{\dfrac{1}{\left[2\cdot\pi\cdot L_{tot\_400}\cdot h\cdot\left(\dfrac{ID}{2}\right)\right]} + \dfrac{\ln\left(\dfrac{OD}{ID}\right)}{(2\cdot\pi\cdot L_{tot\_400}\cdot k)}} = 0.782$$

Equation 9 : Heat Transfer Calculations (Peacock, 1987)

$Q_{400} := UA_{400} \cdot (T_o - T_i) = 156.379$ Watts

The associated mass flow rate for this heat added can be calculated using the subsequent equation relating the mass flow rate to the heat transfer as a function of the temperature difference and the specific heat of the air. The specific heat (c_p) in the equation is found to be 1.026 which is close enough to approximate as 1.

$$Q = c_p \cdot \Delta T \cdot m_{rate}$$

$\Delta T := 200 - 25 = 175$ deg C

$$m_{rate} := \frac{\frac{Q_{400}}{1000}}{1 \cdot \Delta T} = 8.936 \times 10^{-4} \quad \frac{kg}{s}$$

This corresponds to a velocity as a function of the density of the air and the cross sectional area of the flow. The diameter used in the cross sectional area calculation is the outer pipes inner diameter.

$$m_{rate} = vel \cdot A_{cs} \cdot \rho$$

$$A_{cs} := \frac{\pi}{4} \cdot (ID)^2 = 5.533 \times 10^{-4} \quad m^2$$

$$\rho := 0.746 \quad \frac{kg}{m^3}$$

$$vel_{entrance} := \frac{m_{rate}}{A_{cs} \cdot \rho} = 2.165 \quad \frac{m}{s}$$

According to Peacock's report, the maximum safe temperature felt in the room is roughly 65 C above room temperature. If room temperature is assumed to be 25 C within a home, the maximum temperature for the exit flow temperature is about 90 C. The mass flow rate must be equal to that previously calculated since there is nowhere else for the air to flow. Thus we can calculate the heat added to the room from one pipe.

$$Q = c_p \cdot \Delta T \cdot m_{rate}$$

$$Q := 1 \cdot 1000 \cdot \Delta T \cdot m_{rate} = 156.379 \quad W$$

The exit velocity will be slightly decreased because of the transition from a smaller diameter entrance to a larger exit diameter.

$$A_{cs\_exit} := \frac{\pi}{4} \cdot (0.035814)^2 = 1.007 \times 10^{-3} \quad m^2$$

$$vel_{exit} := \frac{m_{rate}}{A_{cs\_exit} \cdot \rho} = 1.189$$

The proposed design included four identical pipe configurations used in parallel. Thus the results can be multiplied by 4 to get an estimate on the total amount of heat added when using the proposed fireplace insert design.

$$Q_{total} := Q \cdot 4 = 625.516 \quad W$$

According to simplified heat transfer calculations applied to the results of testing performed, it was discerned that a four pipe fire grate would add roughly 625 Watts of heat to the adjacent room. The exit velocity of each pipe would be roughly 1 m/s for each pipe. The assumption of no insulated piping can pose a safety concern however, because the exit temperature of the pipes with the corresponding flow rate are around 200°C which is well above the maximum safety of 90°C.

Conclusions

Conventional wood burning fireplaces are a common commodity in residential American homes yet are inefficient space heaters. Many fireplace inserts have been designed and are available commercially to mitigate this inefficiency. These inserts claim various levels of increased heat output but are not backed by quantifiable data that is publicly available. The purpose of this project was to provide quantitative analysis of fireplace systems and the effects inserts have on their thermal efficiency. A fireplace grate was designed based on similar existing designs. This design was built, instrumented, and tested several times. Temperature data was collected and analyzed using simplified heat transfer equations. This allowed for the amount of heat added to the room from the insert to be estimated.

The results of these calculations yielded a gain of roughly 625 Watts of heat added to the room. When compared to the assumed amount of heat felt by a person standing in front of the fire (1500 Watts), the fireplace insert is estimated to add 42% of the heat added by a fireplace alone. In order to improve on the safety of the design, a blower should be considered to increase the flow rate of the system. This increase in flow rate will correspond to lower exit air temperatures. Based on this data, it appears that a fireplace insert similar to that which was used in this project which includes a blower would be a good addition to every residential fireplace system.

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