Economic and Environmental Sustainability of Using Bio-Fuels for Small Nebraska Greenhouses

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Economic and Environmental Sustainability of using Bio-fuels for Heating Small Nebraska Greenhouses

By

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Economic and Environmental Sustainability of Small Nebraska Greenhouses using Bio-fuels for Heating

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The primary goal of this paper was to increase profitability in Nebraska greenhouses by using biomass fuels for heating instead of propane. Several different fuels were tested, including whole shelled corn, dry distiller's grains pelletized, wood pellets and blends between each biomass. The main fuel focus was on whole shelled corn. Bomb calorimetry tests were performed on biomass fuels and their respective ashes. Several furnace and heat exchanger efficiency tests were performed, with cost effectiveness analysis for each fuel type. Emissions data was also collected for each fuel on carbon monoxide, carbon dioxide, nitrous oxides, sulfuric oxides, and particulate matter. The project used a biomass furnace donated to a greenhouse at Firth, Nebraska and an existing propane furnace. Although the biomass furnace generally had a lower efficiency than the 81 percent advertised for the propane furnace, the biomass fuels were more cost effective than propane. The biomass efficiencies typically ranged between 50 and 80 percent. Over a four year period (2008-2011) the cost savings of biomass fuels ranged between 30 and 60 percent and totaled a little over $15,000. Overall, biomass furnaces show great potential to be utilized in Nebraska greenhouses.

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Chapter 1 Introduction and Background

Nebraska, the heartland of America, is well known nationally and internationally for its agricultural field crop production during spring, summer, and fall. This is due to abundant sunshine, warm temperatures and plentiful moisture. However, what is not well known is that some of the sunniest days of the year occur during the winter months. Nebraska is fortunate and has been reported to receive excellent average incident levels of solar radiation of 1000 to 1600 Btu/ft^2 per day (12 to18 MJ/m^2) during the winter months (Bodman et al. 1989). Utilization of solar energy for controlled environment agriculture (CEA) has yet to be exploited and turned into multiple food products in Nebraska. With uncertain transportation costs, concerns about imported food safety, human health/obesity issues, and the need to improve local economies, increasing local production of fresh fruits and vegetables would be a logical step for Nebraska CEA.

Additionally, growing food under protection would allow Nebraska farmers an additional source of income apart from the usual field season. Research by (Hoagland et al. 2008) found that the average corn/soybean farmer has labor and/or time available from December through March which could be utilized to grow alternate crops. Nebraska currently has approximately 360 commercial growers and 2.5 million square feet of production area under glass or other protection. While greenhouse crop production is not a major industry in Nebraska, a potential for economic expansion does exist. Efforts are underway to determine the viability of winter-time grown strawberries (Paparozzi et al. 2010). Tomato house enterprises have been attempted or are underway in Nebraska. A major limitation is that 60-80 percent of production costs are associated with energy input, the cost of which continues to rise.

Sustainability is a primary concern for engineering design of alternative energy systems. According to the Annual Energy Outlook of 2010 (Figure 1.1), petroleum products currently account for about 40% of energy consumed in the United States.

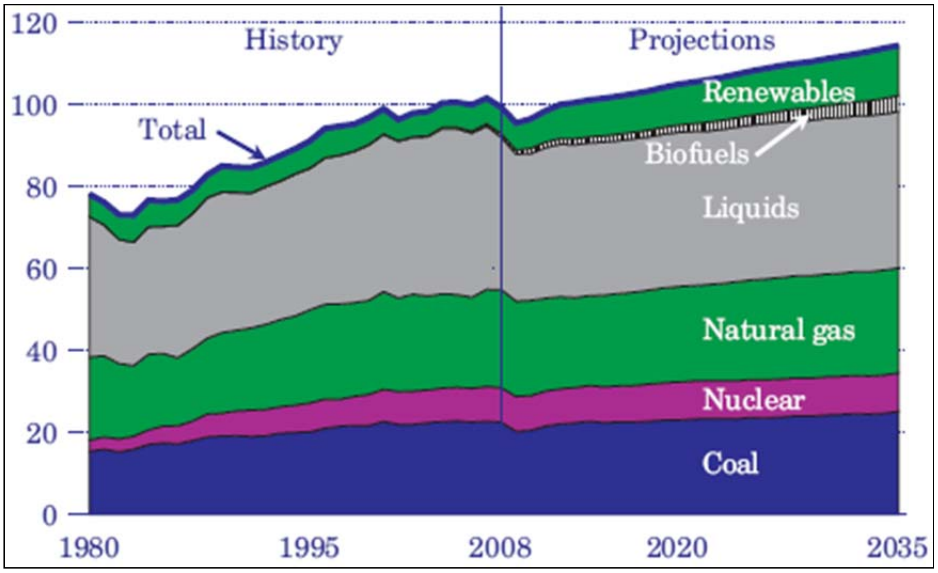


Figure 1.1. U.S. primary energy consumption, 1980-2035 (quadrillion Btu) (DOE EIA-0383 2010 pg. 2).

Figure 1.2 shows that approximately 56% of petroleum is imported. Increasing worldwide demand for petroleum has further limited supplies (EIA, 2010). Renewable resources are still a work in progress. While significant strides have been made, data from the 2010 report suggests that renewable energy accounts for less than 10% of total energy consumed (EIA, 2010).

There are several different forms of renewable energy available in the Midwest, including wind, solar and bio-fuels. While renewable energy is available, it often needs to be converted or stored prior to use.

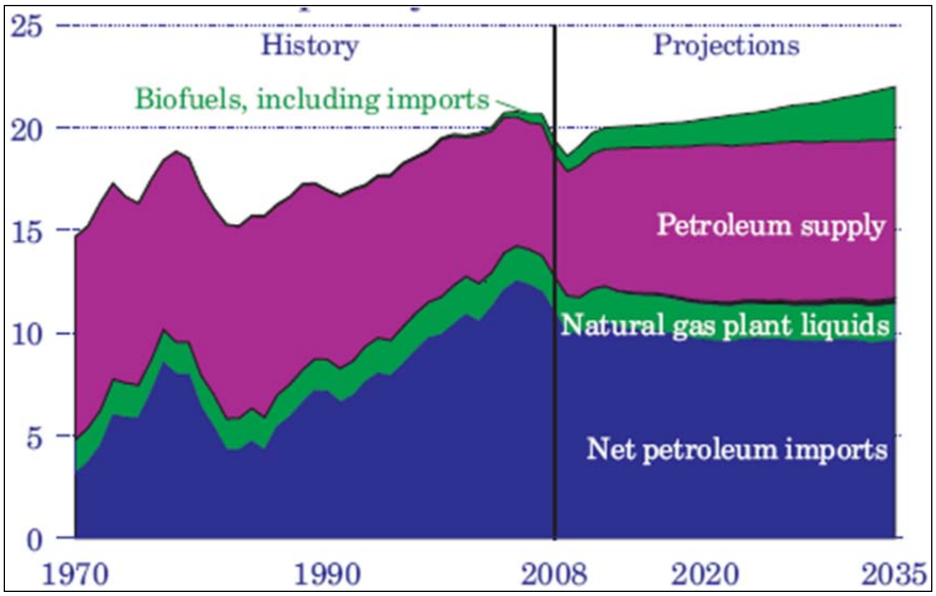


Figure 1.2. U.S. liquid fuels supply, 1970-2035(million barrels per day) (DOE/EIA-0383 pg. 3).

Attempting to quantify the sustainability of alternative energy is a major challenge. Two main methods were selected for this analysis: fuel combustion efficiency and pollutant emissions. Fuel combustion efficiency was chosen because characterizing fuel heat content and combustion efficiency is necessary to compare fuel types, fuel cost and payback period for greenhouse crop production. The second method selected to help determine sustainability was quantifying pollutant emissions. Exact emissions from most biomass combustion are uncertain and site dependent; characterizing these emissions results served to provide a clearer picture on environmental impact.

Background

Greenhouse Heating Requirements

A greenhouse requires sufficient heat energy rates to offset the worst case scenario for wintertime or nighttime heat losses while maintaining a steady air temperature (ASAE EP 406.3). Greenhouse heat loss is based on the thermal resistance properties of the glass glazing and the sidewall perimeter heat loss; the architectural design is also important. Thus, greenhouses with high glazing surface to floor areas generally have higher overall heat loss. With the ability of a greenhouse to trap solar gain during the day, most heating demands occur at night. Therefore, a worst case scenario can be related the lowest probable outside air temperatures during the night. Table 1.1 shows the average weather conditions in Nebraska for each month. The lowest average temperatures occur at night during the months of December, January and February. A greenhouse furnace needs to be sized to match nighttime heat loss accounting for the overall size of the greenhouse.

Table 1.1. Typical Lincoln, NE Weather

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Avg. High °F	32°	37°	50°	64°	74°	84°	90°	86°	77°	66°	50°	35°
Avg. Low °F	10°	15°	26°	38°	50°	60°	66°	64°	54°	40°	27°	15°
Mean °F	21°	27°	38°	52°	62°	74°	78°	75°	65°	54°	38°	26°
Avg. Precip.	0.5 in	0.7 in	2.1 in	2.8 in	3.9 in	3.9 in	3.3 in	3.4 in	3.5 in	2.1 in	1.3 in	0.9 in

Source: http://Countrystudies.us Lincoln/NE.

Determining greenhouse heating needs is a simple general calculation using the equation given as:

$$Q\left(\frac{\mathrm{Btu}}{\mathrm{ft}^2}\right) = (\mathrm{Tinside} - \mathrm{Toutside})(°\mathrm{C}) * \mathrm{Asurface}\ (\mathrm{ft}^2) * \mathrm{Ufactor}\left(\frac{\mathrm{Btu}}{\mathrm{hr\ ctot}^2}\right) \ldots\ldots\ldots\ldots(1)$$

Let's assume that the total exposed area of the roof, sides and ends of a greenhouse is 10,000 ft^2. Let's also assume that we want the capability of maintaining 60 °F with 0°F outside and U factor for glass of 1.13. This results in a heat loss of 678,000 Btu/hour. This loss can be satisfied by two 400,000 Btu/hour heaters or four 200,000 Btu/hour heaters, assuming an 85% heater efficiency (Ball RedBook, 1991).

Most small Nebraska greenhouses will be of the Quonset, double polyethylene greenhouse design. An example of a single span polyethylene Quonset greenhouse can be seen in Figure 1.3. Such greenhouses will be constructed with a light frame and 6-mil clear polyethyelne glazing material that is much cheaper than glass. According to Ball Redbook, polyethylene greenhouses are inexpensive and easy to build. Another reason small Nebraska growers use polyethylene glazing material is that glass greenhouses are far more susceptible to hail damage.

Typical crops grown in a small greenhouse environment are seasonal potted plants for retail sale or home use. These can range from vegetable plants like tomatoes or peppers to flowering plants for home decoration. Typically, crop selection falls under the grower's discretion. This may be based upon market value for the various plants, or personal preference of the grower and knowledge of crop culture.

Figure 1.3. Quonset Style Greenhouse at Firth NE (West Side).

Greenhouse moisture, which attracts disease and insects, is also a major concern. Accounting for this is one of the grower's main tasks. Humid environments tend to provide ideal conditions for disease germination. Thus, control or removal of moisture can also help to reduce the spread of disease in a greenhouse. It is recommended to ventilate a greenhouse once every hour in order to exchange overly moist air for dryer air (ASAE, EP 406.3).

This ventilation also helps to replace lost carbon dioxide from plant uptake. To control disease and insects, the Ball Redbook recommends, "Before planting, the greenhouse should be clean and free from weeds, pests and diseased plant material. If the house was used previously, the entire greenhouse should be sterilized. Steam sterilization should be used as a priority treatment. Any debris such as dead plant material, especially under raised benches, should be removed."

The two primary controls for maintaining greenhouse environments are typically furnace and ventilation controls. These environmental controls usually operate using a thermostat, analog or computer system (Ball Redbook 1991). Most small greenhouses operate using a thermostat controlled atmosphere because of the easiness of operation and installation. The thermostat system is generally the least expensive, and operates simply as an on/off control. Figure 1.4 shows an example of the operation of an on/off furnace system. The cycling effect noticed during the nighttime periods indicates a switching back and forth between furnace heating and passive heat loss. Cycling is greater when the furnace is oversized for the current heat loss, which leads to an increased loss of heat and lower greenhouse efficiency. The on/off pulsing causes a reduction in efficiency due to switching between states. This can be even worse during cold daytime conditions, when there is insufficient solar gain. Essentially the system turns on the ventilation and replaces excess hot air with cold air just to have the furnace reheat the cold air. Excess cycling may also increase humidity levels in the greenhouse. High humidity levels during early morning hours may result in condensation rain off the inside of the glazing onto the crop. As previously discussed, wet leaves are magnets for disease and pests.

Greenhouse Heating Systems

There are two main types of greenhouse heating systems according to Ball Redbook: root zone, or ground heating systems, and overhead unit air heaters. Hot water or steam is distributed by pipes in the former system, while polyethylene fan ducts and air jets are used for distributing warm air in the latter system. A root zone water heating system is shown in Figure 1.5. This system can work effectively to reduce the cost of fuel used because the inside temperature of the greenhouse does not need to be as high as it does with systems that utilize an overhead heater. For instance, a typical overhead, propane-fired unit heater in a greenhouse provides an overall inside temperature around 80°F, while a root zone water boiler system only needs to maintain the thermal environment of the bottom layer, bench, or soil bed of the greenhouse. The upper air of the greenhouse using a root zone system may be closer to 60°F, which can save almost half of the fuel cost (Ball Redbook 1991).

On the other hand, there are distinct advantages to overhead, unit air heater systems. According to Ball Redbook, these systems include a lower initial cost, are more flexible, provide potential fan jets, and are, overall, a more reliable heating system. These advantages become even more profound for a small greenhouse grower who may not be able to initially afford the extra costs of laying water pipes or installing a secondary furnace. An overhead unit in a greenhouse can be seen in Figure 1.6. Often, these units have fan jets attached to them with tubing spanning the length of the greenhouse. Albright (1990) explains the calculation of air jets and their velocities. Air jet holes are evenly spaced along the distribution tubing to provide an even heating environment in the greenhouse. Fan jets can also be directed down to heat the crop surface to reduce outside heat loss.

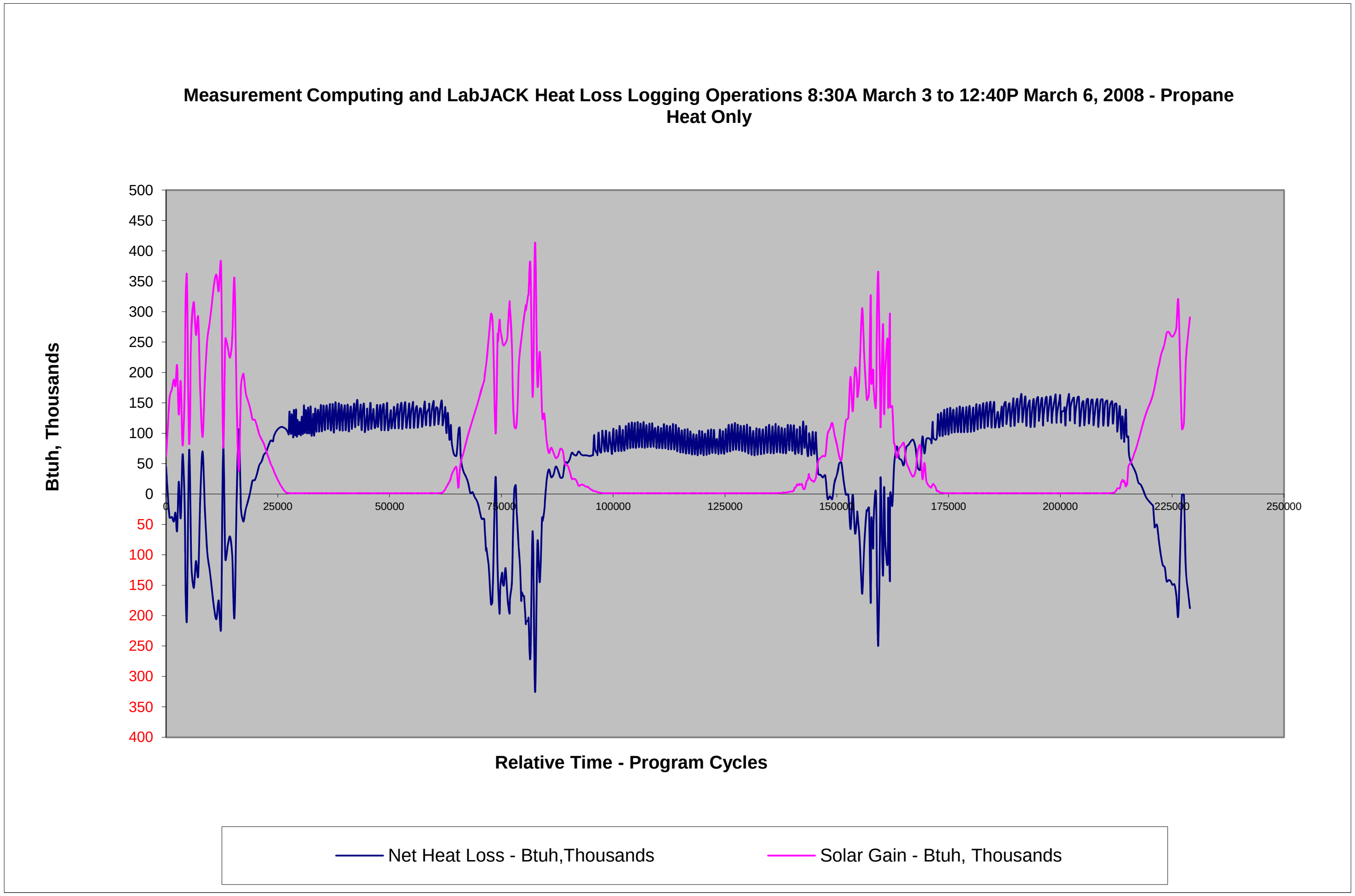


Figure 1.4. An example of On/Off Heating Controls (Meyer, et al, 2009).

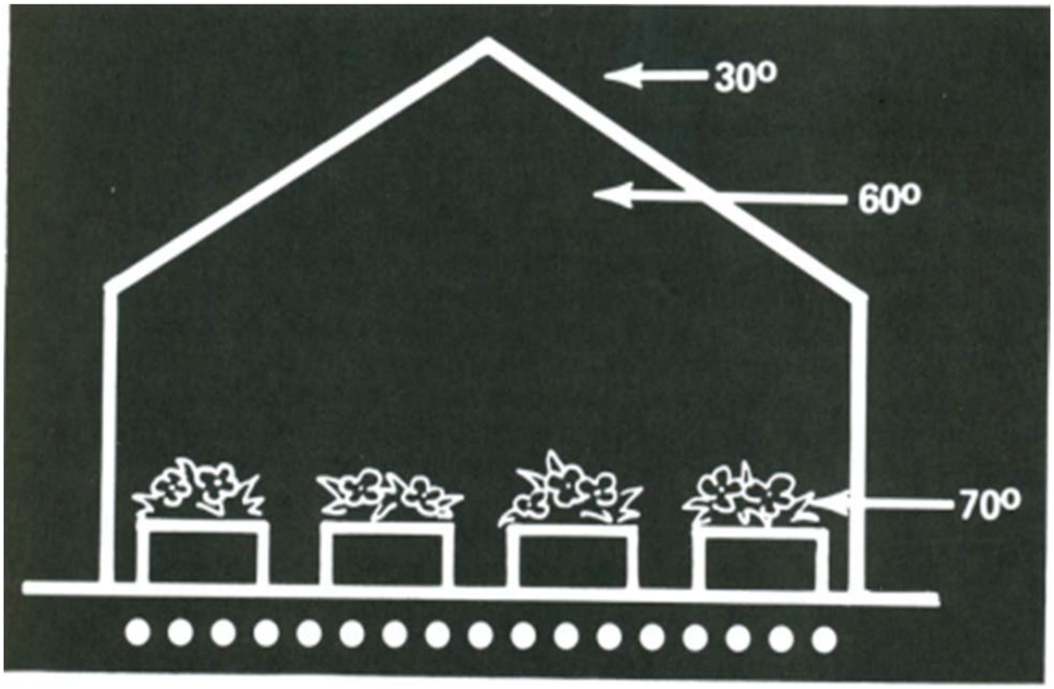


Figure 1.5. Sample Root Zone Heating System (Ball Redbook 1991 pg. 21).

Figure 1.6. Overhead Modine Heating Unit. Windmaster Ventilation Fans are in the background.

Greenhouse Fuels

The most common fuel used in small Nebraska greenhouses is propane, a fuel that is readily available and easy to integrate into a greenhouse. Propane (C_3H_8) is a hydrocarbon fuel with an average energy content of 91,500 Btu/gallon (Cengel and Boles 2006). Propane is primarily produced as a by-product from the natural gas and petroleum industries. A sample propane tank connected to a greenhouse can be seen in Figure 1.7.

Figure 1.7. Propane Tank for supplying fuel to the Greenhouse.

Bio-fuels have become a major area of research in renewable energy, specifically in heating and energy production. The term bio-fuel may refer to anything from ethanol to wooden logs. Bio-fuels can also include several products grown specifically for energy usage, like switchgrass, or waste by-products of other industries, such as sawdust. Bio-fuels have been used by humans for thousands of years; however, increasing amounts of global gas emissions have led to new methods of utilizing these fuels. Many agricultural products are being converted to liquid fuel sources like ethanol and biodiesel. Through the degradation of the raw material, overall conversion efficiency can be lost by the continual refinement process. Some studies have looked into direct combustion of agricultural products. Another option is to use residual biomaterials for direct combustion from processing. Combustion of residual materials offers great promise, but several problems exist with this approach. Such problems include variable moisture content, bulk density, ash content, volatile matter, variable ignition temperatures, and pollutant emissions. When selecting a bio-fuel, all of these factors need to be considered.

The moisture content of the material is important because “high moisture content can lead to poor ignition, reduce the combustion temperature, which in turn hinders the combustion of the reaction products and consequently affects the quality of combustion” (Werther, et al. 2000). High moisture content can also delay the release of volatile material. More flue gas is released in combustion of high moisture materials, requiring additional equipment for flue gas treatment. Not all biomass materials need to be dried subsequent to processing. Products like coffee are dried during the coffee berry extraction process to a moisture content of about 12% dry basis (Werther, et al. 2000).

The bulk densities of most agricultural products are much lower than brown coal (560-600 kg/m^3) and bituminous coal (800-900 kg/m^3). “The low bulk densities of residues complicate processing, transportation, storage and firing” (Werther, et al. 2000).

Increased densities are required for most agricultural residues to automate loading mechanisms and provide adequate storage on site. Advantages to densification include "the rate of combustion can be comparable to those of coal, burning in grate-fired boilers is possible, uniform combustion can be achieved, particulate emissions can be reduced, the possibility of spontaneous combustion in storage is reduced, and transportation, storage and feeding is made more efficient" (Werther, et al. 2000).

The three main forms of densification include baling, briquetting and pelleting, each of which is progressively more complicated and expensive. Comparing baled straw to briquetted straw yields the following results. Baled straw has a bulk density of 70-90 kg/m^3 and 260-360 kWh/M^3 heating value. Briquetted straw has a bulk density of 450-650 kg/m^3 and 1800-2800 kWh/m^3 heating value (Werther, et al. 2000). A major issue with proceeding from baling to briquetting is the potential need for the addition of a binding agent. Straw will not easily bind to itself during the normal briquetting procedure and the addition of a binder such as sawdust, bark, or corn stalk can help to create a better straw briquette. The main conclusion in the area of densification is, "In order to maintain low fuel costs of the residues, it is more economical to use it close to the point of generation with only limited transportation and storage costs involved. In such cases, densification would only be required if it will enable easier feeding and a more efficient combustion process" (Werther, et al. 2000).

Large ash contents have an impact on the burn and melting temperatures of biomass. Ash causes these temperatures to be lower and variable in values. Some products contain low amounts of ash, allowing them to be burned in a number of existing furnaces. However, for products with high ash contents like oat or barley, new equipment would need to be designed specifically for each product. Biomass usually contains greater amounts of Potassium Oxide (K_2O) ash than fossil fuels which can lead to several problems such as agglomeration, fouling, slagging, and corrosion. “The inter-related events through which single particles of solid fuel undergo during combustion are heating up, drying, devolatilization and finally the combustion of the volatiles and char. The temperature at which devolatilization and char combustion start, the influence of drying on the devolatilization process, the composition of the devolatilization products and the effect of volatile release and combustion on the overall combustion process, are all important information required to understand the combustion characteristics of agricultural residues (Werther, et al. 2000).” High volatile amounts of biomass can impact the combustion process. The devolatilization process occurs at low temperatures which can cause the biomass to ignite immediately for dry materials. The volatile characteristics of biomass need to be taken into consideration during the design of the fuel feeding system, furnace configuration and distribution of combustion air.

Agglomeration occurs in biomass combustion when the fuel melts and adheres to the fluidized bed (Werther, et al. 2000). A couple solutions to agglomeration include using a different bed material or blending the biomass fuel with fossil fuels. Quartz sand is typically used in fluidized beds. Some alternatives are feldspar, dolomite, magnesite, ferric oxide, and alumina. Blending coal with biomass can reduce the K_2O ash content significantly.

Fouling is characterized by deposits on surfaces in the heat recovery section of furnaces, and results in a reduction of heat transfer rates and increased corrosion. Slagging refers to depositions on furnace walls or other surfaces exposed to radiant heat. Both of these can lead to problems such as clogging and variable heat patterns in burners. Corrosion of the furnace metals can occur with the presence of certain chemical constituents in the ash. When a large amount of silicates are present during burning, metals can be corroded because the layers of metal oxides become soluble in silicate slag. The main solution to the problems of fouling, slagging, and corrosion is to use additives. Some potential additives include alumina, calcium oxide, magnesium oxide, dolomite, and kaolin. Additives can increase the fusion temperature of residues, increase the softening temperatures of ash, and enrich the ash formed during combustion with non-potassium/sodium compounds.

"The low melting points of the ashes formed by the combustion of some agricultural residues pose serious design and operation problems. A careful analysis of the melting properties of the ash should therefore be the first step in choosing the combustion system and combustion conditions of a given agricultural residue (Werther, et al. 2000)." Depending on the biomass material, specific design may be necessary. However, the inclusion of an additive can reduce the need for specific design.

The ignition temperatures of different biomasses are not often known. The temperatures can vary due to the issues presented above, including moisture content, bulk density, ash content, volatile matter, variable ignition temperatures, and pollutant emissions. Explosion Investigation and Analysis 1990 by Patrick Kennedy and John Kennedy has ignition temperatures for a variety of grain dusts listed in Table 1.2. This data was used for predicting explosion hazards in grain storages facilities. The dusts should have a slightly lower ignition temperature than the direct fuel itself due to increased air flow rate through the material and the larger surface area available for heating. "The ignition process of biomass is similar to that for coal except there will be more volatile matter available for the combustion reaction. It is, therefore, more likely that homogenous ignition will occur for biomass fuels" (Sami et al 2001).

A comparison of direct combustion of shelled corn to that of corn converted to ethanol was presented by Trier, et al. (2006). Some advantages to shelled corn as a biomass material are its availability (especially in Nebraska/Iowa), high net energy content, and low amount of ash. The conclusions of Trier's study suggest that direct combustion has more promise than ethanol conversion. "While the conversion of shelled corn to ethanol has been a growing industry, only

Table 1.2. Explosive Properties of Agricultural Dusts ***Source: Explosion Investigation and Analysis 1990.***

Type of Dust	Ignition temperature of cloud degrees F	Minimum ignition energy joules	Minimum explosive concentration oz./cu. Ft.	Maximum explosion pressure, psig	Maximum rate of pressure rise,	Relative explosion hazard
Alfalfa	860	0.32	0.1	66	1100	Weak
Cocoa	788	0.1	0.045	65	1200	Moderate
Corn	752	0.04	0.045	95	6000	Strong
Corn cob	752	0.04	0.03	110	5000	Severe
Cornstarch	716	0.02	0.04	115	9000	Severe
Cotton linters	968	1920	0.5	48	150	Weak
Cottonseed	878	0.06	0.05	104	3000	Strong
Grain, mixed	806	0.03	0.055	115	5500	Strong
Rice	824	0.04	0.045	93	3600	Strong
Sugar	662	0.03	0.035	91	5000	Severe
Tobacco	788	--	--	7	200	--
Wheat	896	0.06	0.055	103	3600	Strong
Wheat Flour	716	0.05	0.05	95	3700	Strong

33% of the available energy in the corn is actually captured for use" (Trier, et al. 2006). This does not take into account the energy needed to transport the ethanol to distribution facilities. Comparatively, direct combustion could capture up to 70% of the lost energy transportation energy while only requiring the transport of corn to furnaces. The corn is already being transported to ethanol facilities; therefore, choosing a different final destination should not increase the transportation costs significantly. "If 1.8 billion bushels of exported corn were directly combusted in the US, an estimated 4.1 billion gallons of fuel oil per year could be conserved…or 6.6% of the current distillate oil usage in the US. Also if a manufacturer requiring 1.169 MBtu/hr for half the year and currently using propane as an energy source could save $54,490 by burning shelled corn as an alternative fuel. A prototype atmospheric fluidized bed AFB at Ohio Agricultural Research and Development Center (OARDC) can supply 150,000 Btu/hr and would give a cost savings of $6,811 per year if shelled corn replaced propane. The estimated cost for manufacturing this unit is under $10,000 which suggests a payback period in less than two years" (Trier, et al. 2006).

It is also important to look at potential drawbacks on the other side of the corn energy issue. In the case of the corn burner, the simple payback period does not take into account storage or variable shelled corn prices. Storage is necessary since it would be difficult to receive periodic shelled corn shipments year round without some significant changes in price. This issue could be solved by installing a storage facility like a small corn silo, but would increase the payback period of the whole system. Another issue to consider is that direct combustion of corn relates only to heating purposes, while ethanol covers a larger variety of uses: transportation, heating, et cetera. The intended use of the fuel always needs to be taken into account.

To compare energy values, whole shelled corn costs roughly $10.35 per MBTU's while propane costs $20.07 per MBTU's at the current market prices of $4.04/bushel and $1.84/gallon, as shown in Appendix B. Some of this cost advantage can be reduced through efficiency losses. Increasing the process efficiency for bio-fuel usage could allow these materials to be even more cost competitive and sustainable.

There are several factors to account for when using biomass as a fuel source. Many materials can be interchangeable, but need to be evaluated before use. Interchangeable materials could include different types of pelletized biomass or other materials with similar properties. This becomes a major issue because one potential biomass material is not enough to replace fossil fuels. Also, depending on which biomass material is selected, usage practices may be necessary to increase its performance and energy output. Specific design and analysis for any potential biomass material always needs to be taken into account. Sustainability of biomass materials is an emerging field. Many materials suggested for combustion are by-products from other industries, including many types of shells, dry distiller's grains pelletized (DDGPs), corn stalks, et cetera. These products can be obtained rather inexpensively.

Small rural Nebraska greenhouse systems and households typically run on propane gas systems. These houses usually are not connected to natural gas lines and cannot benefit from natural gas energy. A proposed option is to heat these systems using agricultural products like whole shelled corn. Corn is readily available in the Midwest and has a high energy content. Corn is typically fed to a burner using an auger as shown in Figure 1.8. These augers are usually run by on/off controls based on a control point temperature. This control point could be based on one of two options, the inside greenhouse temperature or the furnace temperature. One option is to set low, medium and high heat output settings on the auger based on the heating needs instead of a single on/off stage. Another possibility is to run the augers using a fuzzy logic controller and modify the design presented by Chao, et al. (2000).

Figure 1.8. The Corn Auger in a bio furnace feeding system.

When selecting a biomass furnace system, several factors need to be taken into account. These include fuel type and availability, fuel effectiveness, storage, and furnace selection. A major disadvantage of biomass furnaces is that they require more maintenance and observation than propane furnaces. Reasons for this include variability in biomass fuel, ash removal, and reloading of the fuel bin. Automation of this process, which may be aided by pelleting, can save a grower time and allow them to focus on their plants. If biomass fuel can achieve size uniformity, it can be loaded into a furnace easily using an auger from a fuel bin. However, this process can be quite expensive; pelletizing dry fuel can increase the biomass fuel cost by about 60%, as shown in Table 1.3. If the biomass fuel is wet, this increase can reach almost 500% (Mani et al. 2006). One of the major advantages of heating with whole shelled corn is that it is purchased already pelletized. Also, corn has a high energy content, and is readily available and inexpensive in Nebraska.

Table 1.3. Effect of Various Fuel Options on the Cost of Pellet Production (Mani, et al. 2006).

Burner Fuel Options	Fuel Cost ($/t)	Pellet Cost ($/t)
Wet biomass	10	48.53
Dry biomass	32	50.57
Fuel pellets	52	52.31
Natural gas	10/GJ	64.48
Coal	40	49.75

There are a variety of biomass furnaces available, each dependent on the type of fuel and heating needs. For instance, Biomass Combustion Systems Inc. has two primary types of unit hot air shop heaters and water boilers. The air shop heaters are sized at 250,000 Btu/hr, 500,000 Btu/hr, and 800,000 Btu/hr. The water boilers are sized from 100 to 600 HP. All of their furnaces are designed for wooden logs to be manually loaded into the system when refueling is required. Another company, Fahrenheit Technologies Inc., sells a home biomass furnace which can utilize most pelletized fuels and is listed at 99%

combustion efficiency. A larger list of biomass furnace companies with units less than 1 million Btu/hr can be seen in Appendix I. There are several different configurations for a biomass furnace. Figure 1.9 shows an example of a horizontal draft system. There are several other ways to orient a furnace which can be seen in Appendix H. However, in a biomass furnace, the fuel typically resides at the bottom of the furnace because the biomass cannot be immediately combusted like propane, and requires burning time.

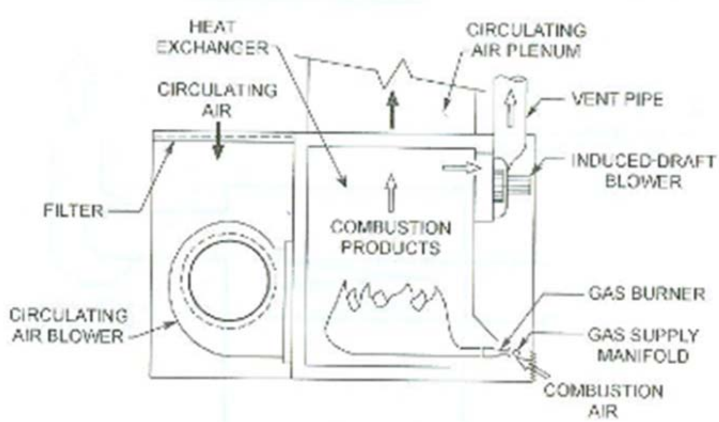


Figure 1.9. Basement Category Horizontal Draft Furnace (ASHRAE Systems and Equipment Handbook 2000).

Little research has been done looking at biomass fuel blending with other biomass. However, two previous studies investigated biomass blending with coal. (Sami et al. 2001) investigated the blending biomass fuels with coal. They reported four main conclusions: “1. Blend combustion resulted in improved combustion efficiencies compared to coal-only combustion. 2. Increasing fuel loading resulted in higher temperatures compared to the coal-only case. A downstream shift in the location of the peak temperature was also observed. Higher temperatures may also increase thermal NOx levels. 3. Decreasing the secondary air resulted in almost the same temperature profiles (hence same thermal NOx level) as those of the coal-only case. However, the burnout was improved significantly. 4. In order to maintain the same equivalence ratio, it is better to reduce the secondary air flow than to increase the fuel flow rate. However, it should be noted that the heat throughput will also decrease slightly." Nussbaumer (2003) reached similar conclusions: "A co-utilization of biomass with other fuels can be advantageous with regard to cost, efficiency, and emissions." The positive effect in regard to emissions is reduced SOx and NOx emissions because biomass has lower sulfur and nitrogen contents than other fuels. Also, biomass has a high volatile content and can be utilized in re-burning emitted air to achieve higher NOx removal. On the other hand, the main drawbacks associated with blending are the additional investment of retrofitting new biomass equipment to coal systems and the increased fouling and corrosion biomass causes.

Biomass Combustion Process

There are two basic verbal descriptions of combustion. The first chemical statement is for a complete combustion, given as:

$$Fuel + Oxygen \rightarrow Water + Heat + Carbon\ Dioxide \qquad (2)$$

This is the ideal state of a combustion process. Practically, though, this reaction will not occur in most applications. Because most combustion processes are open to the atmosphere, oxygen will not be the only gas input or substrate present in the left side of the equation. A second form of combustion is known as incomplete combustion, and is given as:

$$Fuel + Air \rightarrow Water + Heat + CO_2 + CO + NOx + SOx + PM + HC \ldots\ldots\ldots\ldots\ldots\ldots\ldots(3)$$

Where: CO2=Carbon Dioxide; CO=Carbon Monoxide; NOx=Nitrous Oxides; SOx=Sulfuric Oxides; PM=Particulate Matter; HC=Hydrocarbons (including all possible substrates)

The combustion products on the right side of the equation depend on several factors, including fuel type, fuel state, temperature of the fire box, air flow rate, and fuel flow rate. The complete combustion equation assumes a fuel that will be used in the generic form of C_xH_y, such as propane (C_3H_8). Agricultural products, however, are not often identifiable in this form, as their chemical formulae are nearly impossible to generalize. Simply put, agricultural products contain some nitrogen, ash and sulfur. The NOx seen in the product emissions is a result of both atmospheric nitrogen, as well as nitrogen within the material reacting within the combustion chamber. The SOx emissions are a result of the sulfur contained in the fuel. The PM content of the emissions results from the ash content of the burnt fuel breaking down into increasingly smaller particles and escaping with the other flue gases. CO is a result of the fuels' inability to completely oxidize all carbon atoms. At flame temperatures greater than 1000° F, the reactions become progressively more incomplete, resulting in more pollutant emissions being released. Lastly, carbon dioxide is the fully combusted form of carbon seen in emissions.

State and National Emissions Standards and Regulations

The Clean Air Act (CAA) is the main law governing emissions. The extension of the CAA was created in 1970 and detailed the first rules regarding emissions. It was required of the Environmental Protection Agency (EPA) to set and enforce these emission limits. Since 1970, there have been two amendments to the act, in 1977 and 1990. The National Ambient Air Quality Standards (NAAQs) are the goals established by the CAA. There are six main pollutants listed in the NAAQs, shown in Table 1.4. The primary standards are such because they deal with human health and safety; the secondary standards are for aesthetic and natural resource purposes.

An attainment area is a county which is at or below the NAAQs standard for one or more of the criteria pollutants. Conversely, if a county exceeds one of these standards it is known as a nonattainment area and has to more strictly monitor and report its air quality to the EPA. Nonattainment areas also must develop and implement a plan to meet the NAAQs standard. If this is not met, the area risks losing federal funding and faces further sanctions. Figure 1.10 is a map of the nonattainment counties in the United States as of July 2009. As seen in this figure, Nebraska currently has no counties on the nonattainment list. Nebraska's air quality standards follow the CAA and NAAQs and can be found at Nebraska Department of Environmental Quality (NDEQ) website under Title 129, Chapter 4.

Table 1.4. NAAQS Standards

National Ambient Air Quality Standards

	Primary Standards		Secondary Standards	
Pollutant	**Level**	**Averaging Time**	**Level**	**Averaging Time**
Carbon Monoxide	9 ppm (10 mg/m^3)	8-hour (1)	None	
	35 ppm (40 mg/m^3)	1-hour (1)		
Lead	0.15 $\mu g/m^3$ (2)	Rolling 3-Month Average	Same as Primary	
	1.5 $\mu g/m^3$	Quarterly Average	Same as Primary	
Nitrogen Dioxide	0.053 ppm (100 $\mu g/m^3$)	Annual (Arithmetic Mean)	Same as Primary	
Particulate Matter (PM_{10})	150 $\mu g/m^3$	24-hour (3)	Same as Primary	
Particulate Matter ($PM_{2.5}$)	15.0 $\mu g/m^3$	Annual (4) (Arithmetic Mean)	Same as Primary	
	35 $\mu g/m^3$	24-hour (5)	Same as Primary	
Ozone	0.075 ppm (2008 std)	8-hour (6)	Same as Primary	
	0.08 ppm (1997 std)	8-hour (7)	Same as Primary	
	0.12 ppm	1-hour (8)	Same as Primary	
Sulfur Dioxide	0.03 ppm	Annual (Arithmetic Mean)	0.5 ppm (1300 $\mu g/m^3$)	3-hour (1)
	0.14 ppm	24-hour (1)		

(1) Not to be exceeded more than once per year.

(2) Final rule signed October 15, 2008.

(3) Not to be exceeded more than once per year on average over 3 years.

(4) To attain this standard, the 3-year average of the weighted annual mean PM2.5 concentrations from single or multiple community-oriented monitors must not exceed 15.0 μg/m3.

(5) To attain this standard, the 3-year average of the 98th percentile of 24-hour concentrations at each population-oriented monitor within an area must not exceed 35 μg/m3 (effective December 17, 2006).

(6) To attain this standard, the 3-year average of the fourth-highest daily maximum 8-hour average ozone concentrations measured at each monitor within an area over each year must not exceed 0.075 ppm. (effective May 27, 2008)

(7) (a) To attain this standard, the 3-year average of the fourth-highest daily maximum 8-hour average ozone concentrations measured at each monitor within an area over each year must not exceed 0.08 ppm.
(b) The 1997 standard—and the implementation rules for that standard—will remain in place for implementation purposes as EPA undertakes rulemaking to address the transition from the 1997 ozone standard to the 2008 ozone standard.

(8) (a) The standard is attained when the expected number of days per calendar year with maximum hourly average concentrations above 0.12 ppm is $\leq$ 1.
(b) As of June 15, 2005 EPA has revoked the 1-hour ozone standard in all areas except the fourteen 8-hour ozone nonattainment Early Action Compact (EAC) Areas. For one of the 14 EAC areas (Denver, CO), the 1-hour standard was revoked on November 20, 2008. For the other 13 EAC areas, the 1-hour standard was revoked on April 15, 2009.

Source (www.epa.gov/air/criteria.html)

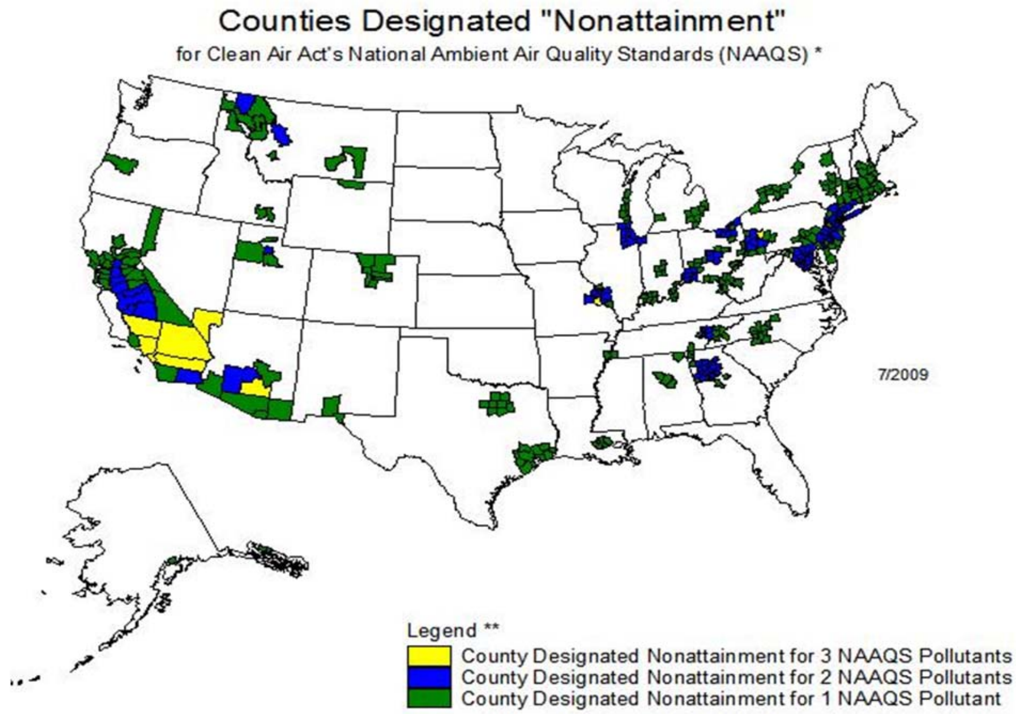


Figure 1.10. Nonattainment counties as of July 2009 (http://www.epa.gov/air/oaqps/greenbk/mapnpoll.html).

In 2009, the EPA proposed new rules regarding greenhouse emissions which can be found on EPA's website under Proposed Endangerment and Cause or Contribute Findings for Greenhouse Gases under Section 202(a) of the Clean Air Act. Proposed Rule 40 CFR Chapter 1. This proposal lists six greenhouse gases that must be monitored and/or regulated, including CO_2, methane (CH_4), nitrous oxide (N_2O), hydrofluorocarbons (HFCs), perfluorocarbons (PFCs), and sulfur hexafluoride (SF_6).

Biofuel emissions can be variable due to several factors, including the cultural environment in which they are grown or exposed to. Some examples of exposure include agricultural chemicals like pesticides or herbicides that may have come into contact with the bio-fuel or plant material while it grew; genetic differences between the crop varieties or genotypes; and other issues stemming from the fact that bio-fuels were once living organisms. Traditional liquid and gas fuels typically used for heating and energy production are very uniform in composition and quality when compared to bio-fuels. Thus, emissions from nonrenewable energy sources like propane can be closely predicted using well-known thermodynamic equations for complete or incomplete combustion (Cengel and Boles, 2004).

There are two main types of pollutants associated with biomass burning. The unburnt type, specifically ash, is primarily an issue with the performance of furnace equipment used. Emitted pollutants are a function of the biomass material used. Some of the main unburnt pollutants are CO, CO_2, NOx, SOx, PM (Davis and Cornwall, 2008).

Since biomass combustion is largely unpredictable, direct sampling is required to determine emission levels. Carbon monoxide is critical to measure for two reasons. First, it is extremely dangerous to human health. At concentrations exceeding 5,000 parts per million (ppm) the gas is lethal to humans within a few minutes. The second reason CO must be measured is that its levels represent the incompleteness of the reaction. To generate full energy out of a combustion system, the chemical compounds need to be fully oxidized. More CO conversion to CO_2 is beneficial to the energy utilization of the system. According to Davis and Cornwall (2008), CO levels have been basically unchanged in the last 20 years. Due to this, two primary sinks have been proposed – "reaction with hydroxyl radicals to form carbon dioxide and removal by soil microorganisms" (Davis and Cornwall, 2008).

Carbon dioxide is useful to measure because, like CO, its emissions translate into completeness of combustion. Carbon dioxide can also be hazardous to human health. At concentrations exceeding one percent, humans begin to feel adverse effects including headaches and drowsiness. As concentrations continue to rise, toxicity may occur, eventually leading to loss of consciousness and death. Biomass carbon dioxide emissions are useful to account for because it is a greenhouse gas. All biomass consists of carbon: therefore, carbon dioxide will always be a by-product of combustion. Emission levels of CO and CO_2 are largely dependent on the amount of fuel burned. As combustion levels rise more CO should fully oxidize to CO_2 in the emissions.

Nitrous oxides are important to measure because they are adverse to human health for two reasons. The first is that the several different nitrous oxides (N_2O, NO, NO_2, NO_3) can all react with ozone in the troposphere and stratosphere to fully oxidize to NO_3 (Davis and Cornwall, 2008). The second is that NO_2 and NO_3 then return to the earth, combining with precipitation to form acid rain (nitric acid HNO_3). The two primary sources of nitrous oxides from fuel combustion are nitrogen in the fuel itself and reaction with N_2 at higher combustion temperatures. While atmospheric N_2 is generally innocuous at combustion temperatures exceeding 1,600 K, 1327 C, or 2421 F, atmosphere N_2 reacts with atmospheric O_2 to form NO.

Sulfur oxides operate similarly to nitrous oxides in that the ultimate fate of most sulfur oxide compounds includes reacting with atmospheric ozone and being redeposited through acid rain. Generally, the sulfur emissions react with O_2 in direct proportion to the amount of sulfur in the fuel. For every gram of sulfur, two grams of SO_2 or SO_3 are formed. Due to ash generation, though, some of the remaining sulfur does end up in the ash created; typically, 95 percent is assumed in sulfuric emissions (Davis and Cornwall, 2008).

Particulates are also a concern with combustion because they are detrimental to human health. Originally, the NAAQs standard was based on total suspend particles but the standard was changed because most of PM particles greater than 10 μm in diameter will not be inhaled deeply into the lungs. The standard now focuses on $PM_{2.5}$ (Davis and Cornwall, 2008). In biomass heating, particulates will always be of concern due to ash generation from agricultural products and the aerosolizing of this ash. Fine particulates lead to several health concerns including asthma, bronchitis, cancer and eventually death.

A major question regarding biomass emissions is which emissions are the most important to monitor. The January 2011 EM magazine wrote about this issue. "While there is no controversy around the fact that the substitution of fossil fuels by sustainably produced biomass leads to the reduction of CO_2 emissions, the emissions from biomass combustion of NOx, organic carbon, and PM are being debated by scientists and legislators and the emphasis is currently being placed on PM emissions" (Musil-Schlaeffer, et al. EM January 2011 pg. 14-15).

In Canada, wood biomass combustion accounted for nearly 15 times the $PM_{2.5}$ emissions of the electric power utilities in 2007. One of the major issues with $PM_{2.5}$ is that the emissions are difficult to follow for high concentration and short term exposure. The Johnson January 2011 EM article suggests that the current $PM_{2.5}$ NAAQs are not effective at protecting the population from PM hazards for three reasons. These reasons include the ineffectiveness of the current EPA monitoring network for monitoring PM emissions in rural areas, ineffectiveness of improved technology and forcing regulations that would remove outdated equipment, and the inability of models for outdoor wood boiler setback distances to adequately account for real world conditions and environmental variability.

Another question arises as to how the different emissions can be reduced. There are two methods available. The first method is to control the initial source. By insuring that the combustion reaction taking place will be as complete as possible; the emissions or criteria pollutants could be reduced significantly. This method would require controlling either the fuel loading rate to the system, the air flow rate to the system, the type of air to the system, or a combination thereof. The second method is to manage the pollutants after they have been created. This could be accomplished by using devices like catalytic converters, cyclone separators, electrostatic precipitators, wet scrubbers, lime/limestone desulfurization, or baghouses.

Another method is to control the heating source using a staged combustion furnace. An example of a staged combustion furnace is shown in Figure 1.11. Staged combustion is beneficial because these systems can generate higher efficiencies and reduce emissions. Considering the January 2011 EM article "Getting There High-Efficiency and Low-Emissions Wood Heating," staged combustion can reduce particulate emissions by nearly 90 percent. Furnace efficiency is increased by using forced heated air through a secondary chamber to achieve a more complete combustion. Figure 1.12 shows the expected emissions levels for different fuels/furnaces specifically the 90 percent reduction.

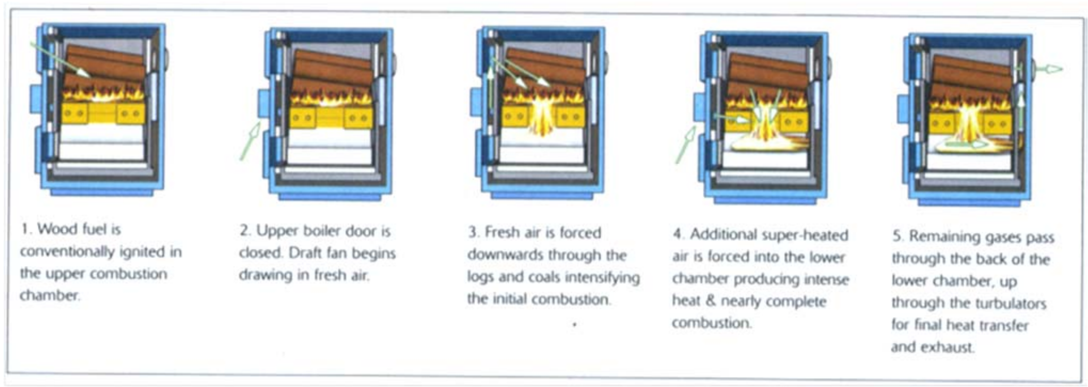


Figure 1.11. Staged Combustion Example Jan 2011 EM pg. 20.

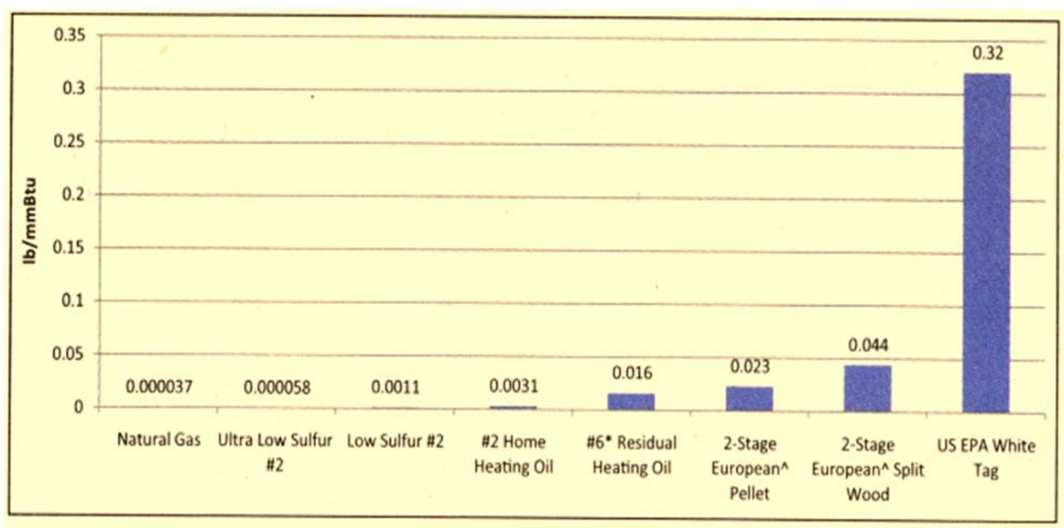


Figure 1.12. PM2.5 Emissions from Different Heating Systems (Jan 2011 EM pg. 22).

Catalytic converters are often used "to promote specific reactions such as: NOx to N_2, CO to CO_2, and hydrocarbons to CO_2 and H_2O" (Davis and Cornwell 2008). A cyclone separator is a device that will agitate as it spins, separating some of the particulate matter from the flue gas. An electrostatic precipitator is a device with a metallic path through which the air flows. The metal plates or tubes are used and have a positive charge to attract the air pollutants to collect them. Wet scrubbers can also be used to collect pollutants because the pollutants can aggregate with the liquid particles collecting them out of the air stream. Lime/limestone desulfurization can be used to remove sulfur from the air. Another method to remove NOx from a system is to inject urea into the emission lines. Finally, baghouses are also an option. A baghouse is essentially a large filter which can collect particles out of the air stream (Heinshon and Kabel 1999).

All of these previous devices can be used to reduce pollution. However, an important necessary step is to determine which, if any, pollutant gases are being over-emitted, and to what degree. An issue that occurs in catalytic conversion is fouling on the catalytic converter surface. This can occur from high sulfur contents in the emission lines. As expected, SOx removal would need to occur before a catalytic convertor was used if SOx was an issue. Four of the methods described above can be effective at PM removal. However, each is designed for different removal rates and air flow volumes, leading to various costs.

In the case of small Nebraska greenhouse systems, it is currently difficult to estimate if any individual system would need to be regulated. With the exception of PM, most pollutants are unlikely to be great enough to cause much concern in such a small system. As long as the pollutants are expelled and do not come back into the greenhouse (backfiring, leaks, etc.), they should not directly impact the grower. However, if a small city were to switch to biomass heating at each household, pollutant emissions could become a more serious issue. Most small biomass furnaces are quite comparable to a campfire running continuously. Little research has been done into the expected emissions from any particular biomass fuel, including shelled corn, and an emission study would be more specific to the burner type than the fuel. If additional emission control methods are required, most of the options listed above would be too costly or large to be of benefit to a small greenhouse system. The most practical options for reducing particulates would be a small cyclone separator, catalytic conversion, bag filters, or a staged combustion furnace. Purchasing a staged combustion furnace initially would be beneficial both to control emissions and increase efficiency.

A life cycle assessment of co-firing biomass with coal was performed using the cradle to the grave method by Mann and Spath (2001). The authors concluded that blending was beneficial to sustainability in several areas. First, blends with coal were created at 5 and 15 percent biomass. These mixes yielded CO_2 reductions of 5.4 and 18.2 percent, respectively. Also, SOx, NOx, non-methane hydrocarbons, particulates and CO were all reduced. The total system energy consumption was reduced by 3.5 and 12.4 percent for each blend, resulting in significant improvements of energy sustainability. This study suggested that biomass is more sustainable in comparison to coal. While energy performance may not be quite as effective with biomass additions, it is still beneficial to reduction of emissions.

Summary

In addition to energy conservation measures, Nebraska greenhouse growers are becoming increasingly interested in the use of alternative fuel sources (biomass, waste, wind, solar, et cetera). New technologies and applications are becoming available continuously and new research is needed to evaluate biomass heating technologies for commercial greenhouse applications. Important instrumentation and control questions regarding granular biomass fed furnaces, especially: how to measure the amount of granular or pelletized bio-fuel use needed for calculating furnace efficiencies; whether the auger feed rate can be changed automatically according to the progress of the fire; and whether these systems can lead to increased profitability and better quality crops.

Objectives

The overall goal of this project was to improve small Nebraska greenhouse profitability and sustainability through alternative fuels sources and to understand the sustainability of greenhouses focusing on efficiency and emissions. These concepts lead to the following specific objectives listed below

The specific objectives of this project were:

1) to determine the thermal properties of potential bio-fuels that might be used for greenhouse heating.
2) to test the performance of common pelletized bio-fuels in a biomass furnace.
3) to compare the performance of bio-fuels with propane heating in a typical Nebraska greenhouse.
4) to compare air quality emissions for various bio-fuels.

Chapter 2 Materials and Methods

This research study was divided into these categories: (a) greenhouse instrumentation, and control of the biomass heater and greenhouse environment, fuel selection (b) properties of the potential bio-fuels including bulk density, moisture content and bomb calorimetry tests, (c) determination of the efficiency of a low heat output biomass pellet furnace based on thermodynamics and heat and mass transfer principles, (d) cost analysis comparing the effective fuel costs against other fuel options, (e) air quality emissions tests, and (d) statistical analysis based on p-value significance tests, f-distributions and analysis of variance.

Instrumentation and Controls of the Biomass Heater and the Greenhouse Environment

A commercial greenhouse located just outside Firth, NE was used. This unit would be classified as a small family operated Nebraska greenhouse and is shown in Figure 2.1. The house produced ornamentals, bedding plants, hanging baskets, and annuals for in-house and farmers market sales during each spring for the last seven years. The greenhouse is a 23,000 ft^3 in volume with a floor area of 2000 ft^2. The house is covered with 6-mil, double polyethylene plastic, where the layers are inflated by a small fan for wind resistance. Figure 2.2 shows a full house ready for market in late April 2008. Vegetation was grown in the greenhouse each year except for 2011.

Figure 2.1. Firth Nebraska Cooperator Greenhouse (East side)

The house has installed a single 162,000 BTU per hour, Modine Aerothermes® propane, single-stage heater unit with an advertised 81% furnace efficiency and two Wind Master® 20-inch ventilation fans. Control of the propane heater and ventilation fans is by ON/OFF thermostat located in the middle of the 92-foot long house. A biomass pellet furnace (Eagle Manufacturing, Webster City, Iowa), was installed in 2007 through a USDA North Central Research Sustainable Agriculture and Research (NCR SARE) grant. It was spec'd at 100,000 Btu/hr and cost about $8000. An AutoCAD (Autodesk Inc. San Rafael, CA) sketch of the furnace is shown in Figure 2.3. Pictures of the biomass furnace are shown in Figure 2.4. This burner was tested for efficiency and used during the growing seasons of 2008, 2009, 2010 and an empty house in 2011.

Figure 2.2. Full Greenhouse ready for market in April 2008.

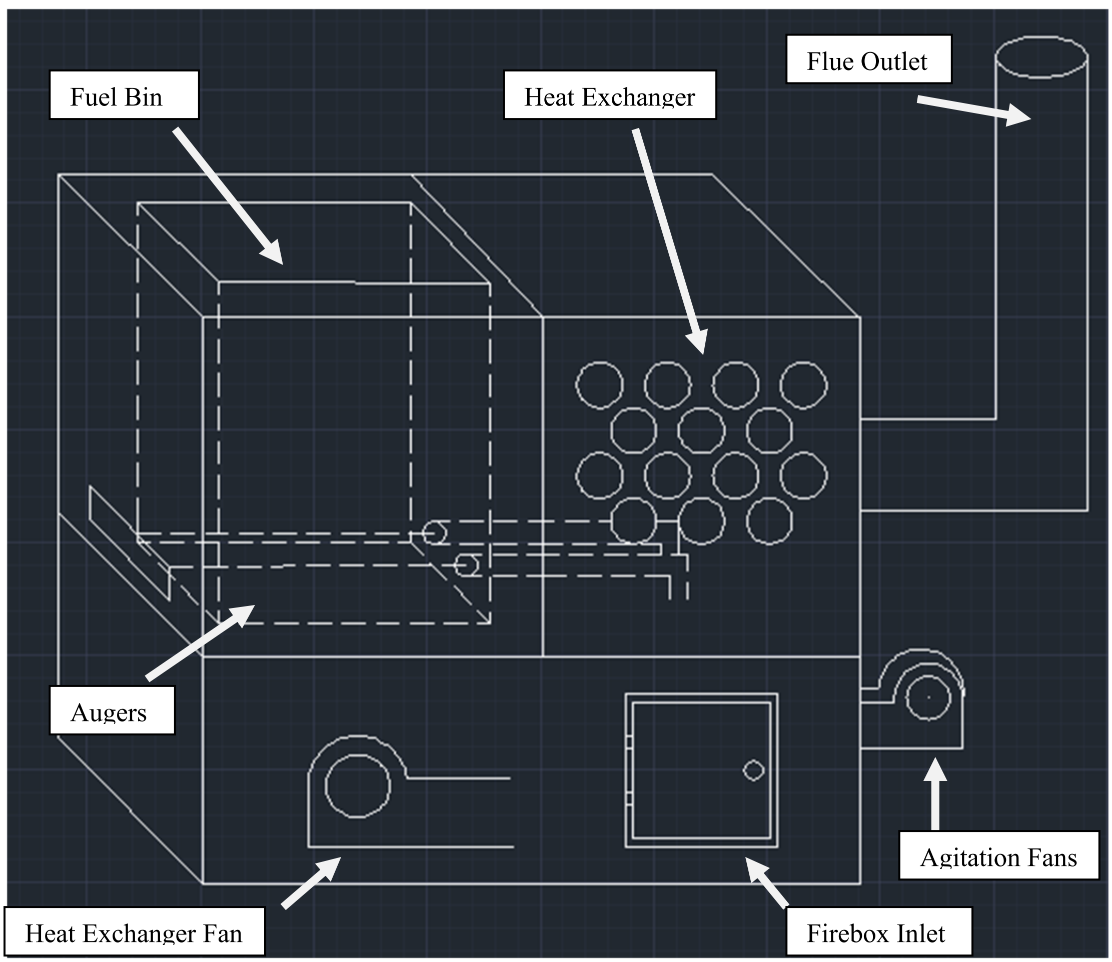


Figure 2.3. Biomass Furnace Side View Schematic.

(a)

(b)

(c)

Figure 2.4. Biomass Furnace (a) biomass pellet burner (upper left), (b) combustion agitation fans (upper right), and (c) twin auger feed (lower left) (Meyer, et al, 2009).

LabVIEW® (National Instruments, Austin, TX) software was used to create data acquisition systems for this project. The front panel of the virtual instrument (vi) created to monitor the runs during the spring of 2011 is shown in Figure 2.5. To monitor the system several devices were used. These include: two EI-1050 humidity and temperature probes (LabJack, inc., Denver, CO) placed on the inside and outside of the greenhouse, two type K thermocouples placed in the path of the biomass temperature and flue temperature, and a double wire connected to the auger voltage. The EI-1050 sensors and the auger voltage wire were connected to an U12 LabJack Datalogger (LabJack, inc., Denver, CO) data logger. The thermocouples were connected to a WLS-Temp wireless IEEE 802.1 data logger (Measurement and Computing, Norton, MA). The WLS-Temp device is operated using Insta-Cal setup software provided with Measurement and Computing devices. The program was developed to record data every 10 min during the running time and to save data to a file on a supervisory computer. The program was also designed to record furnace and auger and ventilation events using split-core current sensors. This data was then uploaded to a Microsoft Excel spreadsheet for data analysis and plotting.

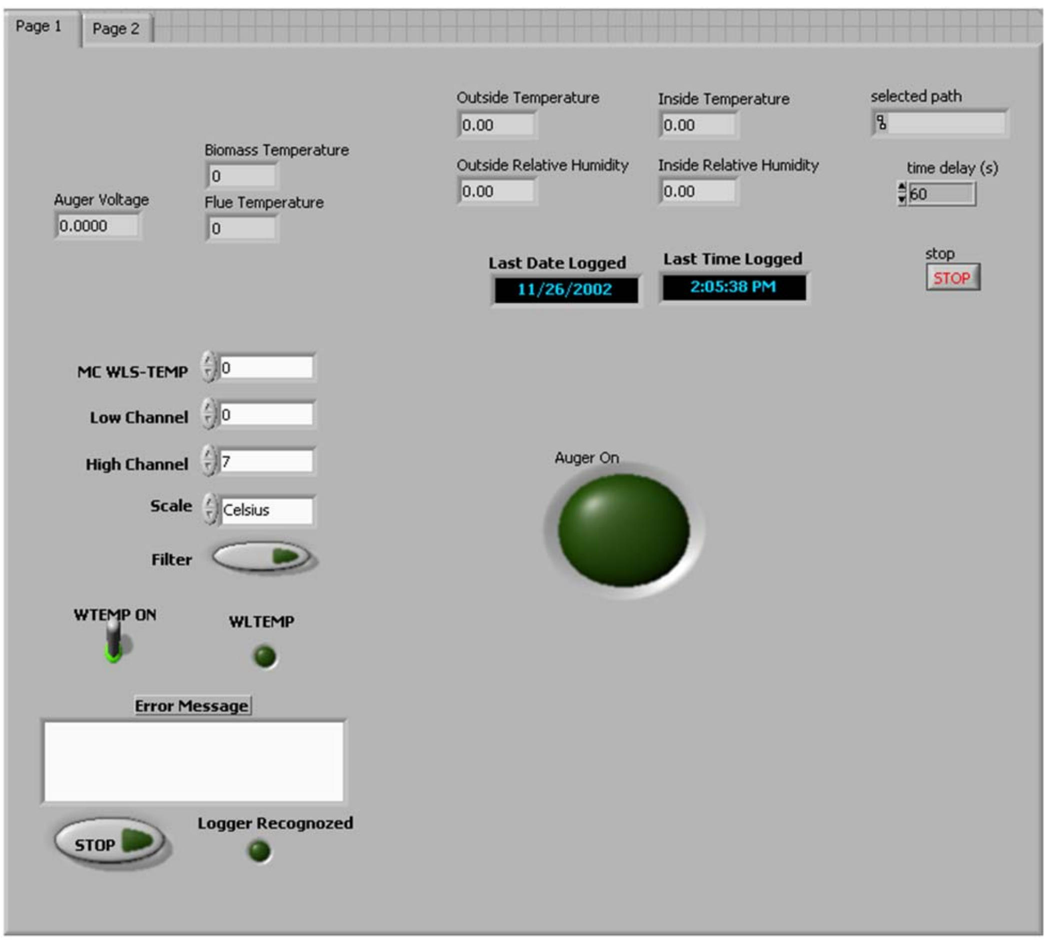


Figure 2.5. 2011 Greenhouse LabVIEW vi Front Panel.

Fuels Selected

Initially, three different pure fuel types were tested in the burner. These were whole shelled corn, wood pellets and DDGPs. When DDGP pellets were tested in the burner by themselves in 2010, they were found to clog the system and heat inefficiently. The DDGPs showed higher emissions, along with a thick plume emitted during the test runs. As a result, only whole shelled corn and wood pellets were run individually as pure fuels.

Two blends of bio-fuels were composed in an attempt to take advantage of the burn properties of each fuel. Wood pellets generally burn colder than the corn (about 930°F) and with little or no ash generation. Shelled corn was found to burn around 1110 °F with typically more ash generated. It was thought that by combining both fuels, one may be able to maximize the fuel energy generation and reduce the ash creation. Two blends were created on a 50/50 mass basis: (a) corn and DDGPs and (b) corn and wood pellets. Three runs per fuel type along with a before/after scenario of cleaning the burner allowed 24 test runs for this study.

Adaptive Modeling of the Greenhouse Environment based on Thermodynamics (First and Second Laws) and Heat and Mass Transfer Principles

The greenhouse environment was modeled using the greenhouse heating equation shown by Equation 4. This heating analysis can be used to determine the average heat loss from the greenhouse either currently or over a period of time. This heat loss can also be used to predict the expected heat loss and compare that against the actual heat loss to see if there are large unexpected losses somewhere in the system. Equation 4 is given as:

$$\frac{dT}{dt} = \frac{1}{\rho*Cp*V}*(qheater + a*S*Af) - \frac{\dot{V}}{V}*(T\text{-}Tout) - \frac{U*As}{\rho*Cp*V}*(T\text{-}Tout) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(4)$$

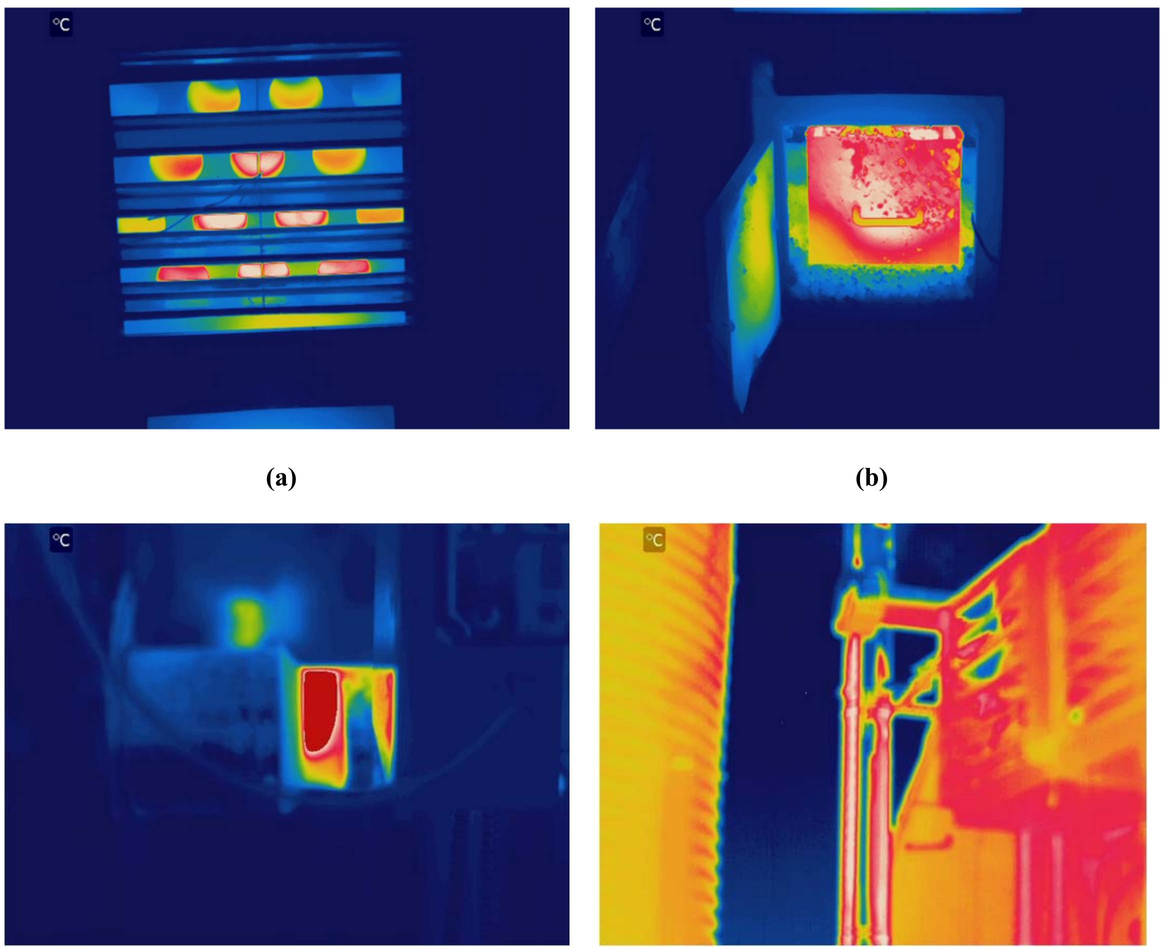
Variables in this equation are defined as:

Af	= floor area (m^2)
As	= surface area (m^2)
a	= building net solar heating efficiency (set to 0.28)
Cp	= specific heat of air (J / kg °C)
qheater	= heater output (W)
ρ	= air density (kg/m^3)
S	= solar irradiance (W/m^2)
T	= interior air temperature (°C)
Tout	= outside air temperature (°C)
U	= overall building thermal conductance (W/m^2 °C)
V	= building volume (m^3)
$\dot{V}$	= volumetric ventilation rate (m^3/s)

Thermal Image Analysis

Thermal images were taken during furnace tests to analyze heat loss from the flue and determine the temperatures of the burning biomass. A FLIR SC640 digital, thermal imaging camera (FLIR Systems, Boston, MA) was used to check and visualize heat losses around the greenhouse, including the biomass furnace and its flue. The camera provided a 640 line tonal image, using the camera itself or attachment of the camera to a PC using special software provided by FLIR systems. These images presented visual information as a series of false colors representing the temperature at various locations of the furnace and flue. Firth greenhouse furnace thermal images can be seen in Figure 2.6. Each thermal camera picture was analyzed using ThermaCAM Research Pro® software.

Each image was uploaded to the software and could be analyzed to find the average temperature of the firebox. This was performed using the software's rectangle and linear analysis focused on the hot zone of each firebox picture as seen in Figure 2.7. The software would then provide the average temperature of the region of interest and this temperature would be uploaded to the data analysis spreadsheet. The temperature found is assumed to be the firebox temperature throughout the entire run.



(a) **(b)**

(c) **(d)**

Figure 2.6. Thermal infrared images of the biomass pellet furnace. (a) hot air outlet port with louvers open and heat exchanger tubes exposed (765- 878 ^{o}F, measured). (b) Fire box open with shelled corn (765 - 1063^{o}C measured). (c) Exhaust pipe at rear of furnace (d) Outside flue pipe.

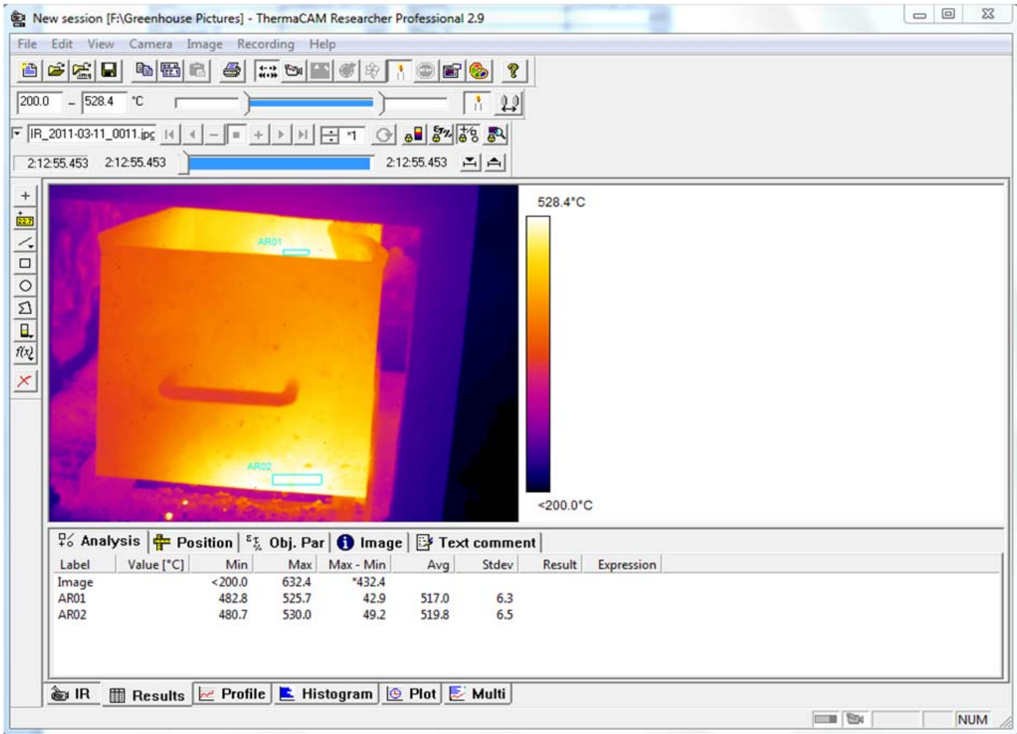


Figure 2.7. FLIR ThermaCAM Researcher Program.

Bulk Density

Laboratory tests were performed on each fuel to determine bulk density, moisture content and energy content. Bulk density tests were performed on each fuel using a 400 ml beaker. The beaker was filled with a known amount of water to determine the exact volume to the brim of the beaker. The empty beaker was then weighed and filled with each fuel three times and reweighed. The results were then compared to literature values when applicable. Equation 5 is the bulk density equation give as:

$$\rho bulk = \frac{mass\ (beaker + fuel) - mass\ (beaker)}{Volume} \quad \text{.......... (5)}$$

Fuel Moisture Content

The moisture content tests were performed according to the ASAE S352.2 DEC97 Standard. Two treatments of moisture testing were performed. The first treatment involved rewetting corn samples to achieve a variety of moisture contents. These samples were then tested for energy content using bomb calorimetry. The purpose of this experiment was to find the effect of moisture content on energy content. The second round of moisture testing was performed on fuel samples for burner combustion at the Firth greenhouse. The fuels sampled were whole shelled corn, wood pellets and DDGPs individually. Each fuel was tested five times using aluminum moisture dishes. Each test was performed using 15 gram samples and placed in a 103 °C oven for a period of three days. The samples were then removed and reweighed to determine the moisture lost. Each moisture content test result is wet basis. Equation 6 is the moisture content equation give as:

$$\text{Moisture Content (Wet Basis) (\%)} = \frac{\text{Mass (before drying)-Mass (after drying)}}{\text{Mass (before drying)}} * 100 \ldots\ldots\ldots\ldots \text{(6)}$$

Bomb Calorimetry

The data was acquired using a Parr 1241 (Moline, IL) adiabatic, oxygen bomb calorimeter, using the American Society Testing and Materials (ASTM) procedure designated D2015. The bomb calorimeter was located at the Industrial Agricultural Products Center (IAPC lab), Chase Hall on East Campus of the University of Nebraska. The oxygen bomb was prepared using a fuel sample, a Parr 45C10 nickel alloy fuse wire, and oxygen, shown in Figure 2.8. The fuel sample was weighed to roughly one gram and never exceeded 1.5 grams. The fuse wire was cut to about ten cm and was attached to the bomb as shown in Figure 2.8. The bomb could then be enclosed and filled with oxygen to about 30 atmospheres but never more than 40 atmospheres. The water bucket was filled with 2000 grams of water at a temperature between 24 and 28 °C. The bucket was placed inside the calorimeter shown in Figure 2.9. The oxygen bomb was then placed inside the bucket and the system was closed. Once the water temperature inside the bucket and inside the calorimeter had come to equilibrium, the initial temperature was recorded and the bomb was ignited. The water temperature of the calorimeter was monitored to find the peak temperature inside the system. After the peak temperature occurred, the bomb was removed from the calorimeter. The inside of the bomb was then washed using distilled water. The remaining unburned fuse wire was collected and measured for the length remaining. The bomb washings were then collected into a beaker and titrated using 0.0725N sodium carbonate solution. The washings were titrated to roughly 7 pH. The initial pH and volume of sodium carbonate used to titrate was recorded. The calculation for the energy content of the fuel was performed using Equation 7 given as:

$$Hg = 1.8 * \frac{tW - e1 - e2 - e3}{m};\ t = tf - ta\ ;\ e1 = c1;\ e2 = 13.17 * c2 * m;\ e3 = 2.3 * c3 \ldots\ldots (7)$$

ta = temperature at time of firing (°C); tf = final maximum temperature (°C)

c1 = milliliters of sodium carbonate solution used to titrate (mL)

c2 = percent of sulfur in sample (assumed 0.1 % for these tests)

m = mass of sample (grams); c3 = fuse wire consumed (cm)

W = energy equivalent conversion; Hg = energy content (Btu/lb)

1.8 is the conversion from cal/gram to Btu/lb

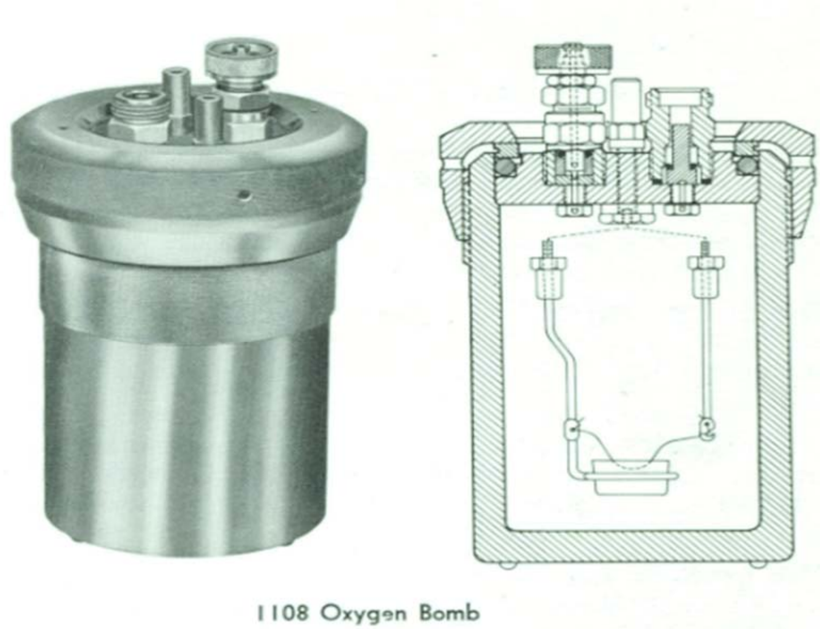


Figure 2.8. Sample Calorimetry Bomb Setup Parr 1241 Handbook pg 8.

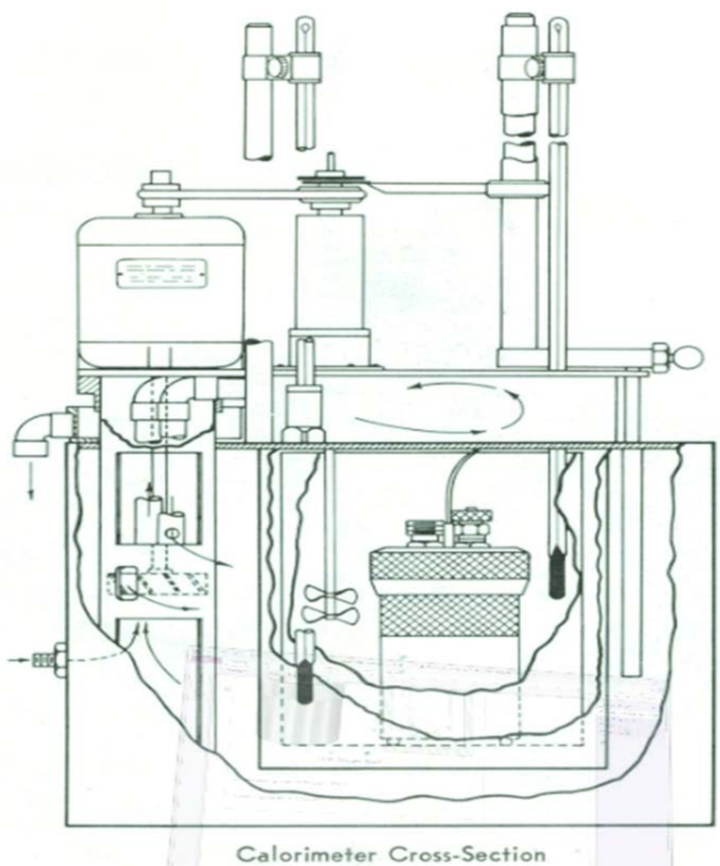


Figure 2.9. Bomb Calorimetry Test Equipment Parr 1241 Handbook pg 4.

Determination of the Efficiency of a Low Heat Output Biomass Furnace

The furnace efficiency calculations followed the analyses presented by the ASHRAE Systems and Equipment Handbook 2000. To determine the efficiency of the furnace, a total mass balance approach was used. The efficiency calculation was simply the actual energy gained from the cold side of the heat exchanger divided by the theoretical heat created by the fuel being burned.

The mass of fuel for determining the theoretical energy presented was measured directly using a scale on site, before it is loaded into the fuel bin. After each test run, the fuel remaining was vacuumed out of the bin and weighed to determine the net fuel mass used. The ash from each test was also collected from the firebox and weighed. The fuel was tested for energy content, moisture content, and bulk density in the lab which were described earlier in this section. The ash was also tested for energy content. The total energy consumed is calculated using the equation given as:

$$\dot{Q}\left(\frac{\text{Btu}}{\text{hr}}\right) = \frac{\left((\text{m in-m out})(\text{lbm})\text{*Fuel EC}\left(\frac{\text{Btu}}{\text{lbm}}\right)\text{-ash (lbm)*Ash EC}\left(\frac{\text{Btu}}{\text{lbm}}\right)\right)}{\text{Test time (hours)}} \quad (8)$$

The actual energy gained by the system is calculated by determining the average heat gained from the cold side of the heat exchanger over the course of each hour long run. The ΔT value is calculated by subtracting the greenhouse temperature from the biomass temperature. This value can then be applied in the following equations to find the total heat gained, given as:.

$$\dot{Q}\left(\frac{\text{Btu}}{\text{hr}}\right) = \dot{m}\left(\frac{\text{lbm}}{\text{hr}}\right)\text{*Cp}\left(\frac{\text{Btu}}{\text{lbm °F}}\right)\text{*}\Delta\text{T (°F)} \quad (9)$$

$$\Delta\text{T(°F)} = \text{Tbiomass (°F)-Tinside (°F)}$$

$$\dot{m}\left(\frac{\text{lbm}}{\text{hr}}\right) = \rho\text{air}\ \left(\frac{\text{lbm}}{\text{ft}^3}\right)*\text{A}(\text{ft}^2)*\tilde{\text{v}}\text{avg}\left(\frac{\text{ft}}{\text{min}}\right)*60\left(\frac{\text{min}}{\text{hr}}\right)\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

Once the theoretical and actual energies have been computed the efficiency calculation is simply:

$$\cap = \frac{\text{Actual Energy Gained (Btu/hr)}}{\text{Theoretical Energy Burned (Btu/hr)}}*100\ldots\ldots\ldots\ldots\ldots\ldots(11)$$

Determination of Heat Exchanger Efficiency

The heat exchanger efficiency is based on Cengel and Boles (2004). It is calculated by finding the specific heats of all four sides of the heat exchanger and the air flow rates on both sides as well. The air flow rate on the cold side of the heat exchanger was found using a Kurz hot wire, air velocity meter (anemometer), model 441S (Kurz Instruments, Inc., Monterey, CA) and the cross sectional area. The inlet area was measured using a meter stick. The anemometer was also positioned over nine separate points at the inlet to obtain an estimate of the air flow rate. The outlet of the cold side of the heat exchanger consists of 15 pipes. Each pipe had the same diameter and the velocity probe was positioned in front of each tube. This yields two results creating a high flow potential and low flow potential flow. The hot side of the heat exchanger has two agitation fans running at 262 cfm. Due to the high temperature of the air flowing from this side of the heat exchanger and the inability to access the firebox without opening the panels and distorting the air flow, this flow rate was not able to be verified because the anemometer cannot handle high air temperatures.

The specific heats were calculated using Table 2.1 from Cengel and Boles (2004). A linear equation was created using the data from the 0 to 600 °C range. The equation yielded two unknowns which are temperature and specific heat. The temperature values were generated using the LabVIEW program for the inlet and outlet of the cold side of the heat exchanger as well as the outlet of the cold side of the heat exchanger. Each of these temperatures created a specific heat.

Table 2.1 Specific Heat of Air at Different Temperatures (Cengle and Boles pg. 886).

Temperature, K	c_p kJ/kg · K	c_v kJ/kg · K	k
		Air	
250	1.003	0.716	1.401
300	1.005	0.718	1.400
350	1.008	0.721	1.398
400	1.013	0.726	1.395
450	1.020	0.733	1.391
500	1.029	0.742	1.387
550	1.040	0.753	1.381
600	1.051	0.764	1.376
650	1.063	0.776	1.370
700	1.075	0.788	1.364
750	1.087	0.800	1.359
800	1.099	0.812	1.354
900	1.121	0.834	1.344
1000	1.142	0.855	1.336

The specific heats were averaged over the course of the run and created an average specific heat for the test. The firebox temperature or the inlet of the hot side of the heat exchanger was calculated using the FLIR ThermaCAM Researcher described earlier. The firebox specific heat was calculated for each picture. This specific heat was assumed to be the constant value throughout each run. Air density was assumed to be constant at 1.2 kg/m^3. The heat transfer rate for each side is calculated and the efficiency is computed from these values:

$$Heat\ Transfer\ \left(\frac{Btu}{\min}\right) =$$

$$\frac{\left(\left(Cpout\left(\frac{KJ}{kg\,℃}\right)*Tout\ (℃) - Cin\left(\frac{KJ}{kg\,℃}\right)*Tin(℃)\right)*\rho air\left(\frac{kg}{m^3}\right)*Q\left(\frac{ft^3}{\min}\right)*.3048^3\left(\frac{m^3}{ft^3}\right)\right)}{1.055\left(\frac{KJ}{Btu}\right)} \text{.....................(12)}$$

$$\cap = \frac{Heat\ Transfer\ Cold\ Side\left(\frac{Btu}{min}\right)}{Heat\ Transfer\ Hot\ Side\left(\frac{Btu}{\min}\right)} * 100 \text{ ....(13)}$$

The net heat transfer total can be calculated by multiplying each run by its length of time as well. The heat lost can then be calculated by subtracting the cold side heat transfer from the hot side heat transfer.

Cost Analysis

The cost of each fuel was found during purchasing and recorded. These fuels include DDGPs, whole shelled corn, wood pellets, propane and natural gas. Each fuel cost was then reduced to a cost/unit of measure. In the case of the biomass materials this was \$/lbm while for propane it was \$/gallon and natural gas was \$/therm or \$/ft^3. This value was then multiplied by the energy content of each fuel to find the fuels cost per Btu. This value was then divided by the efficiency of each fuel's burner to obtain the true cost per each fuel type. This true cost was applied over the entire growing season and found the total savings or losses for the bio-fuel against propane and natural gas. Lastly this total savings calculated the payback period to implement a biomass furnace. The savings from previous years are shown in Table 2.2. These calculations are given as:

Table 2.2. Biomass Cost Savings for Previous Years Research (Meyer, et al, 2009).

	Fall 2007	Spring 2008	Fall 2009	Spring 2009	Spring 2010	Units
Fuel Type	**Shelled Corn ***					**(English)**
Bulk Density	62	62	62	62	62	lbm per bushel
Sample Bio Fuel Used	**1000**	**1000**	**1000**	**1000**	**1000**	bushels
	62,000	62,000	62,000	62,000	62,000	lbm
Energy Content	7,200	7,200	7,200	7,200	7,200	Btu per lbm
Total Energy	446,400,000	446,400,000	446,400,000	446,400,000	446,400,000	Btu
Pellet Furnace Efficiency	**0.67**	**0.67**	**0.67**	**0.67**	**0.67**	
Heat Produced	**299,088,000**	**299,088,000**	**299,088,000**	**299,088,000**	**299,088,000**	Btu
Unit Fuel Price **	**$3.05**	**$5.35**	**$3.21**	**$3.68**	**$3.41**	per bushel
Total Fuel Cost	**$3,050**	**$5,350**	**$3,210**	**$3,680**	**$3,410**	
Fuel Type	**Propane**					
Energy Content	91,500	91,500	91,500	91,500	91,500	Btu per gal
Gas Furnace Efficiency	**0.81**	**0.81**	**0.81**	**0.81**	**0.81**	
Equivalent Fuel Amount	4,035	4,035	4,035	4,035	4,035	gallons
Total Energy	369,244,444	369,244,444	369,244,444	369,244,444	369,244,444	Btu
Equivalent Heat Produced ***	**299,088,000**	**299,088,000**	**299,088,000**	**299,088,000**	**299,088,000**	Btu
Unit Fuel Price **	**$1.89**	**$1.95**	**$1.78**	**$1.28**	**$1.49**	per gallon
Total Fuel Cost	**$7,627**	**$7,869**	**$7,183**	**$5,165**	**$6,013**	
Percent Cost Savings	**0.60**	**0.32**	**0.55**	**0.29**	**0.43**	

* Shelled corn at 12 % dry basis.
** Based on local coop prices, including delivery, Firth, NE.
*** Equivalent amount of heat as would be produced from shelled corn at the respective furnace efficiency.

$$\text{True Fuel Cost (\$)} = \frac{\left(\frac{\text{Fuel Cost}\left(\frac{\$}{\text{unit}}\right)}{\text{Energy Content}\left(\frac{\text{Btu}}{\text{Unit}}\right)}\right)}{\cap\text{fuel burning}} \quad \text{(14)}$$

The total savings from the previous seasons was about \$15,000. Based on five separate heating seasons, the payback period of the furnace was 2.7 heating seasons of three months.

Gaseous Emissions

The emissions monitored during each test were carbon monoxide, carbon dioxide, sulfur oxides, nitrous oxides, and total particulate matter (PMtot). The gases were measured using Draeger test tubes (Draeger Safety, Inc, Pittsburgh, PA). Each draeger test was performed using one of the four test tubes and the hand gas detection air pump. A hand pump was positioned at the flue exit point as shown in Figure 2.10. The tube was then cracked open on each end and inserted into the hand pump. The tube was placed in the path of the flue gas and pumped the suggested number of times suggested by the company: CO=1 CO2=1 NOx= 1or 2 SOx=10. After pumping, the emissions could be estimated using the color changes and ranges on the side of the tube and recorded. The PMtot was measured by performing a complete mass balance calculation from the fuel used and ash remaining shown in equation 15 given as:

$$mass\ (fuel + air) = mass\ (ash + emissions) \quad \text{(15)}$$

The emissions were then compared against the flue temperature at the time of the test to observe the effect of firebox temperature on emissions.

Figure 2.10. Emissions Testing.

Statistical Analysis

P value significance tests were performed on each data set to determine the reasonable range of results. A 95 percent confidence interval and two range standard deviation of a normal distribution results in z values of 1.96 and -.1.96. A sample 95 percent confidence interval can be seen in Figure 2.11 taken from Myers et al 2007. The normal distribution z value can be calculated as seen in equation 16 where X is the data value, μ is the data mean, and σ is the standard deviation. The purpose of P value significant tests is to determine if a significant change has occurred against the mean sand standard deviation. The assumed data mean is calculated from the previous data from 2008 through 2010. The average efficiency of this data is 70.104 percent with a standard deviation of 10.828. This yields a 95 percent confidence interval of 51 to 89 percent efficiency.

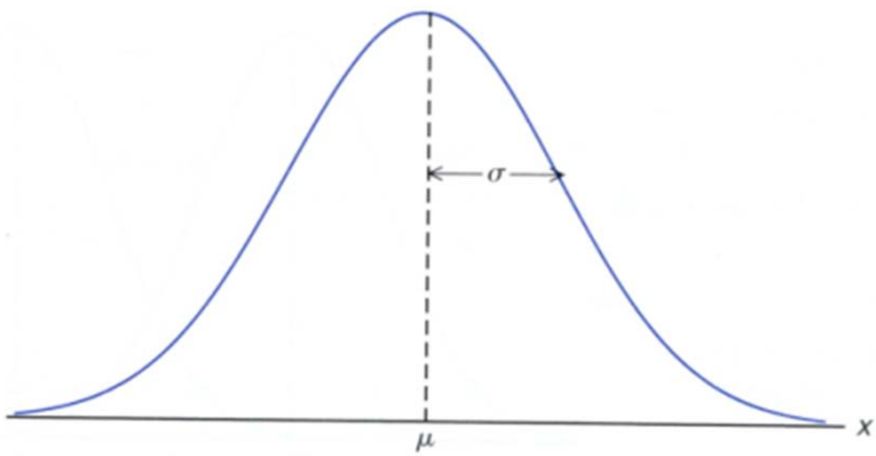


Figure 2.11. Normal Distribution of an experimental sample (Myers et al 2007 pg. 173).

$$Z = \frac{X-\mu}{\sigma} \quad \text{.....(16)}$$

F-distribution tests are useful to compare the statistical differences of two sample variances. A typical F-distribution can be seen in Figure 2.12. F-distributions were calculated using equation 17 where S is the variance of the sample and σ is the standard deviation.

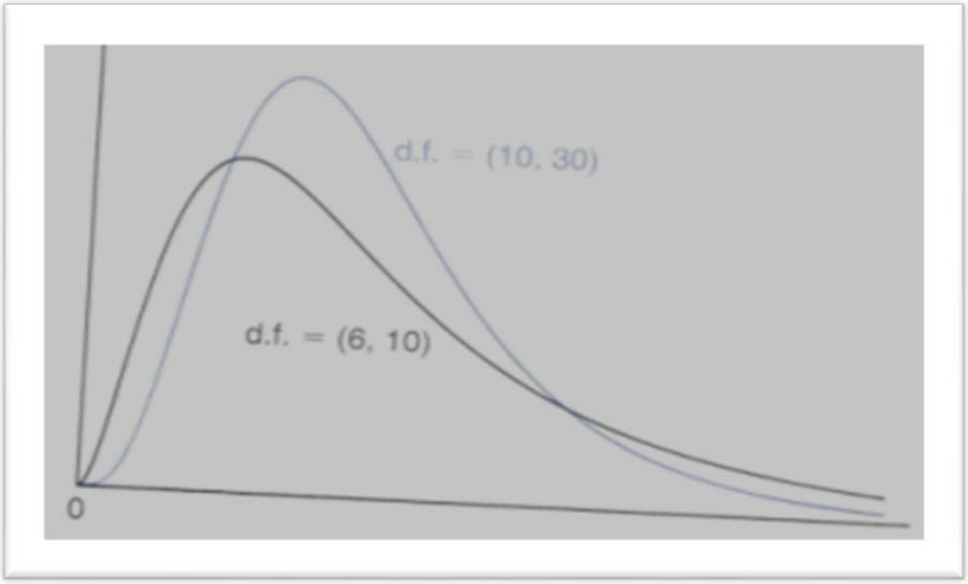


Figure 2.12. Typical F-Distributions (Myers et al 2007 pg. 262).

$$F = \frac{(S1)^2/(\sigma 1)^2}{(S2)^2/(\sigma 2)^2} \qquad (17)$$

Statistical Analysis Software (SAS Institute Inc., Cary, NC.) was used to perform the 1D ANOVA analysis. Dependent and independent variables are chosen to create different comparisons. The comparisons desired for this analysis included: efficiency vs. fuel type, cleaning, outside temperature and inside temperature, and emissions vs. fuel type, and vs. flue temperature. Index ranges were then chosen for the independent variables and results from analysis were used for the dependent variables. The index ranges and number of each chosen for the independent variable included: fuel types (different fuels, 4), cleaning (pre or post, 2), outside temperature (cold (<32 °F), moderate (32 to 60 °F), and hot (>60 °F), 3), inside temperature (cold (<60 °F), moderate (60 to 80 °F), and hot (>80 °F), 3), and flue temperature (cold (<325 °F), moderate (325 to 425 °F), and hot (>425 °F), 3). The important results of each 1D ANOVA test are the p-values and F-values. Low p-values ($<.1$) and high F-values(>1) indicate more significance in the result.

Experimental Design

Twenty-four runs were planned for this experiment using three fuel types: dry distiller's grains pelletized (DDGPs), whole shelled corn, and wood pellets. Along with the pure corn and wood tests, shelled corn was blended with wood, and corn was blended with DDGPs as a 50/50 mixture on a mass basis. The DDGPs were not tested as a pure fuel because a preliminary test in 2010 indicated that they performed quite poorly and clogged the burner system. Each fuel combination tested was replicated three times. After the initial 12 runs, the furnace heat exchanger was cleaned. Another subsequent 12 test runs were performed after cleaning to compare efficiencies before and after cleaning.

Each test run followed the protocol listed below:

1) The computer was set up and turned on to start the LabVIEW data logging software.
2) Enough fuel to fully cover the augers was weighed out (about 20 pounds). More fuel could be needed based on that day's requirements.
3) The firebox ignition (burn pot) was prepped with a one inch layer of wood pellets and one small scoop of corn covered with lighter fluid, and then the burn pot was lit.
4) The firebox was allowed about five to ten minutes to prime and establish a good fire.

5) After priming was complete, the air fans (cold side and hot side agitation fans), auger and furnace control were turned on.
6) When the firebox had reached its minimum internal temperature setting, about 300 °C, the auger automatically began to insert additional fuel.
7) Once the auger had started to automatically add fuel, 15 minutes of new fuel priming time was allowed before the official start of the test to allow the burner to use up the starter fuel.
8) Each test lasted one hour with the system allowed to run steadily and automatically.
9) After about 30 minutes had elapsed, the flue gases were tested for emissions.
10) After testing the flue emissions, the firebox door was opened and temperatures were recorded using thermal images (FLIR 640SC camera) and ThermaCAM Researcher Software.).
11) When the hour-long test was complete, the auger was turned off, but the two warm side air handlers or agitators were left running to use up the remaining fuel in the firebox.
12) The firebox was then allowed 20 minutes to complete the burn and then left to cool down to burning any remaining fuel in the firebox.
13) After cooling, the firebox unit was removed from the burner chamber and the ash was collected and weighed.
14) Finally, the remaining bio-fuel in the hopper was vacuumed out and weighed and subtracted from the initial weight.

The data which included (average fire box temperature, average flue exhaust temperatures, cold side heat output, furnace efficiencies, auger frequencies, percent ash to input fuel amounts, and exhaust emissions) were statistically tested for significance for each treatment and replications using a two-way Analysis of Variance (ANOVA). The heat exchange effectiveness ε was calculated according to Albright (1990). The furnace efficiency was calculated based on measured heat output (based on the temperature rise and air mass flow rate) and theoretical fuel heat content availability, determined through bomb calorimetry.

Chapter 3 Results and Discussion

Bulk Density

The bulk density analysis consisted of obtaining a beaker of a known volume and loading it with fuel. The beaker was weighed with the fuel before and after. The results of this analysis can be seen in Table 3.1

Table 3.1. Bulk Density of Fuel.

	Bulk Density						
	Test 1	Test 2	Test 3	average	(g/ml)	(kg/m^3)	(lbm/ft^3)
Volume (ml)	413						
Corn (g)	304	305	309	306	0.74	741	46.25
Wood Pellets (g)	266	268	268	267	0.65	647	40.42
DDGP (g)	247	245	247	246	0.60	596	37.22
Corn/Wood Blend							42.71
Corn/DDGPs Blend							41.11

The results for the whole shelled corn were found close to the industry standard value of 45 lbm/ft^3. Wood pellets and DDGPs were not found to have a standard bulk densities in the literature, so these tests results are the best estimate. The blends bulk densities were the average between the corn standard of 45 lbm/ft^3 and the experimental result with each other fuel individually.

Moisture Content

The first test was to find the impact of moisture content on fuel energy in whole shelled corn. The results of this analysis can be seen in Figure 3.1. From this figure, it can be concluded that energy content determined from bomb calorimetry decreases with increased fuel moisture content.

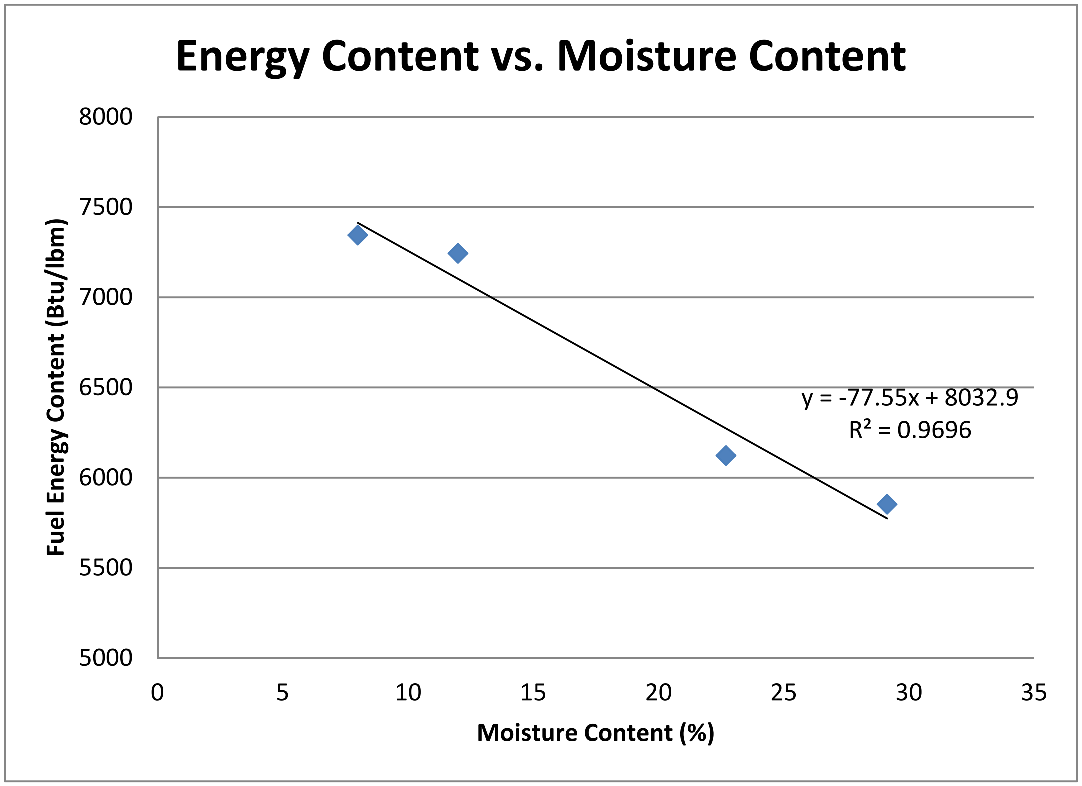


Figure 3.1. Corn Moisture Content vs. Energy Content determined by bomb calorimetry.

The second analysis provided data on the moisture content of the three individual fuel types. These results are presented in Table 3.2. The results of the moisture content tests show that each fuel type had an uniform moisture content throughout with little variation between samples. Assuming that the corn's moisture was consistent at about 11.7%, one might assume a consistent energy content of 7200 Btu/lbm for the entire shelled corn supply.

Table 3.2. Fuel Moisture Content.

						water weight	
Jar	Dish	Dish Weight (g)	Sample Weig	eight after dr	ht after dryi	% moisture	Averages
Corn	6	1.34	10.26	10.39	9.05	11.79	
Corn	3	1.33	10.24	10.41	9.08	11.33	
Corn	7	1.33	10.16	10.28	8.95	11.91	
Corn	9	1.33	10.11	10.24	8.91	11.87	
Corn	3	2.13	10.04	10.99	8.86	11.75	11.73
Wood Pellets	5	1.33	9.9	10.59	9.26	6.46	
Wood Pellets	1	2.13	10.09	11.56	9.43	6.54	
Wood Pellets	11	1.33	10.25	10.89	9.56	6.73	
Wood Pellets	8	1.32	9.96	10.61	9.29	6.73	
Wood Pellets	1	1.31	10.17	10.82	9.51	6.49	6.59
DDGP	4	1.32	10.14	10.26	8.94	11.83	
DDGP	1	1.33	9.94	10.06	8.73	12.17	
DDGP	188p	2.15	10.22	11.13	8.98	12.13	
DDGP	10	1.32	9.93	10.04	8.72	12.19	
DDGP	26-4-01	2.18	10.22	11.15	8.97	12.23	12.11

Energy Content

Bomb calorimetry tests were performed on the DDGPs, Wood Pellets, Corn Ash, Wood Pellet Ash, and Corn/DDGP Ash. The Corn and Corn/Wood Ash were also available from previous analysis and averaging between other samples. The results of these analyses are shown in Table 3.3.

Table 3.3. Fuel Energy Contents.

Units	Btu/lbm				
Energy Content Tests	Test 1	Test 2	Test 3	Average	St. Dev
Corn (11.5% moisture)	7279	7242.9	7251.1	7257.7	18.9
Wood Pellets	7929.7	7828.1	7918.4	7892.1	55.7
DDGPs	8284.8	8352.9	8026.9	8221.5	171.9
Corn Ash	3247.8	2481.2	1563.3	2430.8	843.4
Wood Ash	4256.1	5844.6	7147.9	5749.5	1448.3
Corn/DDGP Ash	1204.8	4469.4	3186.4	2953.5	1644.7
Corn/Wood Ash				4090.2	
Corn/Wood Blend				7546.0	
Corn/DDGP Blend				7710.8	

The DDGPs and Wood Pellets results had low standard deviations for each test suggesting that these fuel energy content values are consistent and reliable. However, the ash tests indicated a large amount of variability between tests suggesting the ash energy content really just depends on the furnace burn test. Most ash samples indicated that the material was not entirely combusted. Some ash samples exploded out of the fuel container into the collection basin of the oxygen bomb. Values of ash energy content for each fuel are probably close to an average value, but the ash results in such a low percentage of unburned energy that the value is negligible. These results for bio-fuel energy content were used throughout the rest of the efficiency analysis.

Heat Exchanger Cold Side Air Flow Rate

The inlet is a simple duct opening; however, the outlet is a set of 14 parallel pipes. The results of the air velocity and flow tests are shown in Table 3.4.

Table 3.4. Cold Side Air Flow Rates.

Inlet	Velocity (ft/min)
Quadrant 1	300
Quadrant 2	100
Quadrant 3	280
Quadrant 4	300
Quadrant 5	260
Quadrant 6	300
Quadrant 7	430
Quadrant 8	375
Quadrant 9	430
average (ft/min)	308.3 +/- 100.5
X Area (ft^2)	3.2
Flow Rate (ft^3/min)	1000.8

Outlet	Velocity (ft/min)
Pipe 1	400
Pipe 2	325
Pipe 3	250
Pipe 4	300
Pipe 5	275
Pipe 6	350
Pipe 7	325
Pipe 8	300
Pipe 9	350
Pipe 10	375
Pipe 11	375
Pipe 12	450
Pipe 13	250
Pipe 14	500
Average Velocity (ft/min)	344.6 +/- 72.18
Pipe X Area (ft^2)	0.05
14 Pipe X Area (ft^2)	0.7
Flow Rate (ft^3/min)	236.8

From the opening area and average velocities data, the inlet side was found to have a 1000.8 ft^3/min air flow rate, while the outlet has a 236.8 ft^3/min air flow rate (quite different). The air flow tests were performed when the furnace was off using cold air. Theoretically, it would be expected that the mass flow rate should not change through the cold side of the heat exchanger. If temperature and relative humidity on both sides were the same when converted to mass flow rate the difference between mass flow rate and air flow rate should not change side should yield the same results. There are two explanations for these results. There were friction losses through the pipe resulting in the reduced the air flow rate or more probably that the flow rate tests were not accurate. Typically, air flows are measured using 10 pipe diameter straightening tubes into or out of an air handler. In this case, the furnace had no duct work attached and the multiple exit pipe openings represent a challenge of air velocity measurement. The 1000 ft^3/min inlet flow rate in this study was similar to Dr. Meyer's inlet test results from the previous year of approximately 1200ft^3/min. However, he did not measure the outlet velocities.

The greenhouse grower cooperator (Stacy Adams) had reported significant financial savings using bio-fuels over the three year period which would imply the biomass heating was more effective. Measuring the air flow rate in this case was difficult to obtain a good estimate and therefore is probably the main source of error. Published fan air flow rates were not available because the cold side fan was embedded deep within the furnace and could not be accessed without deconstructing the unit. Also, portions of the fan label were missing and the fan could not be specifically identified by model, therefore these results are the best estimate. When comparing the two flow rates to their equivalent efficiencies, the outlet air flow rate resulted in the propane always being more cost effective while the inlet air flow rate typically resulted in the biomass being more cost effective. Due to all of these reasons the 1000 ft^3/min air flow rate was assumed for the rest of the analysis.

Thermal Imaging of the Firebox

Fire box temperatures are difficult to measure with standard contact sensors. The thermal imaging camera was used to determine furnace temperatures and other heat losses from the greenhouse, furnace, and flue. Each firebox image was analyzed using FLIR ThermaCAM® researcher software. A sample picture analysis is shown in Figure 3.2. That figure shows the picture clarity and the analyses that can be performed. Using the rectangle tool, a region of interest can be created for the thermal image by visually identifying the flame or hot zones shown in Figure 3.3. These regions of interests provide the maximum, minimum and average flame temperature and standard deviation. The average temperature for each picture was then uploaded to an Excel® sheet for additional analyses. These temperatures were assumed to be firebox temperature throughout the entire test run. Once each picture's temperature was recorded, the results were plotted against the time of the picture and the corresponding flue temperature at the specific time. The results of these tests are shown in Figure 3.4.

Figure 3.2. Firebox Thermal Image (°C).

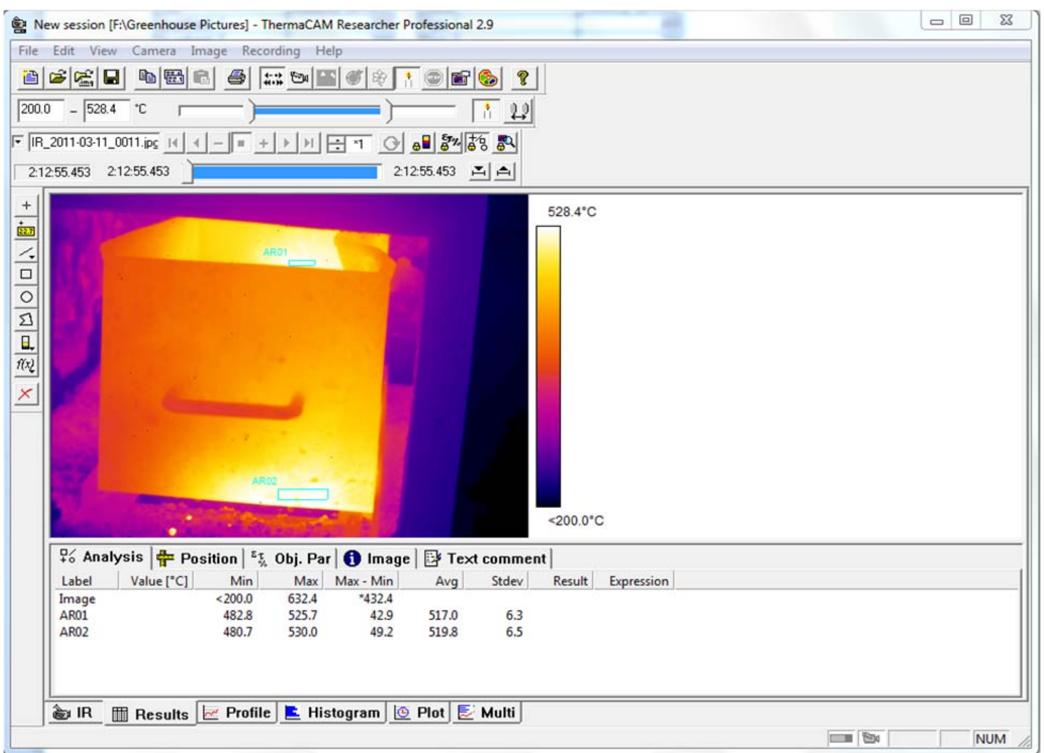


Figure 3.3. Sample FLIR Program Analysis.

Considering the trend lines shown in Figure 3.4, the firebox temperature can then be estimated for any data point during each run. Using thermal infrared images is very helpful due to the difficulty of directly measuring the firebox temperature, personal safety, and the possibility of damaging equipment and the user during firebox tests.

The ability to determine the firebox temperature also allowed the investigator to test different biomass fuels for their flame temperatures and fuel effectiveness. A power or non-linear trend line appeared to be more accurate due to the higher r^2 value. However, these trendlines are not useful outside of the heating operating range and were not used except during the heating periods.

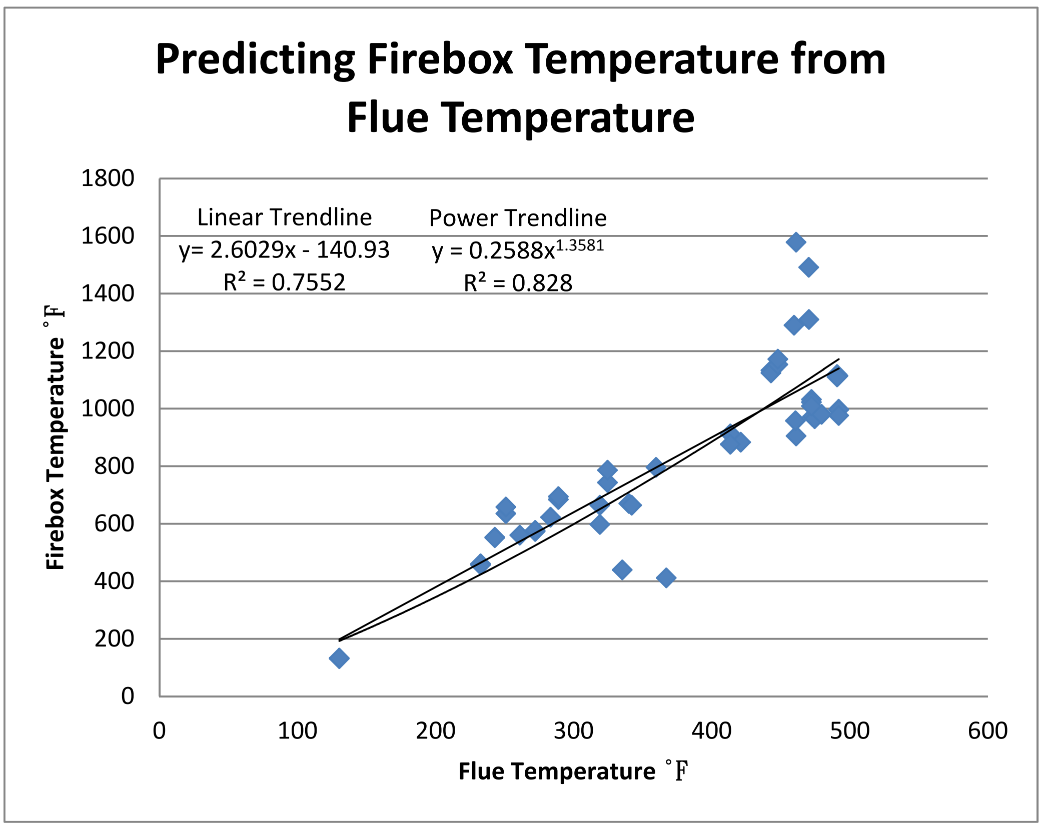


Figure 3.4. Firebox Temperature vs. Flue Temperature.

Bio-Furnace Analysis

The burner analysis was performed according to the process described above. The theoretical side of the equation was tested in two methods. The first six tests performed including the three shelled corn tests and the three shelled corn/DDGPs blend tests were conducted using the drawdown method of fuel usage. The drawdown method involved leveling the fuel in the grain bin before and after each test and measuring the amount of fuel lost volumetrically. To measure fuel used, this method was performed during the previous two years studies. This method was difficult to conduct and required a fuller supply grain bin. Due to the uncertainty of the amount of fuel used and the lower quantity of fuel needed for shorter runs, this method was scrapped and replaced by directly weighing the fuel added and remaining during each test. The remaining 18 tests used the direct weighing method. This method found much more accurate and gave a more reliable fuel used estimate. The difference in these fuels sampling methods may be a cause of the lower efficiencies observed during the initial corn tests.

Outside environmental conditions varied. Some days allowed for the same fuel to be tested in succession. On these days, enough fuel was loaded into the supply bin for two tests. The remaining fuel and ash would be weighed and the total energy content consumed for both tests was calculated. This amount of fuel was then split between individual tests based on the time period that the auger was on over the course of the entire testing period, giving each run a percent of the fuel used and ash remaining. This was done to reduce the downtime between testing, thus completing more tests during the colder part of the morning. The furnace efficiency results for each test can be seen in Figure 3.5.

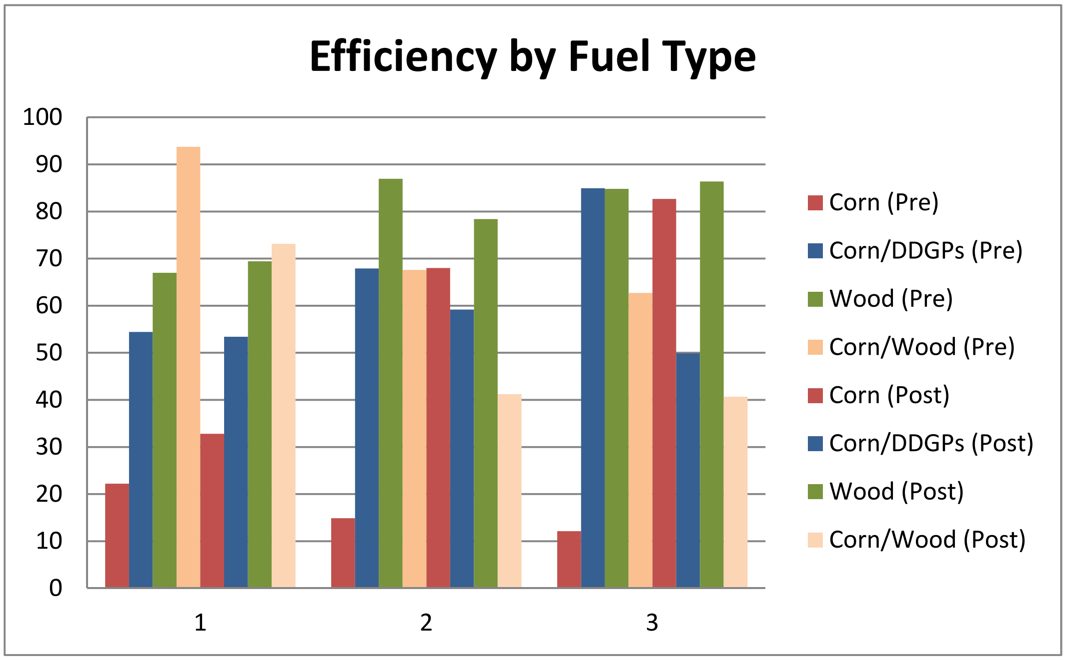


Figure 3.5. Furnace Efficiency for various bio-fuels.

A major issue with these tests is that they were performed during the morning and afternoon, because it was unfeasible to conduct an hour long test at the Firth greenhouse during the middle of night. The efficiencies found then sometimes represent the hottest part of a day. It can be assumed that the efficiencies could rise if the tests had occurred at night. The testing was conducted in the early morning to utilize what cold weather was available during the testing period. The average efficiency for each fuel type, before and after cleaning is presented in Table 3.5. The efficiencies seen in the 2011 data compare quite favorably against those data obtained during crop production in Spring 2009 and 2010 and are shown in Tables 3.6 and 3.7. These also include estimated night-time heat losses from the greenhouse.

Table 3.5. 2011 Furnace Efficiencies.

Cleaning	Fuel Type Burned Averaged	Volume of Fuel Burned lbm	Average Heat Exchanger Temperature Rise °F	Gross Energy Available Btu/hr	Burner Output Rate Btu/hr	Burner Efficiency
Pre	Corn	16.4	16.0	116000.0	16100.0	16.4
Pre	Corn/DDGPs	16.4	88.4	124000.0	89100.0	69.1
Pre	Wood Pellets	11.5	68.4	88500.0	69000.0	79.6
Pre	Corn/Wood	12.0	57.5	84000.0	58000.0	74.7
Post	Corn	16.9	69.0	120000.0	69600.0	61.2
Post	Corn/DDGPs	15.9	64.3	120000.0	64800.0	54.2
Post	Wood Pellets	8.0	48.3	62500.0	48700.0	78.1
Post	Corn/Wood	11.4	44.6	84500.0	44900.0	51.7

Table 3.6. 2010 Furnace Efficiencies.

Evaluation Period. Spring 2010 Start-Finish (From event logging)	Volume of Corn Fuel Burned - bushels	Average Exchanger Temperature Rise - °F	Gross Energy Available Btuh/hr	Burner Output Rate Btu/hr	Burner Efficiency
3/10 22:21- 3/11 15:01	17.85	66.63	128546.59	80596.37	62.70
3/11 15:01 - 3/12 15:31	18.30	80.64	131784.48	97542.14	74.02
3/12 15:31 - 3/13 10:00	12.42	53.25	89391.06	64414.83	72.06
3/13 16:00 - 3/14 20:00	12.30	55.72	88592.92	67404.23	76.08
3/14 20:00 - 3/15 19:00	12.36	63.54	89020.91	76857.98	86.34
3/15 19:00 - 3/16 19:30	11.27	54.00	81159.96	65318.40	80.48

Table 3.7 2009 Furnace Efficiencies.

Evaluation Period. Spring 2009 Start-Finish (From event logging)	Volume of Corn Fuel Burned - lbm	Average Exchanger Temperature Rise - °F	Gross Energy Available – Btu/hr	Burner Output Rate Btu/hr	Greenhouse Heat Loss Rate Btu/hr	Overall Burner Efficiency
3/12 17:06-3/13 13:48	22.54	53	154737	95292	94269	61.58
3/13 21:41-3/14 09:28	25.17	49	172753	107522	117225	62.24
3/14 20:03-3/15 08:58	26.24	53	180100	115869	115655	64.34
3/15 19:42-3/16 08:51	26.77	53	183727	115864	104922	63.06
3/16 21:32-3/17 08:40	28.73	47	197210	103993	76502	52.73
3/17 20:30-3/18 11:48	27.37	50	187858	109392	72843	58.23
3/18 18:35-3/19 12:28	22.15	51	152029	112252	77946	73.84
3/19 19:54-3/20 09:05	26.28	67	180380	145947	100210	80.91
3/20 21:23-3/21 08:28	28.91	66	198418	144781	97883	72.97
3/21 19:55-3/22 08:18	24.83	71	170419	156145	95920	91.62
3/22 19:31-3/23 07:28	22.01	40	151090	88490	68218	58.57

When the first round of tests were completed, it was noticed that the corn effciencies were not as high as they were in previous years. The average efficiency from 2011 in the first round of testing was only 16.409 percent. This was significantly lower than the efficiencies from 2009 and 2010 near the same dates. The inside of the system was then inspected to try to determine why the efficiency was lower. The results of this inspection are shown in Figures 3.6, 3.7 and 3.8.

Figure 3.6. Firebox Ash Buildup.

Figure 3.7. Fouling of Heat Exchanger Pipes.

Figure 3.8. Agitation Fans Breaks.

The inside of the firebox was found to have three years worth of ash collected on the pipes and walls of the burner. This ash was suspected to have increased the thermal resitance of the pipes and was typically around 3 to 5 mm thick. This ash layer was theorized to have caused the reduction in efficiency and heat exchange. Also, the unit was rusted over on parts of the cold side of the heat exchanger. The second half of the data collection would then focus on reperforming the twelve tests after attempting to clean the system.

The 1D ANOVA test between fuel efficiency and cleaning resulted in an F-value of 0.02 and a P-value of 0.89. These results suggest cleaning was not significant in determing the efficiency of the test. The before runs showed much higher efficiencies across the board after the corn tests. During the after cleaning tests, all fuels showed similar or lower efficiencies when compared to the precleaning tests with the exception of corn. A few new theories were created to attempt to explain the varying efficiencies noticed. These include: the inside temperature effects the efficiency, the outside temperature effects the efficiency, and that the efficiency was dependent on the date the fuel was tested.

Figure 3.9 shows the results of comparing the inside greenhouse temperature against the efficiency of the test. The data shows that burner efficiency tends to be highest around a greenhouse temperature in the range of 70 to 90 F. Also it is apparent that at really high greenhouse temperature the efficiency drops quite substantially. This makes sense due to the kill switch on the auger and the pulsing that the system undergoes at high temperatures. It was observed that when the burner exceeded the temperature of the kill switch about (85 °F) the auger would go into a pulsing state. The system did this to ensure that the fire would stay lit in the burner by adding the small pulses of raw fuel. At the same time, the ventilation in the system would attempt to kick on to lower the internal temperature. The lower efficiencies seen at the lower inside temperatures however appear to be counter intuitive. If the system could run at full speed without pulsing then why would there be a drop in efficiency. One explanation for the low efficiencies is that on the coldest days the system had to burn more fuel to create heat causing the auger to run full over the entire time period. This increased fuel would not have been fully utilized thus creating the low efficiencies. The outside temperature is also probably having an effect on these results. The results of the ANOVA test were a P-value of 0.8928 and a F-value of 0.02. This indicates there is no significance between inside temperature and efficiency.

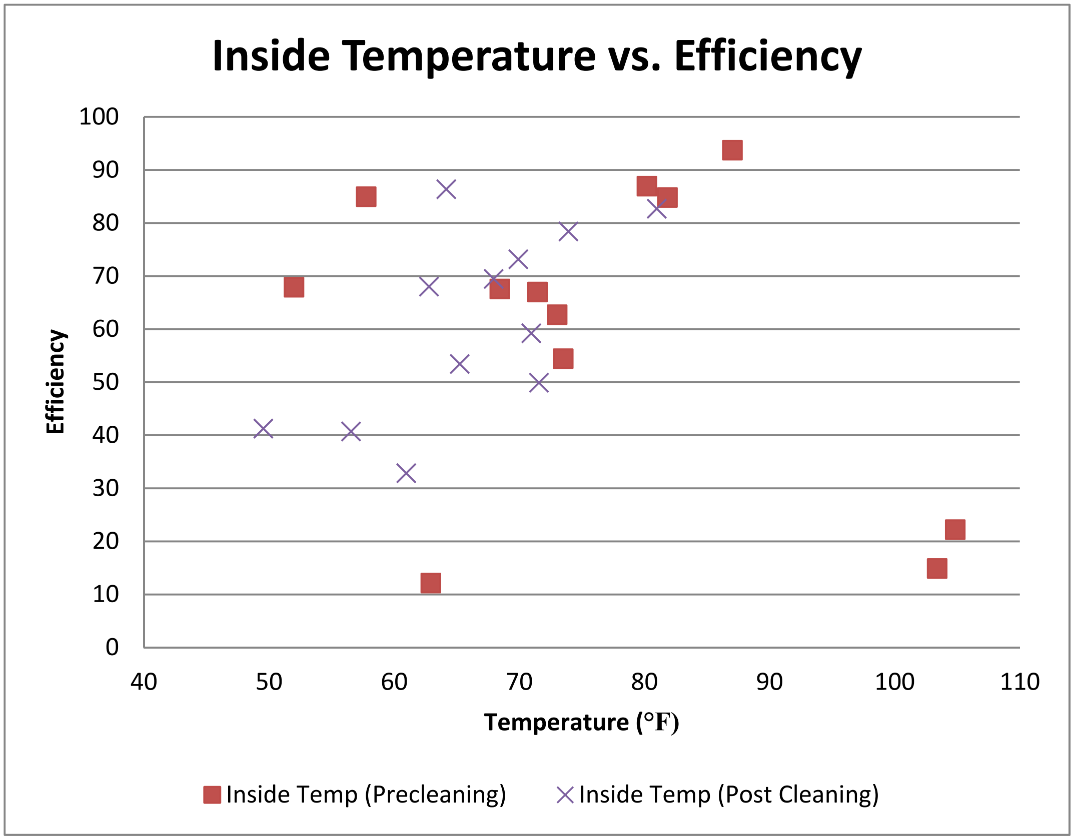


Figure 3.9. Inside Temperatures Effect on Efficiency.

The outside temperature vs. efficiency results are shown in Figure 3.10. Looking at this figure, it could be inferred that the outside temperature does not have a large impact on the results. The results shown in the figure appear random with no observable trend. The inside temperature appears to have a more significant impact on the efficiency than the outdoor temperature. The ANOVA test resulted in a P-value of 0.5217 and a F-value of 0.67. The results confirm the eye test suggesting there is no significance between efficiency and outside temperature.

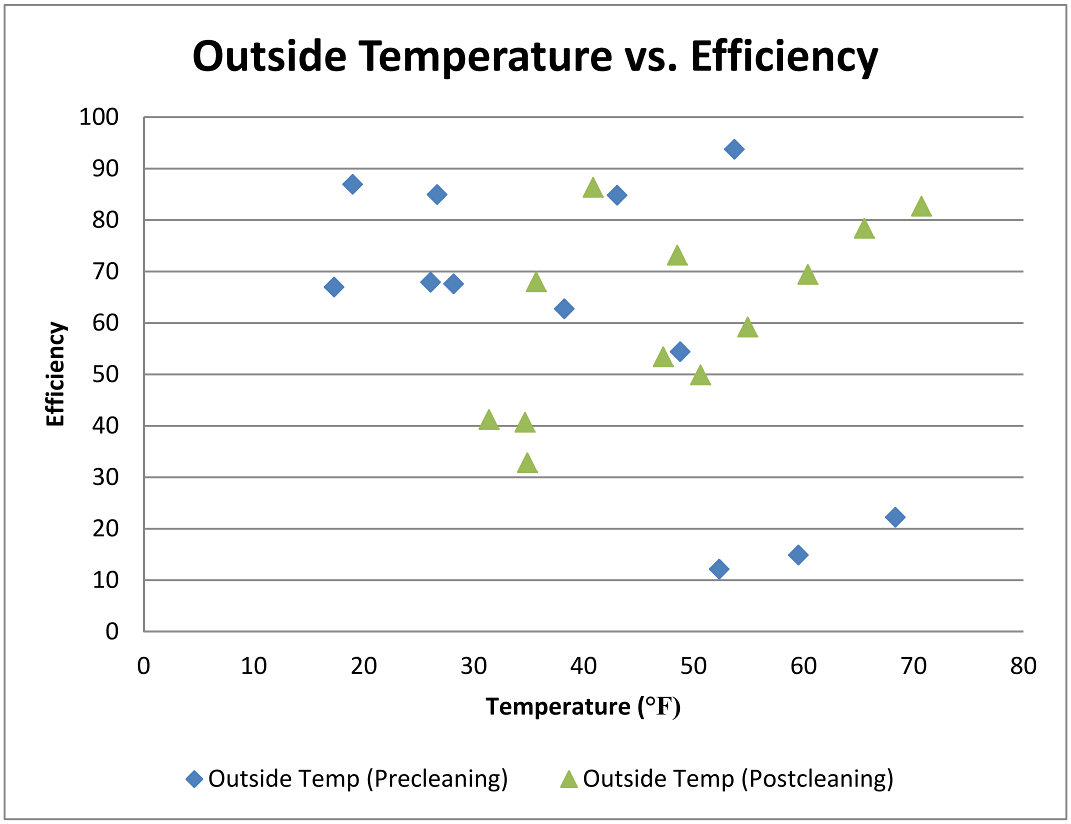


Figure 3.10. Outside Temperatures Effect on Efficiency.

Comparing the efficiencies according to the day that the test was performed showed that the efficiencies on each day were similar to the other tests that day. Those data are shown in Figure 3.11. For instance, both tests performed on March 2nd resulted in efficiencies of 63 and 68% while both tests performed on March 23rd had 50 and 73% efficiencies. Overall, there were nine days with two or more runs on the same day. Of those nine days, three had different fuel types tested. Those three days were March 1st, March 11th and March 24th. On March 1st, wood pellets and corn/wood were tested with efficiencies of 85% and 94% respectively. On March 11th , one corn test and two wood pellet tests were performed with efficiencies of 83%, 69% and 79% respectively. On March 24th wood pellets and two corn/wood tests were conducted with efficiencies of

86%, 41% and 41%. While no specific conclusions can be made from these data, it did suggest that efficiency might change somewhat by switching fuel types during a single day. Of the three days, only the 24^{th} showed significant differences between fuel type tests.

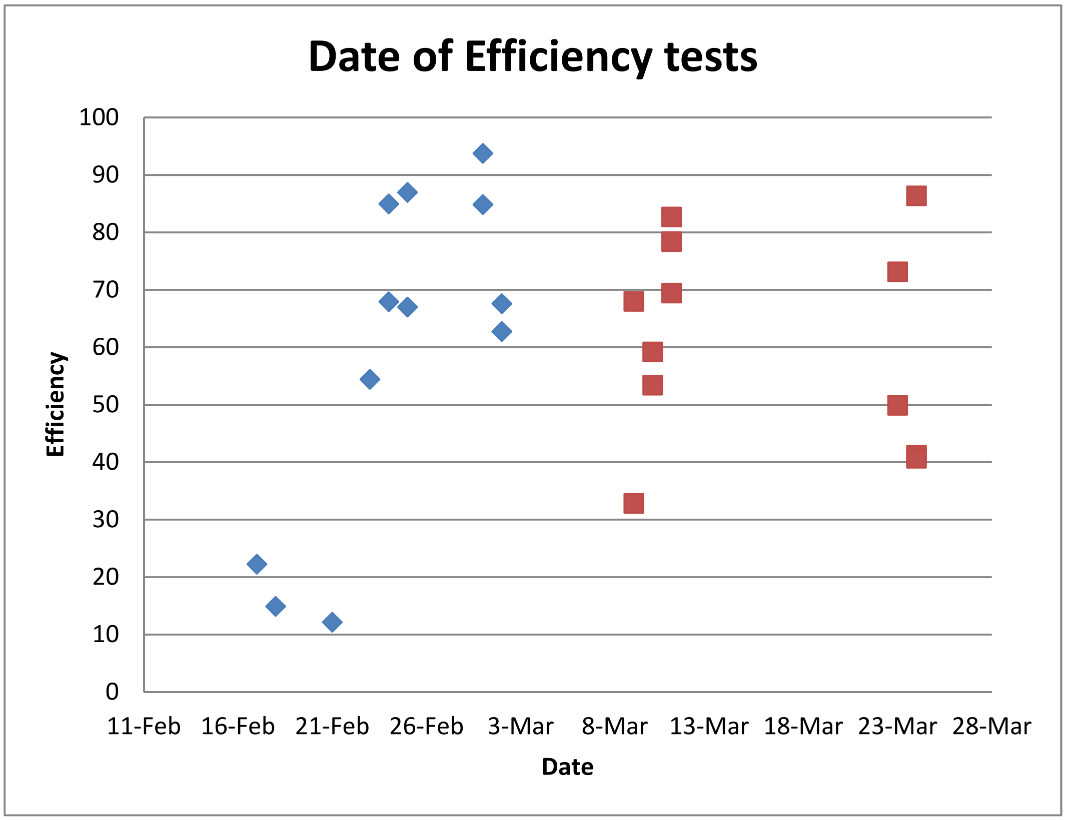


Figure 3.11. Efficiency vs. Date of Test.

The last ANOVA comparison was between efficiency and fuel type. The results of this analysis were a P-value of 0.0175 and an F-value of 4.27. This indicates there is significance between efficiency and fuel type and is the best indication of expected efficiency. The overall average efficiency of 2011 tests was 60.6 percent. This yields a Z-value of negative 0.878 when applied to equation 16. This is less than one standard deviation of change and suggests the results are reasonable compared to previous years.

Heat Exchanger Analysis

The heat exchanger analysis shows similar results as the burner efficiency analysis. The results of this analysis are shown in Table 3.8. Using the air flow rate of 1000 ft^3/min, yielded efficiencies close to 50 percent. Table 3.9 shows the average flue temperature, biomass temperature and firebox temperature for each test along with the difference between flue temperature and firebox temperature and the percent the biomass temperature represents of the firebox temperature.

Table 3.8. Average Heat Exchanger Efficiency.

Cleaning	Fuel Type Burned Averaged	Average Heat Exchanger Temperature Rise °F	Hot Side Heat Exchanger Btu/min	Cold Side Heat Exchanger Btu/min	Heat Exchanger Efficiency
Pre	Corn	16.015	2100.209	285.503	13.594
Pre	Corn/DDGPs	88.406	4097.431	1609.699	39.286
Pre	Wood Pellets	68.430	1823.632	1237.287	67.847
Pre	Corn/Wood	57.547	3666.121	1053.426	28.734
Post	Corn	68.997	2744.532	1246.795	45.428
Post	Corn/DDGPs	64.286	1677.530	1161.201	69.221
Post	Wood Pellets	48.314	1680.846	872.050	51.882
Post	Corn/Wood	44.552	1853.882	803.771	43.356

In Table 3.9, the difference between the flue temperature and firebox temperature is quite substantial. This suggests heat is being lost between the firebox and the outlet that could be recollected. Hopefully, this would be occurring through the heat exchanger but there is more duct work and more surface area after the heat exchanger where more heat loss is possible. Also, by looking at the biomass temperature as a percent of the firebox temperature, the heat transferred can be seen. During a 100% heat transfer, all of the energy created by the firebox would be transferred to the cold side, however, it was found to be about 20%. When observing the high flue temperature and large amount of heat loss in the system, there is good potential for secondary heat exchange. If secondary heat exchange was implemented using staged combustion then this could also reduce emissions.

Cost Analysis

The cost analysis included: comparing propane against whole shelled corn for a variety of fuel costs, moisture contents and efficiencies, comparing cost effectiveness of whole shelled corn with other bio-fuels, comparing whole shelled corn against natural gas, and determining a payback period for switching to a biomass burner. From 2009 to 2010, whole shelled corn varied during the year but was typically purchased around $3 to $4 per bushel while propane varied between $1.28 to $1.89 per gallon. In 2011, DDGPs were purchased at $9.32 per 50 pound bag, wood pellets cost $3.88 per 40 pound bag and whole shelled corn cost $9.29 per 50 pound bag. These prices alone suggest that buying in bulk is far more cost effective for corn, and if at all possible would be recommended for all fuel types.

Table 3.9. Average Temperatures with Heat Lost.

Date	Fuel	Average	Average		ΔT1	
	Type	Biomass Temperature	Flue Temperature	Firebox Temperature	Firebox - Flue	Biomass T/ Firebox T
		°F	°F	°F	°F	%
2/17/2011	Corn	111.8	284.2	598.9	314.6	18.7
2/18/2011	Corn	118.7	302.6	622.6	320.0	19.1
2/21/2011	Corn	88.1	480.5	1118.5	637.9	7.9
2/23/2011	Corn/DDGPs	108.2	451.7	1132.7	681.0	9.6
2/24/2011	Corn/DDGPs	168.5	445.4	1171.9	726.6	14.4
2/24/2011	Corn/DDGPs	173.1	466.7	1490.7	1024.0	11.6
2/25/2011	Wood Pellets	146.9	423.6	883.4	459.8	16.6
2/25/2011	Wood Pellets	158.6	414.2	911.8	497.6	17.4
3/1/2011	Wood Pellets	133.3	291.5	439.7	148.2	30.3
3/1/2011	Corn/Wood	115.5	179.0	132.8	-46.2	87.0
3/2/2011	Corn/Wood	151.2	458.5	1578.4	1119.9	9.6
3/2/2011	Corn/Wood	136.7	367.6	694.2	326.6	19.7
3/9/2011	Corn	108.1	376.1	972.3	596.2	11.1
3/9/2011	Corn	140.8	461.0	997.5	536.5	14.1
3/11/2011	Corn	162.9	460.8	979.9	519.0	16.6
3/10/2011	Corn/DDGPs	137.4	398.3	670.3	272.0	20.5
3/10/2011	Corn/DDGPs	134.9	349.9	795.9	446.0	17.0
3/23/2011	Corn/DDGPs	128.3	352.0	657.7	305.7	19.5
3/11/2011	Wood Pellets	114.6	285.9	576.5	290.6	19.9
3/11/2011	Wood Pellets	116.1	283.6	664.0	380.4	17.5
3/24/2011	Wood Pellets	120.3	313.2	674.3	361.1	17.8
3/23/2011	Corn/Wood	139.6	413.8	1031.7	618.0	13.5
3/24/2011	Corn/Wood	85.3	274.3	460.4	186.1	18.5
3/24/2011	Corn/Wood	84.8	235.8	560.3	324.5	15.1

A major advantage that whole shelled corn has against other biomass fuels is that it is already available in a pelletized form. Distiller's grains and wood need to be pelletized to operate in this burner and would likely need the same treatment to be usable in most all auger fed systems. This need to pelletize can dramatically increase the fuel cost and reduce the potential cost savings. In Nebraska, shelled corn is readily available and easy to obtain thus making it even more cost effective compared to other fuels.

Due to varying corn and propane costs, moisture content in corn and uncertain efficiencies of biomass burners, whole shelled corn requires a closer evaluation when comparing against propane. Figure 3.12 shows the effect of moisture content of fuel costs. While the differences are not dramatic, the higher the corn moisture content, the more expensive the propane needs to be to achieve savings. The way this figure works is that if you are above the line then corn is more cost effective, and if you are below the line then propane is more cost effective. This figure assumes an 100 percent efficiency for both propane and corn, which most likely is not the case.

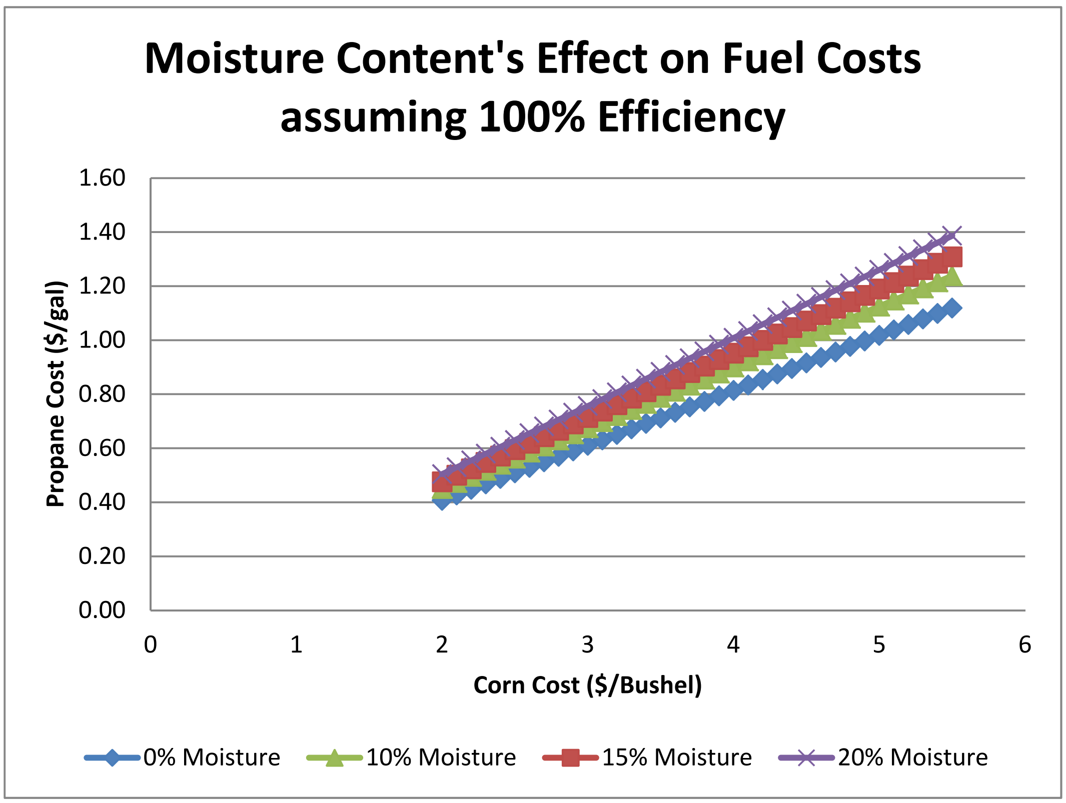


Figure 3.12. Moisture Content's Effect on Fuel Cost.

Figure 3.13 shows the effect of different efficiencies has on the equivalent cost of propane. Figure 3.13 operates on the same principle as Figure 3.12, when above the line propane is more cost effective and below the line, corn is more cost effective. The 1% efficiency shown is a worst case scenario which is quite unlikely but worth noticing. Since corn varied between $3 and $4 per bushel and propane varied between $1.30 and $1.90 per gallon, this yields a typical operating region demonstrated by the circle on Figure 3.13. Depending on which efficiency one chooses, the more favorable fuel will switch between corn and propane.

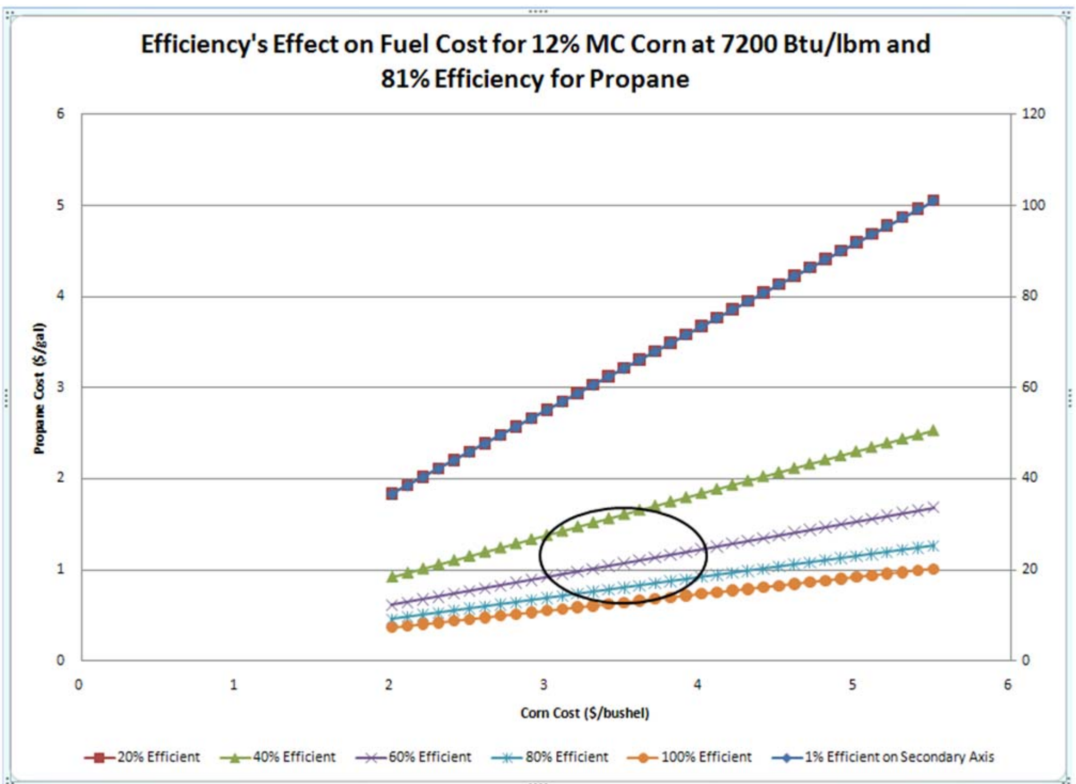


Figure 3.13. Efficiencies Effect on Equivalent Costs.

Table 3.10 shows a cost comparison between different biofuel types. Each fuel cost was translated into an effective cost per million Btu's. A quick observation shows that corn can be much cheaper but for high bushel prices for corn, wood pellets were quite compareable. This data also assume a 100% efficiency however different biofuels tested in our burner did show similar efficiencies, so those differences should be negligible. The wood pellets, DDGPs and Spring 2011 shelled corn were all purchased in small quantities. These small amounts purchased no doubt increased the cost of fuel per pound. If either fuel could be purchased in bulk or pelletized at reduced rates then, they will be more cost effective.

Table 3.10. Biomass Fuel Cost per MBtu of Energy.

Fuel	Cost	Cost/lbm	Energy Content	Cost/MMBtu
		$/lbm	Btu/lbm	$/MMBtu
Fall 07 Corn	$3.05/bu	0.05	7200.00	7.56
Spring 08 Corn	$5.35/bu	0.10	7200.00	13.27
Fall 08 Corn	$3.21/bu	0.06	7200.00	7.96
Spring 09 Corn	$3.68/bu	0.07	7200.00	9.13
Spring 10 Corn	$3.41/bu	0.06	7200.00	8.46
Spring 11 Corn	$9.29/ 50 lb	0.19	7200.00	25.81
Wood Pellets	$3.88/40 lb	0.10	7892.06	12.29
DDGPs	$9.32/50 lb	0.19	8221.54	22.67

Tables 3.11 and 3.12 show the impact efficiency and fuel cost have on the cost savings potential. Table 3.11 displays the average energy required per fuel type, the high and low efficiencies based off the high and low air flow rates and costs for each fuel. The results display a cost per test and then compare that against the costs of natural gas and propane during the same heating season this year. The percent savings are then also calculated for each fuel and efficiency. Table 3.12 then performs all of the same calculations but using a cheaper bulk corn price. The results of this analysis again show the importance of the fuel price at the time of purchase. At the $9.32/50 lb corn price, propane is substantially

more cost effective regardless of the efficiency of the biomass burner; while shelled corn at the $3.41/bu corn price, is more cost effective at the the higher efficiency. Another observation is that regardless of the circumstance, natural gas is generally more cost effective. Unfortunately, most greenhouse farmers can not connect to city natural gas lines. If the biomass burner can operate at least at 50% efficiency then corn burning could compete with propane and save the grower money. These savings can then offset by the fixed cost of purchasing the biomass burner. However, if the biomass burner is operating closer to the lower efficiencies, then biomass heating will not be able to compete with propane heating unless one of two things occur, the cost of each fuel change or the process increases its efficiency.

Table 3.11. Fuel Costs per Test at Purchase Price.

Average Results	Units	Corn	Corn/DDGPs	Wood	Corn/Wood
Energy Required	**Btu**	47636.4	87849.7	63041.8	55133.6
High Efficiency	**%**	38.8	61.6	78.8	63.2
Energy Content	**Btu/lbm**	7200.0	7710.8	7892.1	7546.0
2011 Fuel Cost	**$/lbm**	0.19	0.19	0.10	0.14
Cost Per Test	**$**	3.17	3.45	0.98	1.64
Propane assuming 81% Efficient Burner					
Energy Content	**Btu/gal**	91500.0			
2011 Fuel Cost	**$/gal**	1.87			
Equivalent Propane Cost	**$**	1.20	2.22	1.59	1.39
Percent Savings	**%**	-163.4	-55.3	38.3	-17.4
Natural Gas assuming 81% Efficient Burner					
Energy Content	**Btu/ft^3**	1028.0			
2011 Fuel Cost	**$/1000 ft^3**	5.30			
Equivalent Propane Cost	**$**	0.30	0.56	0.40	0.35
Percent Savings	**%**	-945.3	-516.3	-145.0	-366.0

Table 3.12. Fuel Cost Estimates for bulk \$3.41 per Bushel of Corn Price.

Average Results	Units	Corn	Corn/DDGPs	Wood	Corn/Wood
Energy Required	Btu	47636.4	87849.7	63041.8	55133.6
Efficiency η	%	38.8	61.6	78.8	63.2
Energy Content	Btu/lbm	7200.0	7710.8	7892.1	7546.0
\$3.41/Bu Corn Fuel Cost	\$/lbm	0.06	0.12	0.10	0.08
Cost Per Test	\$	1.04	2.29	0.98	0.91
Propane assuming 81% Efficient Burner					
Energy Content	Btu/gal	91500.0			
2011 Fuel Cost	\$/gal	1.87			
Equivalent Propane Cost	\$	1.20	2.22	1.59	1.39
Percent Savings	%	13.7	-3.0	38.3	34.4
Natural Gas assuming 81% Efficient Burner					
Energy Content	Btu/ft^3	1028.0			
2011 Fuel Cost	\$/1000 ft^3	5.30			
Equivalent Propane Cost	\$	0.30	0.56	0.40	0.35
Percent Savings	%	-242.6	-308.8	-145.0	-160.2

Emissions Analyses

The emissions test results are presented in Table 3.13. The draeger test tubes were difficult to read. The results are presented in the ranges indicated on the side of each tube. The results of the emissions statistical analysis are shown in Table 3.14. Some observable trends were noticed during this analysis including that SOx, NOx, and CO_2 increased as the firebox temperature increased. Carbon Monoxide followed a parabolic curve with the flue temperature. Lastly, PMtot was consistently about 0.1 lbm/MBtu for all tests. The flue temperature was worth plotting at the time of the emissions tests to observe the effect of increased temperature on completeness of oxidation. Increased flue temperatures correlated with increased combustion chamber temperature.

Table 3.13. Estimated Emissions Results.

	Flue Temp	Sox	NOx	CO	CO_2	PM_{tot}
Fuel Type	(°F)	(ppmV)	(ppmV)	(ppmV)	(ppmV)	(lbm/Mbtu)
Corn	550	10 +/- 5	75 +/- 25	2000 +/- 200	10000 +/- 5000	0.093
Corn	535	10 +/- 5	75 +/- 25	1300 +/- 200	40000 +/- 5000	0.085
Corn	911	125 +/- 25	125 +/- 50	225 +/- 50	90000 +/- 5000	0.080
Corn/DDGPs	867	500 +/- 100	500 +/- 100	500 +/- 100	90000 +/- 5000	0.091
Corn/DDGPs	863	200 +/- 100	400 +/- 100	200 +/- 50	80000 +/- 5000	0.099
Corn/DDGPs	876	250 +/- 100	380 +/- 100	300 +/- 50	110000 +/- 5000	0.124
Wood Pellets	793	0	0	2000 +/- 200	90000 +/- 5000	0.100
Wood Pellets	779	0	0	2200 +/-200	60000 +/- 5000	0.125
Wood Pellets	415	0	0	2000 +/- 200	20000 +/- 5000	0.091
Wood/Corn	850	20 +/- 10	100 +/- 25	1300 +/-200	50000 +/- 5000	0.103
Wood/Corn	480	10 +/- 5	75 +/- 25	1400 +/- 200	25000 +/- 5000	0.129
Corn	867	20 +/- 10	75 +/- 25	1300 +/- 200	30000 +/- 5000	0.108
Corn	887	10 +/- 5	75 +/- 25	1000 +/- 100	40000 +/- 5000	0.135
Corn	950	20 +/- 10	60 +/- 25	900 +/- 100	65000 +/- 5000	0.103
Corn/DDGPs	702	40 +/- 15	100 +/- 25	2200 +/- 200	15000 +/- 5000	0.099
Corn/DDGPs	437	30 +/- 15	45 +/- 15	1300 +/- 200	10000 +/- 5000	0.124
Corn/DDGPs	526	20 +/- 10	40 +/- 15	650 +/- 100	5000 +/- 5000	0.095
Wood Pellets	594	0	0	2200 +/- 200	10000 +/- 5000	0.100
Wood Pellets	572	0	0	2000 +/- 200	10000 +/- 5000	0.125
Wood Pellets	493	0	0	1700 +/- 200	5000 +/- 5000	0.099
Wood/Corn	796	15 +/- 5	30 +/- 15	1900 +/- 200	30000 +/- 5000	0.103
Wood/Corn	571	5 +/- 5	20 +/- 15	1400 +/- 200	5000 +/- 5000	0.103
Wood/Corn	545	10 +/- 5	40 +/- 15	2600 +/- 200	10000 +/- 5000	0.128

Table 3.14. Emissions ANOVA results.

ANOVA Analysis Results					
	CO	**CO_2**	**NOx**	**SOx**	**PMtot**
	Fuel Type				
P-value	**0.0334**	**0.3862**	**0.0039**	**0.0184**	**0.1319**
F-value	**3.53**	**1.06**	**6.14**	**4.21**	**2.1**
	Flue Temperature				
P-value	**0.0044**	**0.0001**	**0.0012**	**0.00073**	**0.0013**
F-value	**7.12**	**15.82**	**9.49**	**6.27**	**9.31**

ANOVA tests were performed for each emission against fuel type and flue temperature. The results of these tests suggest that fuel type is significant to CO, NOx and SOx. This confirms the expectation that NOx and SOx emissions would be greater for certain fuels. PM and CO_2 are not significantly different based on fuel type. This is also expected because all biomass will have these emissions. All emissions were very significantly dependent on flue temperature. This confirms the expectation that combustion temperature will more fully oxidize emissions.

Propane emissions for CO and CO_2 are similar to biomass emissions. Propane is 81 percent carbon and will emit roughly 0.00133 pounds of CO and CO_2 per pound of propane combusted (http://www.epa.gov/greenpower/pubs/calcmeth.htm). NOx and SOx emissions typically be about 75 ppm for both pollutants and should be lower than biomass (Clean Combustion Technology Part B, pg 436). Since propane is a hydrocarbon fuel, it generally has no nitrogen or sulfur except for impurities in the fuel. Most propane NOx emissions will be from atmospheric nitrogen. PM emissions should be considerably lower than biomass emissions. Figure 1.16 shows the typical total PM emissions for different heating oil's typically less than 0.02 lbs/MBtu.

Corn was observed to have some sulfur and nitrogen content as expected. Corn/DDGPs emissions typically emitted much greater levels of NOx and SOx. There are two explanations for these results. The first is that distiller's grains are concentrated corn residue from ethanol production removing most of the corn sugar (carbon) and concentrating the nitrogen and sulfur in the material. The second explanation is because in some ethanol processing plants, sulfuric acid is added to the corn to during pretreatment to break down the bonds and facilitate the process (Dipardo 2000).

The wood pellet emissions results displayed little to no SOx emissions and NOx emissions were rarely noticed. Like the corn emissions, CO_2 increased with combustion temperature. As well CO stayed consistent for different flue temperatures. This is probably not accurate but would require more testing to refute. A likely explanation is that the wood emissions require higher combustion temperatures to fully oxidize CO to CO_2 than the furnace is providing. The Corn/Wood results show the same trends as the wood pellet results. NOx and SOx were reduced from shelled corn tests, which when combined with the wood pellet results is expected.

Each emission was compared with all of the other fuel types to note overall trends regardless of fuel. Figure 3.14 shows the CO_2 emissions with an obvious upward trend as combustion temperature is increased regardless of fuel. This is not surprising because carbon becomes fully oxidized at higher combustion temperatures. Figure 3.15 shows the results of the CO emissions which display an obvious parabolic curve showing that CO peaks at low combustion temperatures and reduces as combustion increases.

The SOx and NOx emissions can be seen in Figures 3.16 and 3.17, respectively. Since these emissions were shown to be largely fuel dependent earlier, only simple conclusions can be made about each. Both show similar upward trends with increasing emission temperature. Both pollutants were noticeably greater for the corn fuels tested. Lastly PMtot emissions are presented in Figure 3.18. These results show an obvious linear trend as combustion increases. This is to be expected because more fuel fully combusts and ash generation is reduced. The energy in the fuel is utilized more effectively and efficiently.

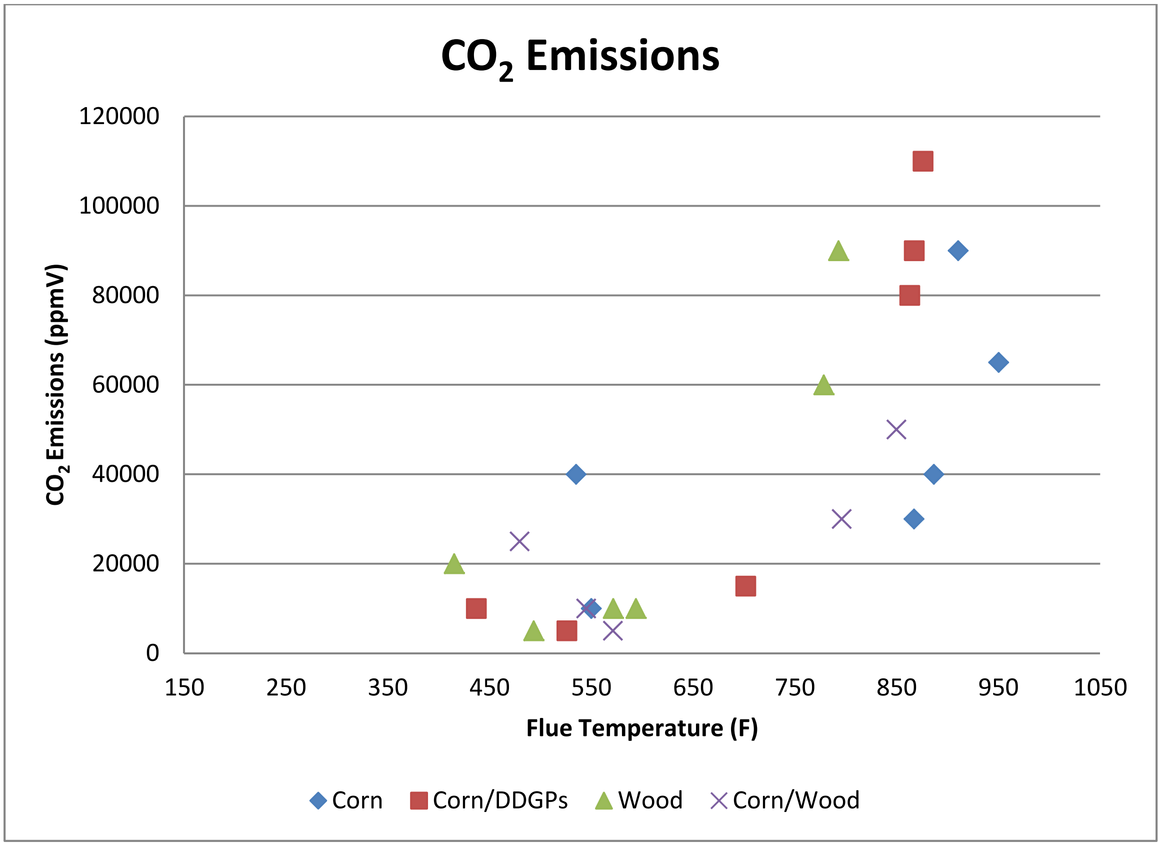


Figure 3.14. CO_2 Emissions for all Fuel Types.

Typical biomass combustion emissions were difficult to locate. Nussbaumer (2003) found wood chips NOx emissions to typically be about 200 ppmV. The June 2010 Biomass Energy Resource Center (BERC) pamphlet suggests the typical pellet stove PM emissions to be about 0.4 lbs/MBtu. The results of the emission testing showed lower emissions for NOx and PM. Also, recently the EPA decided to exclude biomass emissions from requiring regulations for at least three years (Barnard, 2011). The unit should be within typical emitting ranges and would require no regulation.

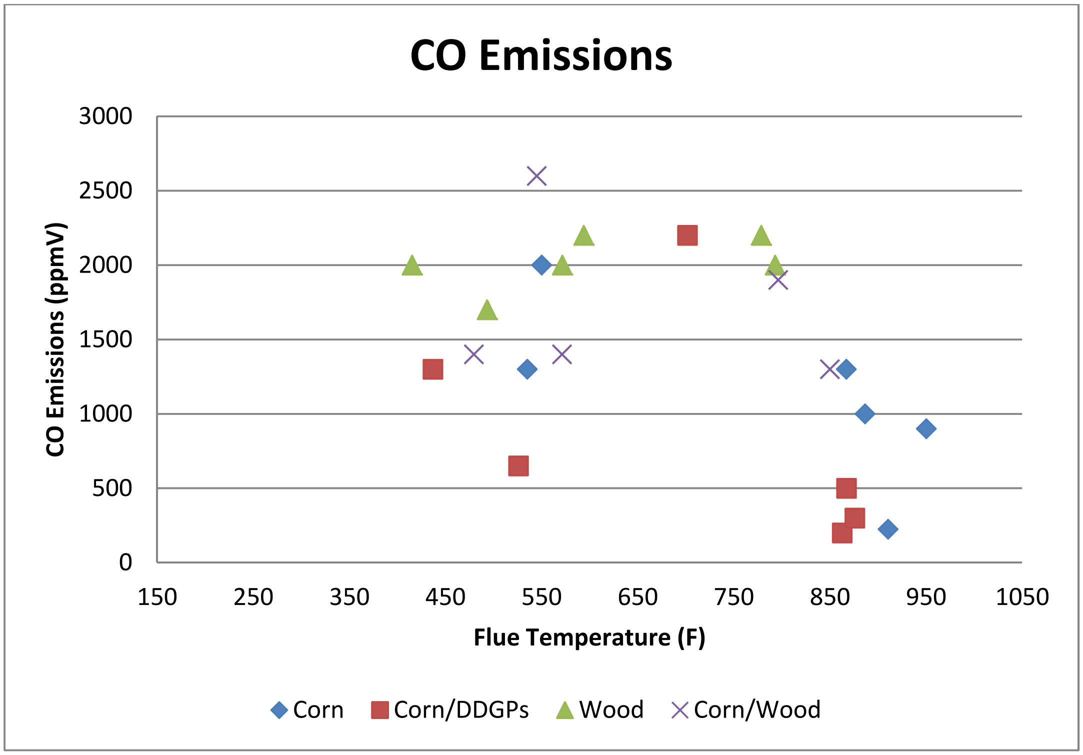


Figure 3.15. CO Emissions for all Fuel Types.

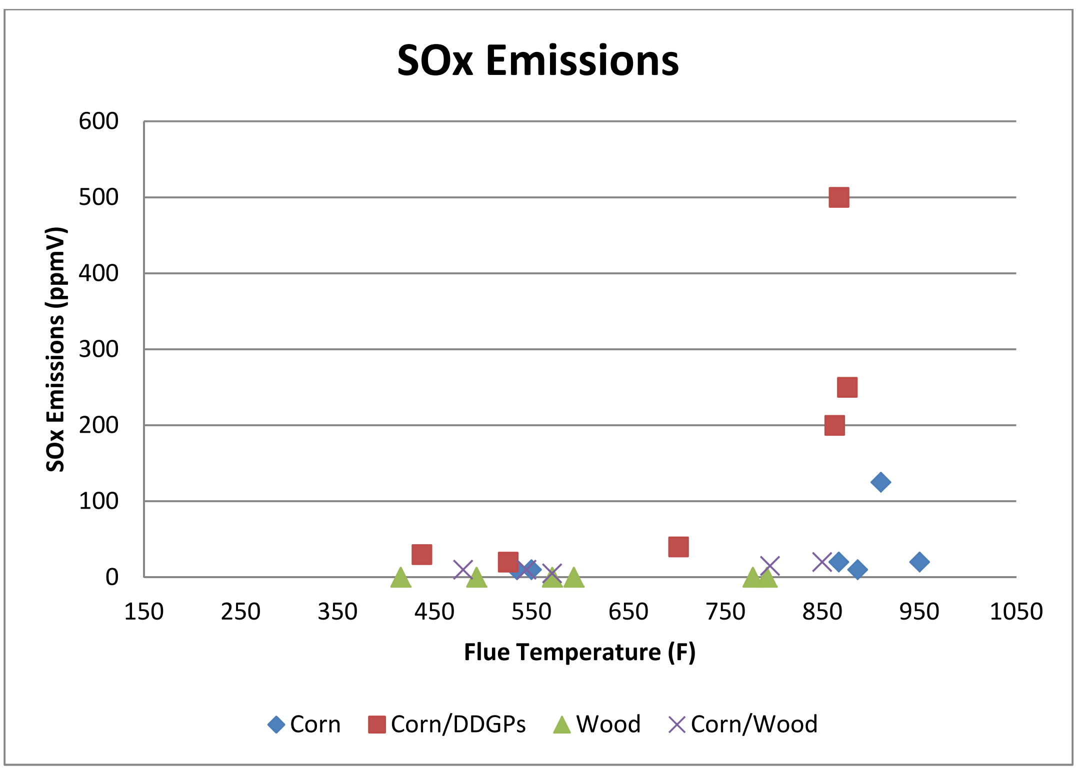


Figure 3.16. SOx Emissions for all fuels.

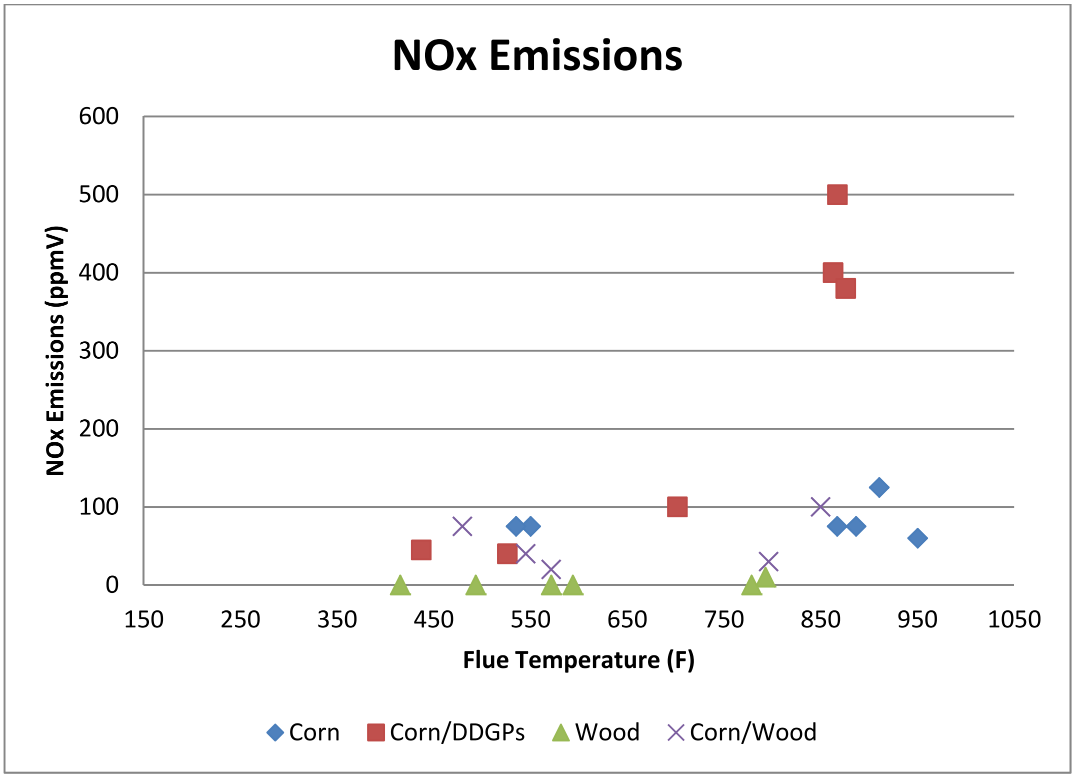


Figure 3.17. NOx Emissions for all Fuels.

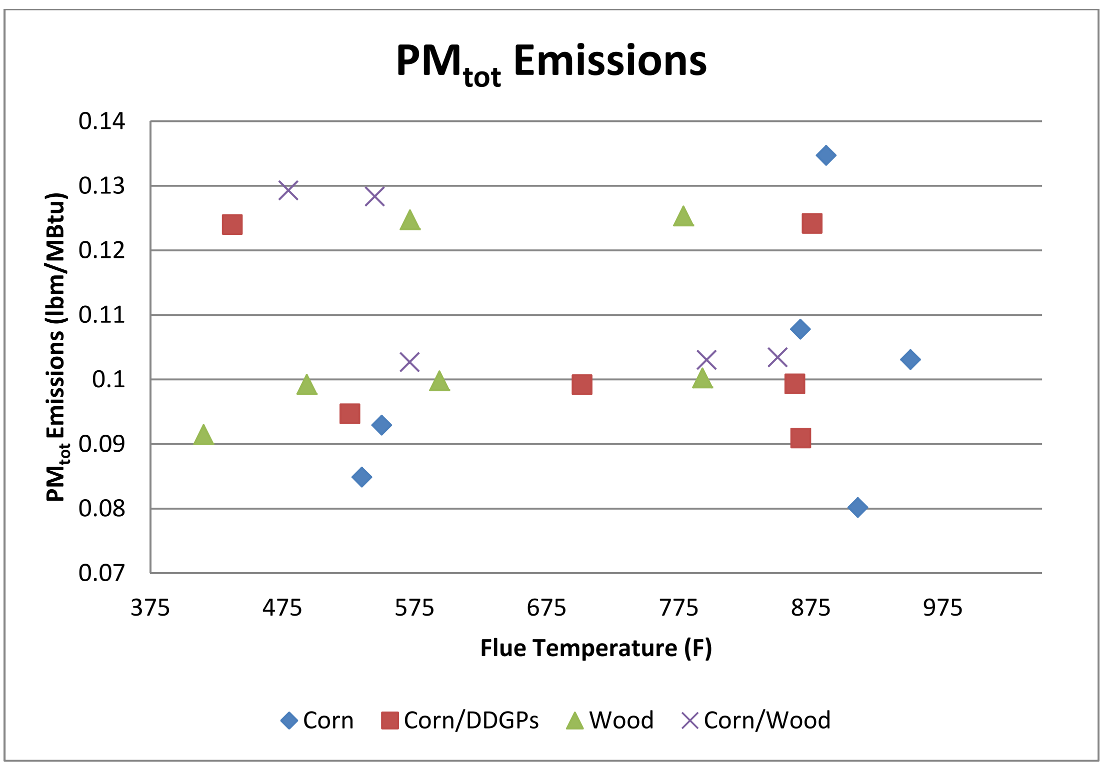


Figure 3.18. PMtot Emissions for all fuels.

Conclusions

The thermal properties of different bio-fuels and their respective ashes were determined using bomb calorimetry, moisture content tests and bulk density tests. In combination with previous research (Claussen Ucare) a broader range of biomass fuels have been scoped for future combustion utilization. The average efficiency for 2011 tests was 61% which is slightly lower than the 71% for all previous year's research. This result was less than one standard deviation different and improved the overall studies range of efficiency calculations.

Efficiencies greater than 50% generally result in corn or wood pellets being more cost effective than propane for common market prices, however, at typical natural gas rates, natural gas will be more cost effective. In rural greenhouses this is not an issue due to the inability to utilized natural gas. A grower should watch market prices and buy in bulk when costs are cheapest, specifically during non heating seasons if possible.

Several air emission samples were collected for four biomass fuels. These results were statistically analyzed and determined to be significant. These tests provide more understanding into biomass emissions, although they are furnace specific. The statistical analysis confirmed the hypothesis that emissions increase with increased combustion temperature and that corn and corn blends would emit more NOx and SOx than wood pellets.

Future accommodations for this project include continued hydrocarbon testing biomass blending research, life cycle assessment, and improved furnace controls. Hydrocarbons were not tested because they are not listed in the NAAQs. Hydrocarbon emissions are important carcinogenic compounds which need to be taken into account. Testing for these materials is important to continue improving sustainability. A fuzzy logic design model was built to attempt to improve the efficiency of the system. It was not implemented however because the environment was susceptible to a variety of different influences. Tests would need to be performed on a fully controlled environment to determine if the fuzzy logic control caused a significant change in efficiency. The biomass blends show potential as suggested by previous studies on biomass and coal blending. Two of the biomass tests showed significantly hotter firebox temperatures as seen in Appendix K. This suggests there is potential to improve heat exchange by taking advantage of the hotter flame temperatures with blending. The last suggestion is to perform a life cycle assessment of the biomass fuel being implemented. This would be useful to continue characterizing sustainability in this process.

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Appendix A. Options for Future Implementation

Fuzzy Logic Control System

LabVIEW was used for the design of the Fuzzy Logic control system. Two separate LabView programs were built. The first is a heat loss calculator which is seen in Figures A.1 and A.2. Figure A.1 shows the front panel of the program which includes several inputs and outputs. The inputs can be selected for greenhouse dimensions and material types. Running the program will calculate the total heat loss for each section of the greenhouse and finally compute the total heat loss. The block diagram, in Figure A.2, shows the math calculations which occur in the program.

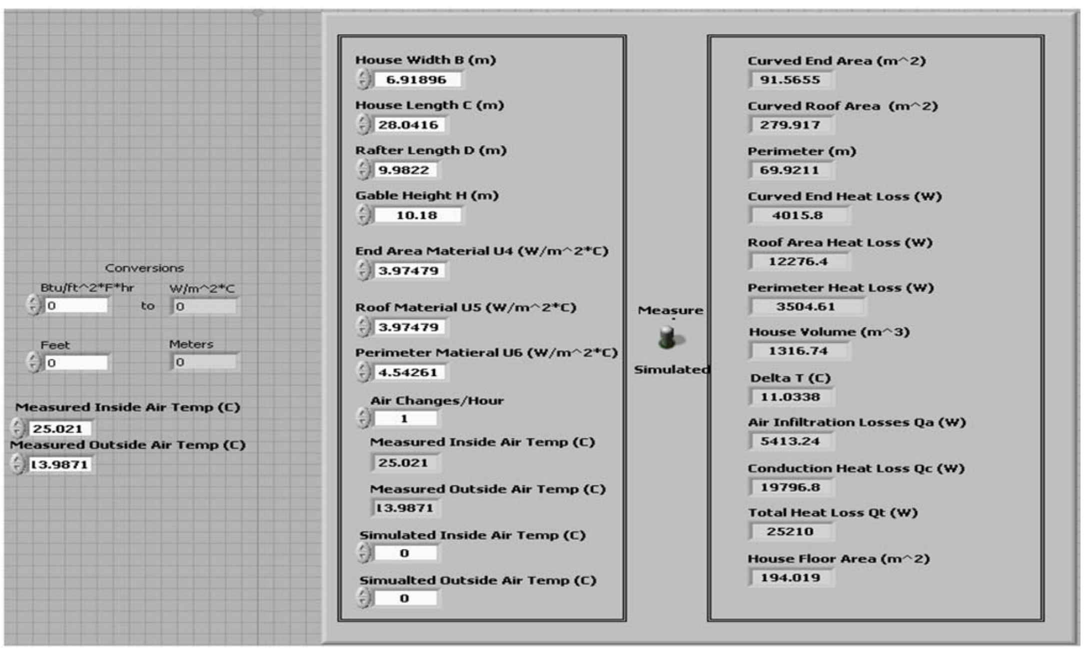

Figure A.1. Heat Loss Calculator Front Panel.

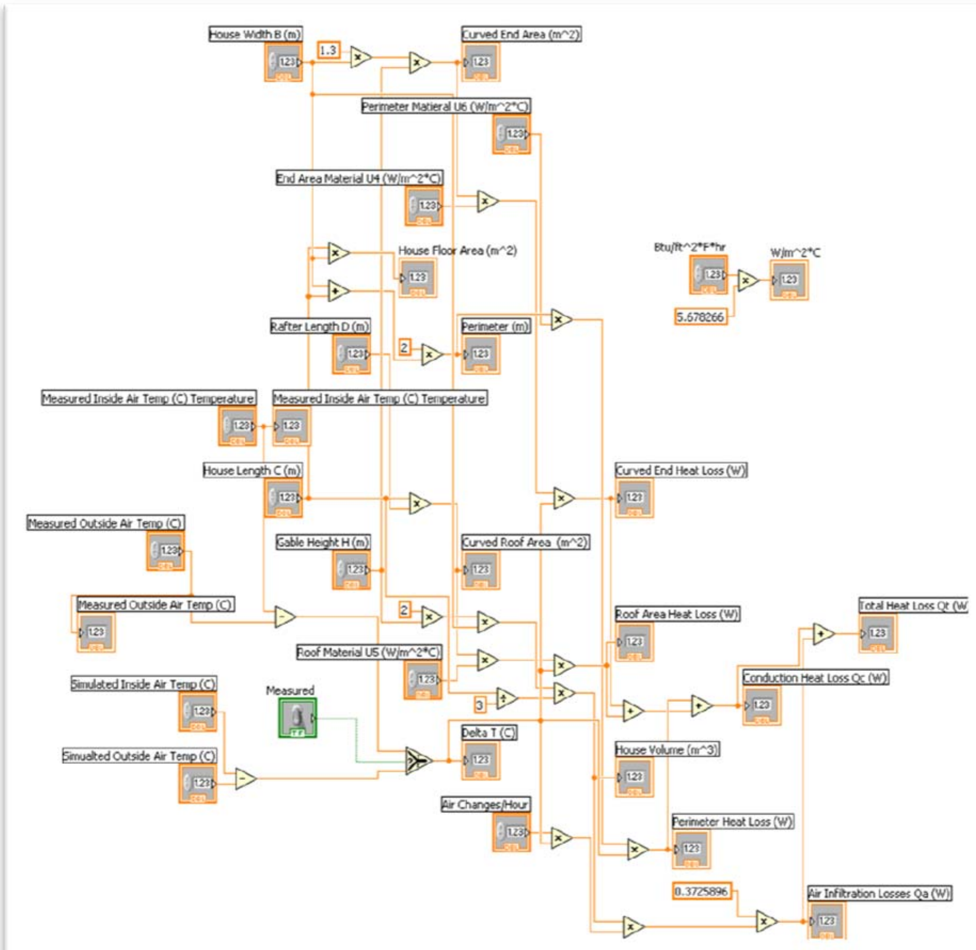

Figure A.2. Heat Loss Calculator Block Diagram.

The second LabView program is the greenhouse emulator. Figures A.3 and A.4 show the working program. The front panel in Figure A.3 allows for several input variables to be adjusted. Several of these inputs could be connected to sensors inside a greenhouse to allow onsite monitoring. The block diagram in Figure A.4 contains a few math calculations and the fuzzy system designer. There is one overriding greenhouse seen in equation 4. This was taken from Chao et al 2000. Some adjustments were made in the calculations of this program.

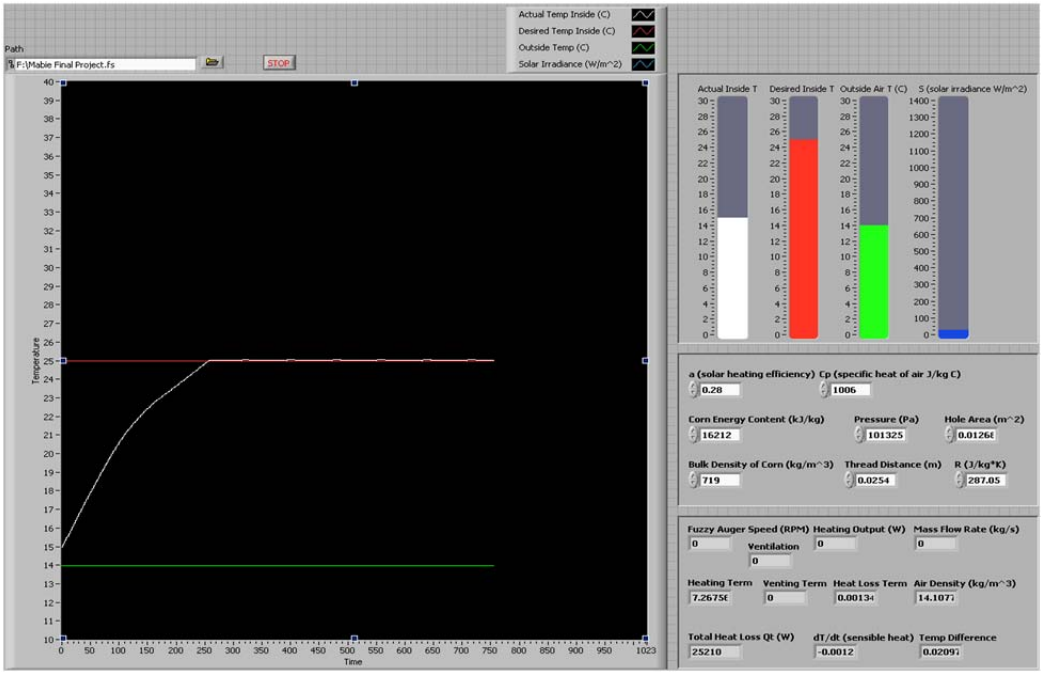

Figure A.3. Fuzzy Greenhouse Front Panel.

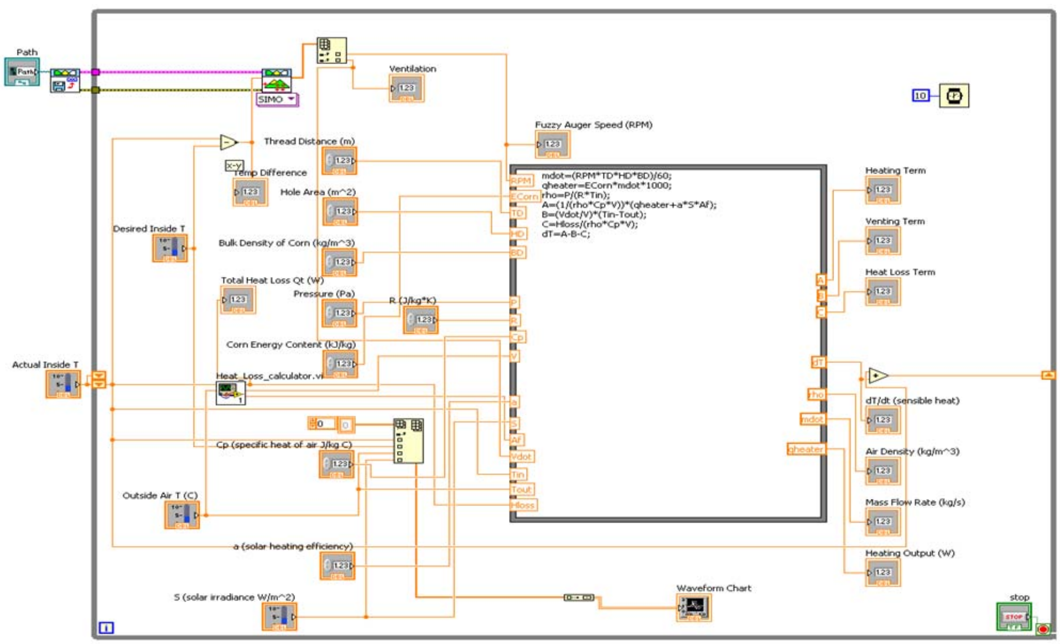

Figure A.4. Fuzzy Greenhouse Block Diagram.

The density was not assumed and was calculated from the greenhouse inside temperature by equation 4.

The heat loss term equation 4 was calculated from the heat loss VI program. Lastly the fuzzy RPM speed and ventilation rate were calculated using the fuzzy system designer. The fuzzy RPM speed was used to then calculate a heating rate by the two following equations:

Mass Flow Rate (kg/s) = RPM speed *Thread Diameter * Open Hole Area * Corn Bulk Density / 60 s/min

Qheater (W) = Mass Flow Rate * Corn Energy Content * 1000 J/kJ

The fuzzy system designer can be seen in figures A.5, A.6, A.7, and A.8. The membership functions were determined from Chao et al 2000 but with some slight modifications. The fuzzy input of temperature difference has five regions which heating occurs during and two which ventilation occurs. This was done because the goal of the project is to heat more efficiently. This allows the auger to vary its speed more and be able to slow down more effectively and ease into the desired temperature. The auger RPM speed runs off five regions of interest with even distribution from 0 to 60 RPMs. Five regions were chosen with peak membership of one at the full, ¾, ½, ¼, and 0 speeds of the maximum. The ventilation runs as an on/off system running 0, 1 or 2 fans. The rules shown in figure 10 allow either heating or venting to occur individually but not simultaneously.

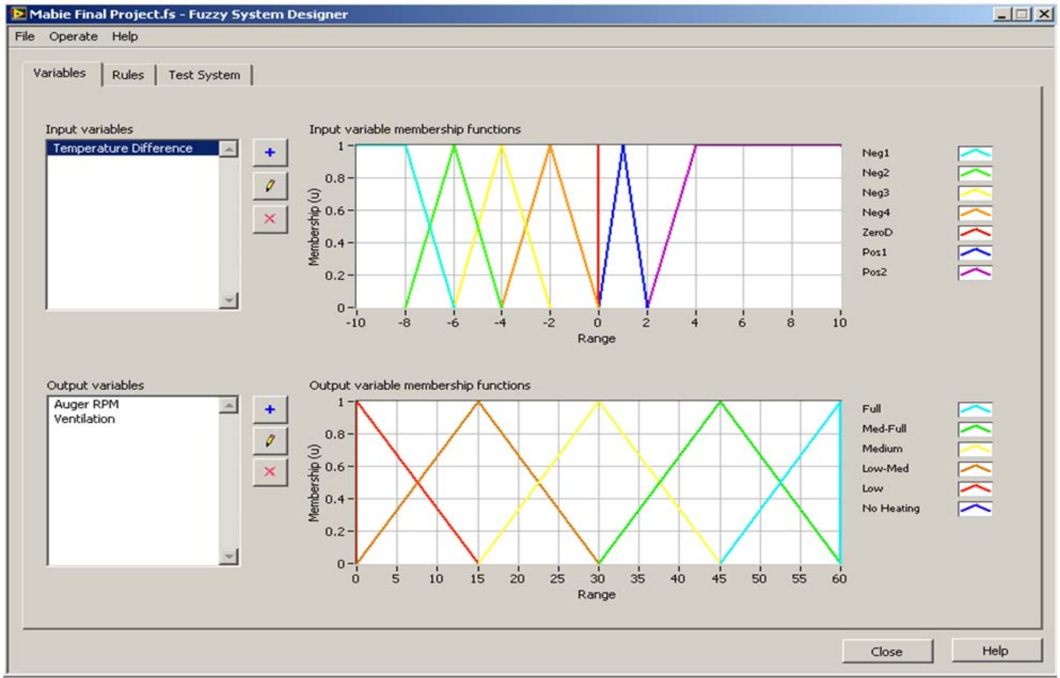

Figure A.5. Fuzzy System Membership Function for ΔT and Auger RPM.

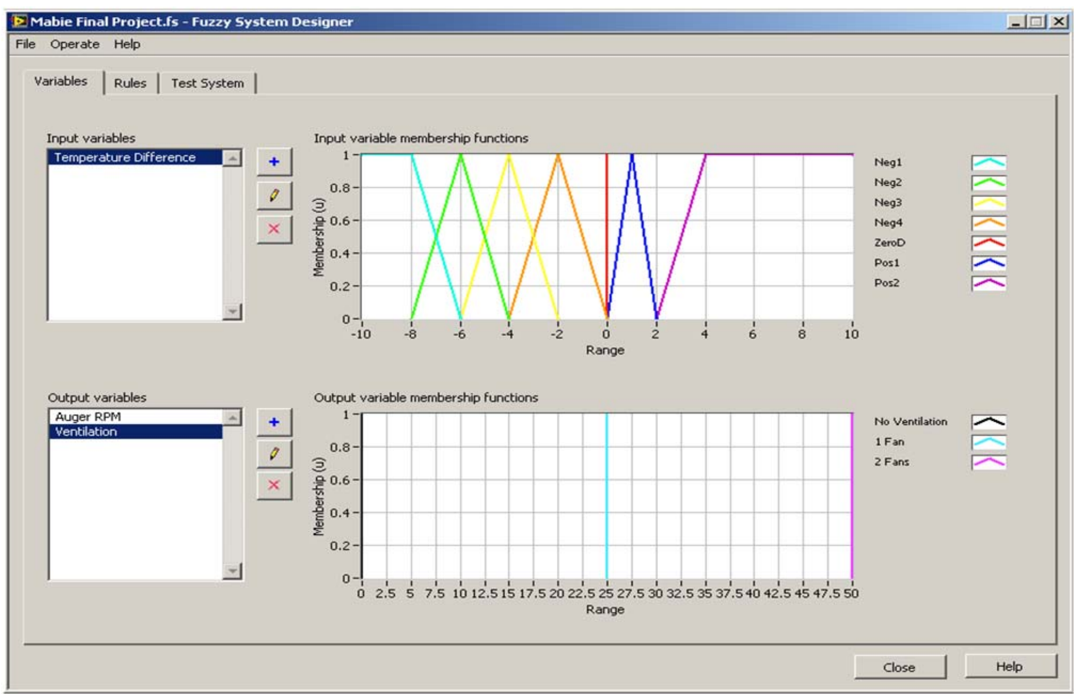

Figure A.6. Fuzzy Membership Functions for ΔT and Ventilation.

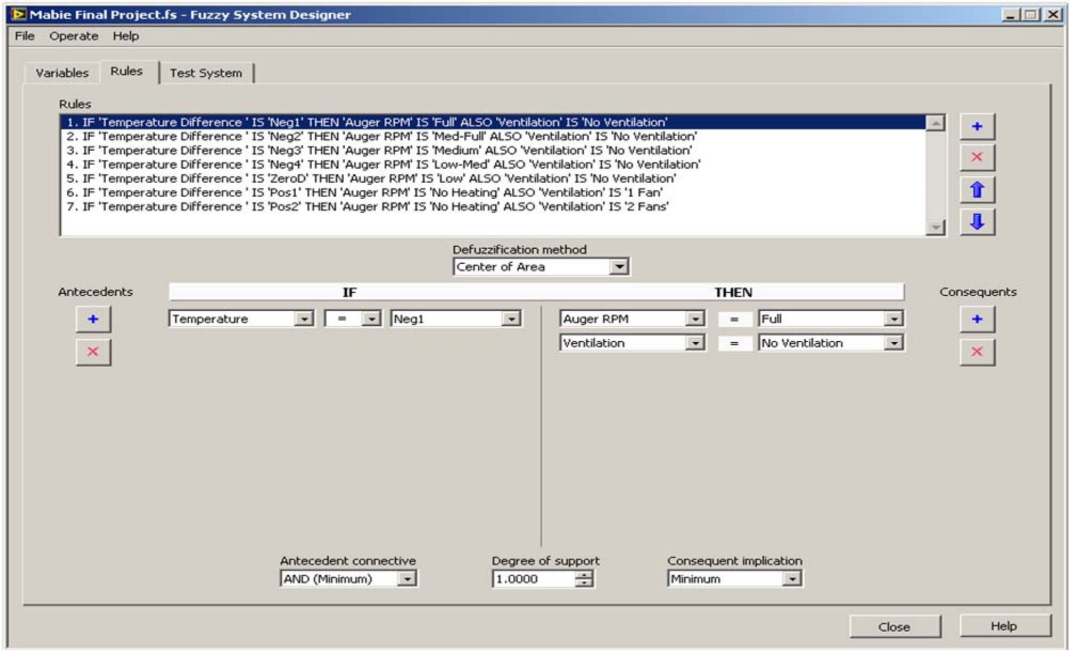

Figure A.7. Fuzzy Rules.

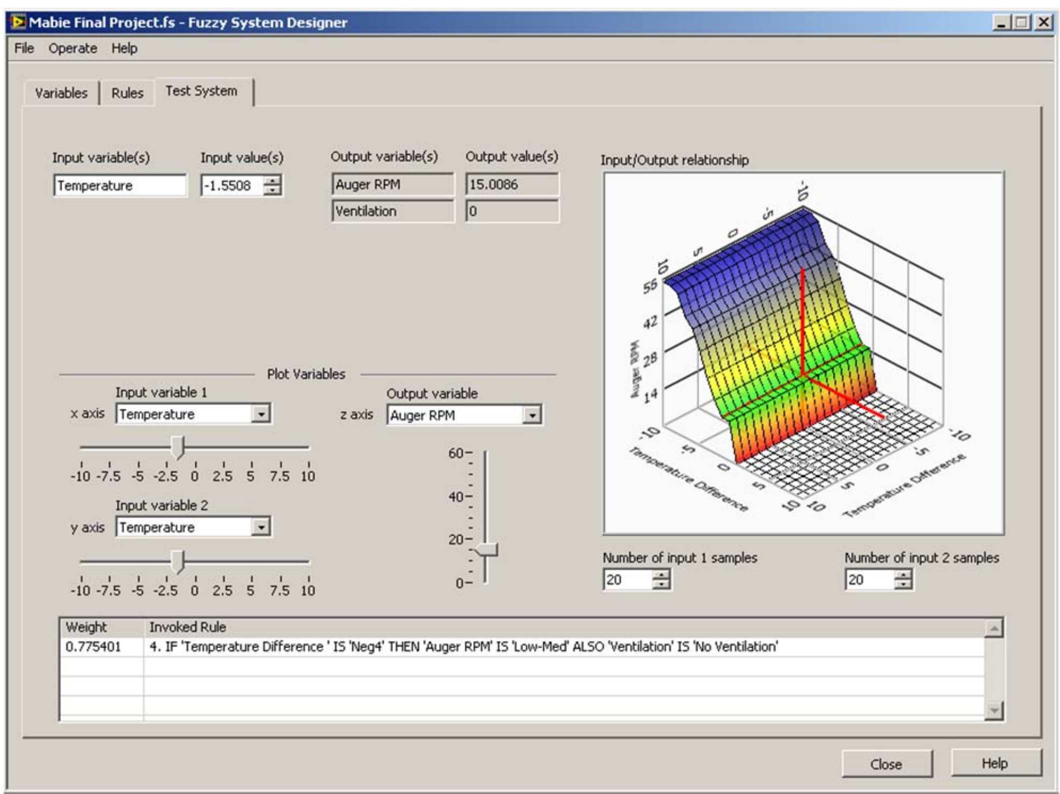

Figure A.8. Fuzzy System Test.

This program was built and compared with an on/off auger running at 60 RPMs. The results can be seen in figures A.9 and A.10. The fuzzy system results in much smaller oscillations compared to the on/off system. This is desirable because it reduces temperature variability and should result in less ash waste. The ash quantity should be reduced because less fuel is being added to the flame with each pulse because of the RPM reduction. This could allow the fire to use more of the corn fuel before a new pulse is added.

Some more modifications could be made to make the system more realistic. First the firebox temperature, ignition requirements of the corn (enthalpy of combustion and initial corn temperature), and remaining ash content should be taken into account. All of these parameters will have a significant impact on energy usage efficiency. The heat exchanger process and flue gas quality could be incorporated as well. The air humidity and greenhouse plants will impact the main greenhouse equation and heating parameters. Some of these values could be fuzzy. In this case modifying the rules and membership functions would be advisable. The ventilation process could be added as a fuzzy system too. Changing fuel types between tests could be added to the program. A LabView subvi could be created allowing the user to select fuel type with density and energy content parameters added into the system. Overall the fuzzy system designer works well to reduce variability and oscillating outputs and could be implemented into a biomass furnace.

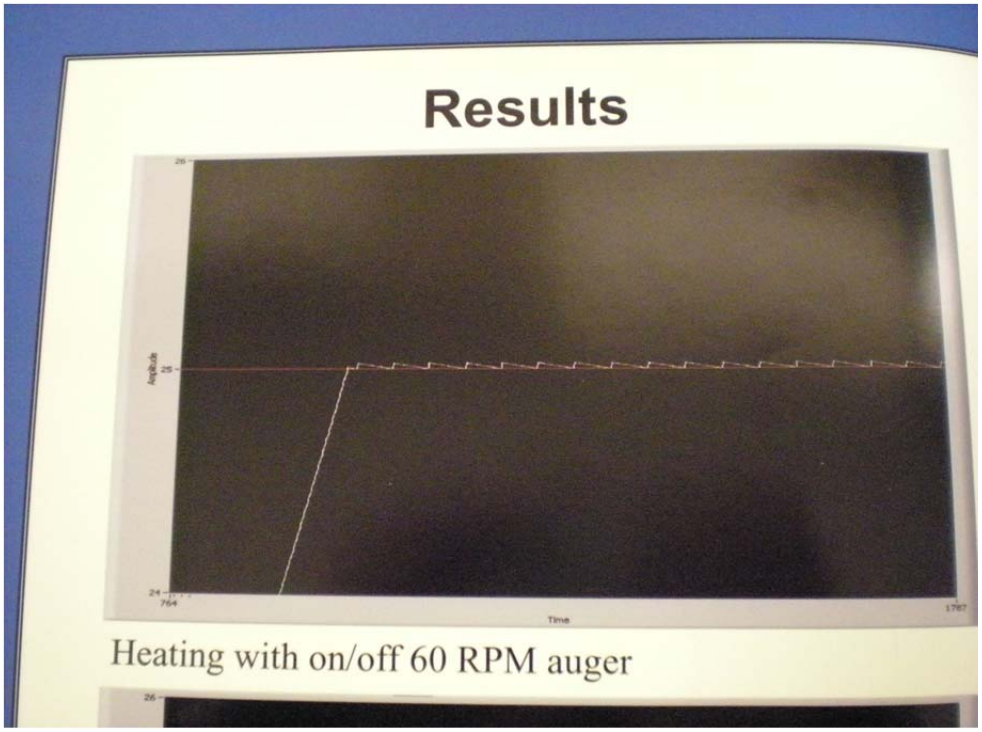

Figure A.9. Graph of Oscillations from On/Off Auger.

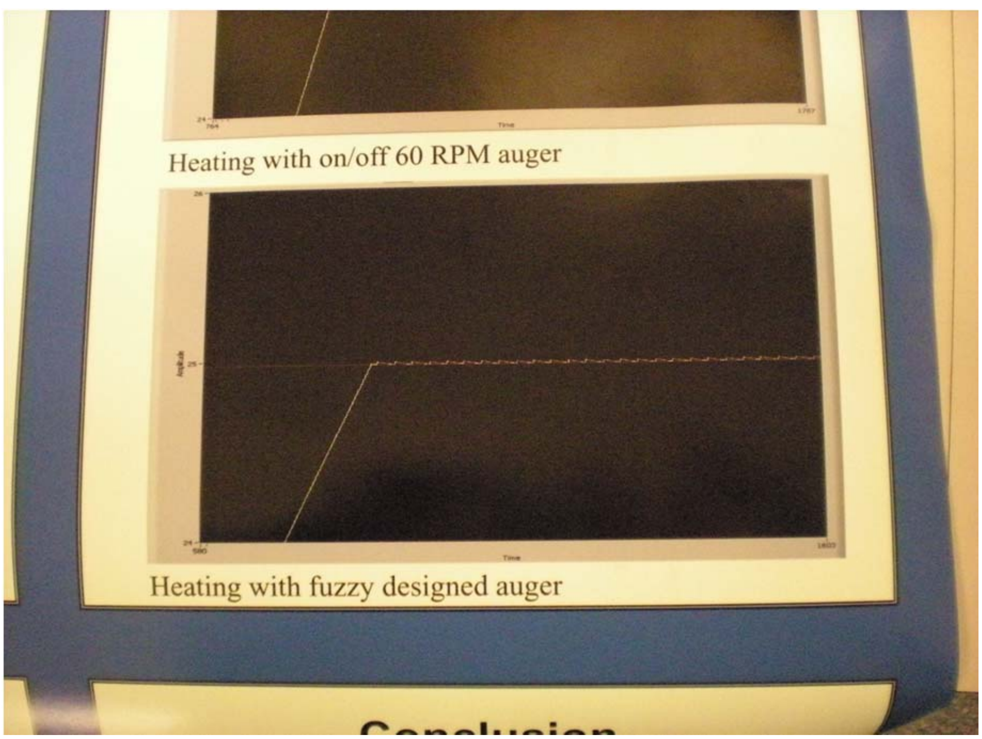

Figure A.10. Graph of Oscillations from a Fuzzy Designed Auger.

Appendix B Example Calculation of propane vs. corn cost/energy output

Assumptions:

Whole Shelled Corn energy content: 6970 BTUs/lb

Bulk Density of Whole shelled corn: 45 lbs/ft^3

Corn cost per bushel: $4.04/bushel as of 12/09

Feet cubed per bushel: 0.80356 bushels/ft^3

Corn cost per million BTU:

4.04 $/bushel x 0.80356 bushels/ft^3 x 1/45 ft^3/lbs x 1/6970 lb/BTUs x 1000000 BTUs/MBTUs

= $10.35 / MBTUs

Propane cost: 1.84 $/gallon as of 12/09

Propane energy content: 91690 BTU/gallon

Propane cost per million BTU:

1.84 $/gallon x 1/91690 gallons/BTU x 1000000 BTUs/MBTUs

= $20.07 / MBTUs

Appendix C Energy Content Test Sheet Example

Gross Energy Determinations – Liquid Fuels

Operator:____

Normality of standard solution used in acid titration: **.0725 N**

Test Date:__________		Test Date:__________		Test Date:__________	
Sample Description: __________		Sample Description: __________		Sample Description: __________	
Bucket Water Weight:__________g		Bucket Water Weight:__________g		Bucket Water Weight:__________g	
Sample mass __________g		Sample mass __________g		Sample mass __________g	
Initial Length of fuse__________cm		Initial Length of fuse__________cm		Initial Length of fuse__________cm	
Final Length of fuse__________cm		Final Length of fuse__________cm		Final Length of fuse__________cm	
Time (min)	Temp ºC	Time (min)	Temp ºC	Time (min)	Temp ºC
0		0		0	
Initial pH__________		Initial pH__________		Initial pH__________	
Initial titration reading__________ml		Initial titration reading__________ml		Initial titration reading__________ml	
Finial titration reading__________ml		Finial titration reading__________ml		Finial titration reading__________ml	
Finial pH__________		Finial pH__________		Finial pH__________	

Appendix D. Energy Content Data from NCESR 203 Final Report courtesy of Michael Claussen's UCare Research

Table D.1. Summary of Bomb Calorimetric Tests

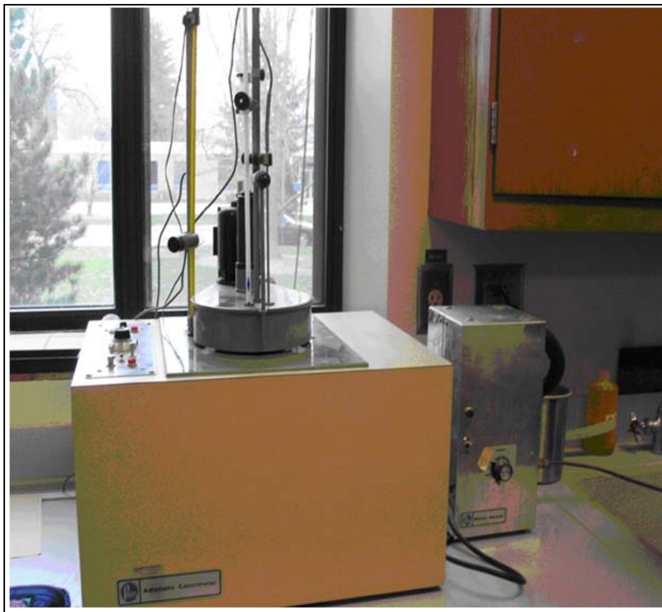

Figure D.1. Adiabatic Bomb Calorimeter.

Products Center, University of Nebraska. Samples at approximately 14% wet basis.

Fuel Type	Average Gross Heat of Combustion (BTU per lbm)
Hazelnut Shells	8,159±624
Pecan Shells	8,983±527
Shelled Corn	7,857±349
Walnut Shells	8,951±680
DDG Pellets	8,364±257
Wood Pellets	8,217±27
Ash from Greenhouse Furnace (2008)	7,044±1204
Sorghum	6,890±3

Appendix E. Propane Prices

Table E.1. Average Nebraska Residential Propane Prices 2008/2009 – 2010/2011

	Heating Season						Off Season					
	Oct	**Nov**	**Dec**	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**
2008/2009												
First Monday	$2.03	$1.69	$1.58	$1.56	$1.55	$1.49	$1.35	$1.32	$1.35	$1.31	$1.25	$1.27
Second Monday	$1.89	$1.66	$1.54	$1.56	$1.56	$1.48						
Third Monday	$1.76	$1.61	$1.55	$1.56	$1.54	$1.46						
Fourth Monday	$1.73	$1.60	$1.57	$1.57	$1.50							
Fifth Monday			$1.55									
	Heating Season						**Off Season**					
	Oct	**Nov**	**Dec**	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**
2009/2010												
First Monday	$1.29	$1.46	$1.56	$1.74	$1.80	$1.79	$1.60	$1.58	$1.52	$1.43	NA	NA
Second Monday	$1.34	$1.49	$1.59	$1.83	$1.84	$1.75						
Third Monday	$1.38	$1.50	$1.62	$1.77	$1.85	$1.71						
Fourth Monday	$1.45	$1.51	$1.70	$1.78	$1.81							
Fifth Monday		$1.52										
	Heating Season						**Off Season**					
	Oct	**Nov**	**Dec**	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**
2010/2011												
First Monday	$1.58	$1.62	$1.67	$1.73	$1.77	$1.88	NA	NA	NA	NA	NA	NA
Second Monday	$1.63	$1.64	$1.68	$1.76	$1.78	$1.86						
Third Monday	$1.63	$1.66	$1.70	$1.77	$1.77							
Fourth Monday	$1.62	$1.65	$1.72	$1.78	$1.87							
Fifth Monday		$1.67		$1.78								

Appendix F. Natural Gas Prices

Table F.1. Nebraska Natural Gas Cost per Thousand Cubic Feet.

Natural Gas Citygate Price in Nebraska (Dollars per Thousand Cubic Feet)												
Year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2003	4.78	5.19	7.38	6.16	6.42	5.82	5.89	5.61	5.73	5.63	5.31	5.68
2004	6.38	6.51	6.3	6.24	6.71	7.62	6.59	6.95	5.71	6.03	7.54	7.53
2005	6.93	7.21	7	7.87	8.09	6.9	7.4	7.06	9.36	11.21	11.07	9.87
2006	10.26	9.14	8.7	8.88	7.39	6.93	6.75	7.25	7.36	5.57	7.75	7.84
2007	7.93	8	8.34	7.69	8.16	7.7	7.52	6.6	6.06	6.27	7.44	7.5
2008	7.58	8.33	8.77	9.23	10.16	11.06	12.23	9.35	7.74	5.98	5.92	7.05
2009	7.25	7.01	6.69	5.45	3.76	4.42	4.3	4.55	4.24	4.48	5.29	5.33
2010	6.17	6.15	5.93	5.49	5.04	5.03	5.57	5.18	4.65	5.01	4.84	5.34
2011	5.3											

Appendix G. SAS Results

09:22 Friday, April 15, 2011

The ANOVA Procedure

Class Level Information

Class	Levels	Values
Cleaning	2	0 1

Number of Observations Read	24
Number of Observations Used	24

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: Efficien

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	1	10.58018	10.58018	0.02	0.8928
Error	22	12528.20966	569.46408		
Corrected Total	23	12538.78984			

R-Square	Coeff Var	Root MSE	Efficien Mean
0.000844	39.37746	23.86345	60.60179

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Cleaning	1	10.58017604	10.58017604	0.02	0.8928

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: CO

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	1	1365651.04	1365651.04	2.60	0.1209
Error	22	11543697.92	524713.54		
Corrected Total	23	12909348.96			

R-Square	Coeff Var	Root MSE	CO Mean

09:22 Friday, April 15, 2011

The ANOVA Procedure

Class Level Information

Class	Levels	Values
T In	3	0 1 2

Number of Observations Read	24
Number of Observations Used	24

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: Efficien

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	106.77017	53.38509	0.09	0.9141
Error	21	12432.01967	592.00094		
Corrected Total	23	12538.78984			

R-Square	Coeff Var	Root MSE	Efficien Mean
0.008515	40.14909	24.33107	60.60179

Source	DF	Anova SS	Mean Square	F Value	Pr > F
T In	2	106.7701719	53.3850859	0.09	0.9141

The ANOVA Procedure

Class Level Information

Class	Levels	Values
T OUT	3	0 1 2

Number of Observations Read	24
Number of Observations Used	24

The ANOVA Procedure

Dependent Variable: Efficien

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	753.41504	376.70752	0.67	0.5217
Error	21	11785.37480	561.20832		
Corrected Total	23	12538.78984			

R-Square	Coeff Var	Root MSE	Efficien Mean
0.060087	39.09098	23.68984	60.60179

Source	DF	Anova SS	Mean Square	F Value	Pr > F
T OUT	2	753.4150420	376.7075210	0.67	0.5217

09:22 Friday, April 15, 2011

The ANOVA Procedure

Class Level Information

Class	Levels	Values
Flue2	3	0 1 2

Number of Observations Read 24
Number of Observations Used 24

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: Efficien

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	46.53489	23.26745	0.04	0.9617
Error	21	12492.25495	594.86928		
Corrected Total	23	12538.78984			

R-Square	Coeff Var	Root MSE	Efficien Mean
0.003711	40.24624	24.38994	60.60179

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	46.53489319	23.26744659	0.04	0.9617

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: CO

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	5217379.71	2608689.86	7.12	0.0044
Error	21	7691969.25	366284.25		
Corrected Total	23	12909348.96			

R-Square Coeff Var Root MSE CO Mean

0.404155	44.58984	605.2142	1357.292

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	5217379.712	2608689.856	7.12	0.0044

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: CO2

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	157.4756944	78.7378472	15.82	<.0001
Error	21	104.5243056	4.9773479		
Corrected Total	23	262.0000000			

R-Square	Coeff Var	Root MSE	CO2 Mean
0.601052	59.49325	2.230997	3.750000

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	157.4756944	78.7378472	15.82	<.0001

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: NOx

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	200430.6548	100215.3274	9.49	0.0012
Error	21	221668.3036	10555.6335		
Corrected Total	23	422098.9583			

R-Square	Coeff Var	Root MSE	NOx Mean
0.474843	110.8213	102.7406	92.70833

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	200430.6548	100215.3274	9.49	0.0012

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: SOx

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	113341.7659	56670.8829	6.27	0.0073
Error	21	189757.1925	9036.0568		
Corrected Total	23	303098.9583			

R-Square	Coeff Var	Root MSE	SOx Mean
0.373943	176.1696	95.05818	53.95833

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	113341.7659	56670.8829	6.27	0.0073

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: PM

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	2	373.8633011	186.9316506	9.31	0.0013
Error	21	421.4890309	20.0709062		
Corrected Total	23	795.3523320			

R-Square	Coeff Var	Root MSE	PM Mean
0.470060	34.21338	4.480056	13.09446

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Flue2	2	373.8633011	186.9316506	9.31	0.0013

09:22 Friday, April 15, 2011

The ANOVA Procedure

Class Level Information

Class	Levels	Values
Fuel	4	0 1 2 3

Number of Observations Read	24
Number of Observations Used	24

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: Efficien

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	4892.77310	1630.92437	4.27	0.0175
Error	20	7646.01674	382.30084		
Corrected Total	23	12538.78984			

R-Square	Coeff Var	Root MSE	Efficien Mean
0.390211	32.26392	19.55251	60.60179

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	4892.773104	1630.924368	4.27	0.0175

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: CO

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	4472578.12	1490859.37	3.53	0.0334
Error	20	8436770.83	421838.54		
Corrected Total	23	12909348.96			

R-Square	Coeff Var	Root MSE	CO Mean

0.346460	47.85198	649.4910	1357.292

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	4472578.125	1490859.375	3.53	0.0334

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: CO2

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	36.0833333	12.0277778	1.06	0.3862
Error	20	225.9166667	11.2958333		
Corrected Total	23	262.0000000			

R-Square	Coeff Var	Root MSE	CO2 Mean
0.137723	89.62473	3.360927	3.750000

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	36.08333333	12.02777778	1.06	0.3862

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: NOx

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	202353.1250	67451.0417	6.14	0.0039
Error	20	219745.8333	10987.2917		
Corrected Total	23	422098.9583			

R-Square	Coeff Var	Root MSE	NOx Mean
0.479397	113.0646	104.8203	92.70833

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	202353.1250	67451.0417	6.14	0.0039

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: SOx

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	117328.1250	39109.3750	4.21	0.0184
Error	20	185770.8333	9288.5417		
Corrected Total	23	303098.9583			

R-Square	Coeff Var	Root MSE	SOx Mean
0.387095	178.6139	96.37708	53.95833

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	117328.1250	39109.3750	4.21	0.0184

09:22 Friday, April 15, 2011

The ANOVA Procedure

Dependent Variable: PM

Source	DF	Sum of Squares	Mean Square	F Value	Pr > F
Model	3	190.7513045	63.5837682	2.10	0.1319
Error	20	604.6010275	30.2300514		
Corrected Total	23	795.3523320			

R-Square	Coeff Var	Root MSE	PM Mean
0.239832	41.98865	5.498186	13.09446

Source	DF	Anova SS	Mean Square	F Value	Pr > F
Fuel	3	190.7513045	63.5837682	2.10	0.1319

Appendix H. Furnaces 2000 ASHRAE Systems and Equipment Handbook (si)

CHAPTER 28

FURNACES

RESIDENTIAL FURNACES

RESIDENTIAL furnaces are available in a variety of self-enclosed appliances that provide heated air through ductwork to the space being heated. There are two types of furnaces: (1) fuel-burning furnaces and (2) electric furnaces.

Fuel-Burning Furnaces. Combustion takes place within a combustion chamber. Circulating air passes over the outside surfaces of a heat exchanger such that it does not contact the fuel or the products of combustion, which are passed to the outside atmosphere through a vent.

Electric Furnaces. A resistance-type heating element heats the circulating air either directly or through a metal sheath enclosing the resistance element.

Residential furnaces may be further categorized by (1) type of fuel, (2) mounting arrangement, (3) airflow direction, (4) combustion system, and (5) installation location.

NATURAL GAS FURNACES

Natural gas is the most common fuel supplied for residential heating, and the central system forced-air furnace, such as that shown in Figure 1, is the most common way of heating with natural gas. This type of furnace is equipped with a blower to circulate air

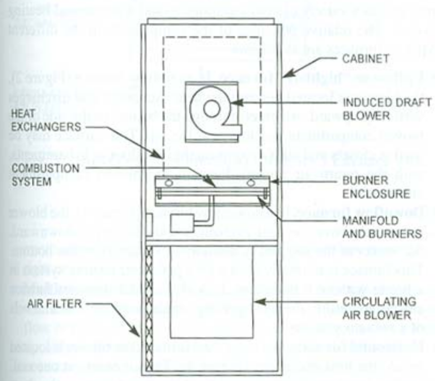

Fig. 1 Induced-Draft Gas Furnace

The preparation of this chapter is assigned to TC 6.3, Central Forced Air Heating and Cooling Systems.

through the furnace enclosure, over the heat exchanger, and through the ductwork distribution system. A typical furnace consists of the following basic components: (1) a cabinet or casing; (2) heat exchangers; (3) a combustion system including burners and controls; (4) a forced-draft blower, induced-draft blower, or draft hood; (5) a circulating air blower and motor; and (6) an air filter and other accessories such as a humidifier, an electronic air cleaner, an air-conditioning coil, or a combination of these elements.

Casing or Cabinet

The furnace casing is most commonly formed from painted cold-rolled steel. Access panels on the front of the furnace allow access to those sections requiring service. The inside of the casing adjacent to the heat exchanger is lined with a foil-faced blanket insulation and/or a metal radiation shield to reduce heat losses through the casing and to limit the outside surface temperature of the furnace. On some furnaces, the inside of the blower compartment is lined with insulation to acoustically dampen the blower noise.

Heat Exchangers

Heat exchangers are normally made of mirror-image formed parts that are joined together to form a clam shell. Heat exchangers made of finless tubes bent into a compact form are also found in some furnaces. Standard indoor furnaces are generally made of cold-rolled steel. If the furnace is exposed to clean air and the heat exchanger remains dry, this material has a long life and does not easily corrode. Some problems of heat exchanger corrosion and failure have been encountered because of exposure to halogen ions in the flue gas. These problems were caused by combustion air contaminated by substances such as laundry bleach, cleaning solvents, and halogenated hydrocarbon refrigerants.

Coated or alloy steel is used in top-of-the-line models and in furnaces for special applications. Common corrosion-resistant materials include aluminized steel, ceramic-coated cold-rolled steel, and stainless steel. Furnaces certified for use downstream of a cooling coil must have corrosion-resistant heat exchangers.

Research has been done on corrosion-resistant materials for use in condensing (secondary) heat exchangers (Stickford et al. 1985). The presence of chloride compounds in the condensate can cause a condensing heat exchanger to fail, unless a corrosion-resistant material is used.

Several manufacturers produce liquid-to-air heat exchangers in which a liquid is heated and is either evaporated or pumped to a condenser section or fan-coil, which heats circulating air.

Burners and Internal Controls

Burners are most frequently made of stamped sheet metal, although cast iron is also used. Fabricated sheet metal burners may be made from cold-rolled steel coated with high-temperature paint or from a corrosion-resistant material such as stainless or aluminized steel. Burner material must meet the corrosion protection requirements of the specific application. Gas furnace burners may be of either the monoport or multiport type; the type used with a particular furnace depends on compatibility with the heat exchanger.

Furnace controls include an ignition device, a gas valve, a fan control, a limit switch, and other components specified by the manufacturer. These controls allow gas to flow to the burners when heat is required. The four most common ignition systems are (1) standing pilot, (2) intermittent pilot, (3) direct spark, and (4) hot surface ignition (ignites either a pilot or the main burners directly). The section on Technical Data has further details on the function and performance of individual control components.

Venting Components

Natural-draft indoor furnaces are equipped with a **draft hood** connecting the heat exchanger flue gas exit to the vent pipe or chimney. The draft hood has a relief air opening large enough to ensure that the exit of the heat exchanger is always at atmospheric pressure. One purpose of the draft hood is to make certain that the natural-draft furnace continues to operate safely without generating carbon monoxide if the chimney is blocked, if there is a downdraft, or if there is excessive updraft. Another purpose is to maintain constant pressure on the combustion system. Residential furnaces built since 1987 are equipped with a blocked vent shutoff switch to shut down the furnace in case the vent becomes blocked.

Fan-assisted combustion furnaces use a small blower to force or induce the flue products through the furnace. Induced-draft furnaces may or may not have a relief air opening, but they meet the same safety requirements regardless.

Research into common venting of natural-draft appliances (water heaters) and fan-assisted combustion furnaces shows that nonpositive vent pressure systems may operate on a common vent. Refer to manufacturers' instructions for specific information.

Direct vent furnaces use outdoor air for combustion. Outdoor air is supplied to the furnace combustion chamber by direct connections between the furnace and the outdoor air. If the vent or the combustion air supply becomes blocked, the furnace control system will shut down the furnace.

ANSI *Standard* Z21.47/CSA 2.3 classifies venting systems. Central furnaces are categorized by temperature and pressure attained in the vent and by the steady-state efficiency attained by the furnace. While ANSI *Standard* Z21.47/CSA 2.3 uses 83% as the steady-state efficiency dividing furnace categories, a general rule of thumb is as follows:

Category I—A central furnace that operates with a nonpositive vent pressure and a flue loss no less than 17%.

Category II—A central furnace that operates with a nonpositive vent pressure and a flue loss less than 17%.

Category III—A central furnace that operates with a positive vent pressure and a flue loss no less than 17%.

Category IV—A central furnace that operates with a positive vent pressure and a flue loss less than 17%.

Furnaces rated in accordance with ANSI *Standard* Z21.47/CSA 2.3 that are not direct vent are marked to show that they are in one of the four venting categories listed here.

Blowers and Motors

Centrifugal blowers with forward-curved blades of the double-inlet type are used in most forced-air furnaces. These blowers overcome the resistance of the furnace air passageways, filters, and ductwork. They are usually sized to provide the additional air requirement for cooling and the static pressure required for the cooling coil. The blower may be a direct-drive type, with the blower wheel attached directly to the motor shaft, or it may be a belt-drive type, with a pulley and V-belt used to drive the blower wheel.

Electric motors used to drive furnace blowers are usually custom designed for each furnace model or model series. Direct-drive motors may be of the shaded pole or permanent split-capacitor type. Speed variation may be obtained by taps connected to extra windings in the motor. Belt-drive blower motors are normally split-phase or capacitor-start. The speed of belt-drive blowers is controlled by adjusting a variable-pitch drive pulley.

Electronically controlled, variable-speed motors are also available. This type of motor reduces electrical consumption when operated at low speeds.

Air Filters

An air filter in a forced-air furnace removes dust from the air that could reduce the effectiveness of the blower and heat exchanger(s). Filters installed in a forced-air furnace are often disposable. Permanent filters that may be washed or vacuum cleaned and reinstalled are also used. The filter is always located in the circulating airstream ahead of the blower and heat exchanger.

Accessories

Humidifiers. These are not included as a standard part of the furnace package. However, one advantage of a forced-air heating system is that it offers the opportunity to control the relative humidity of the heated space at a comfortable level. Chapter 20 addresses various types of humidifiers used with forced-air furnaces.

Electronic Air Cleaners. These air cleaners are much more effective than the air filter provided with the furnace, and they filter out much finer particles, including smoke and pollen. Electronic air cleaners create an electric field of high-voltage direct current in which dust particles are given a charge and collected on a plate having the opposite charge. The collected material is then cleaned periodically from the collector plate by the homeowner. Electronic air cleaners are mounted in the airstream entering the furnace. Chapter 24 has detailed information on filters.

Automatic Vent Dampers. This device closes the vent opening on a draft hood-equipped natural-draft furnace when the furnace is not in use, thus reducing off-cycle losses. More information about the energy-saving potential of this accessory is included in the section on Technical Data.

Airflow Variations

The components of a gas-fired, forced-air furnace can be arranged in a variety of configurations to suit a residential heating system. The relative positions of the components in the different types of furnaces are as follows:

- **Upflow or "highboy" furnace.** In an upflow furnace (Figure 2), the blower is located beneath the heat exchanger and discharges vertically upward. Air enters through the bottom or the side of the blower compartment and leaves at the top. This furnace may be used in closets and utility rooms on the first floor or in basements, with the return air ducted down to the blower compartment entrance.
- **Downflow furnace.** In a downflow furnace (Figure 3), the blower is located above the heat exchanger and discharges downward. Air enters at the top and is discharged vertically at the bottom. This furnace is normally used with a perimeter heating system in a house without a basement. It is also used in upstairs furnace closets and utility rooms supplying conditioned air to both levels of a two-story house.
- **Horizontal furnace.** In a horizontal furnace, the blower is located beside the heat exchanger (Figure 4). The air enters at one end, travels horizontally through the blower and over the heat exchanger, and is discharged at the opposite end. This furnace is used for locations with limited head room such as attics and crawl spaces, or is suspended under a roof or placed above a suspended ceiling. These units are often designed so that the components

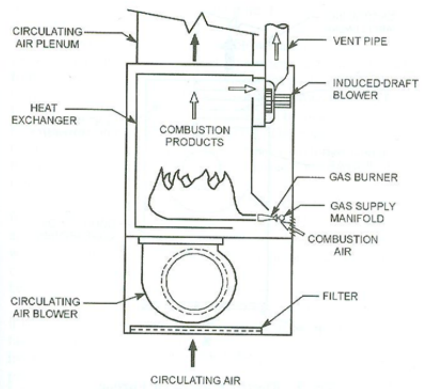

Fig. 2 Upflow Category I Furnace with Induced-Draft Blower

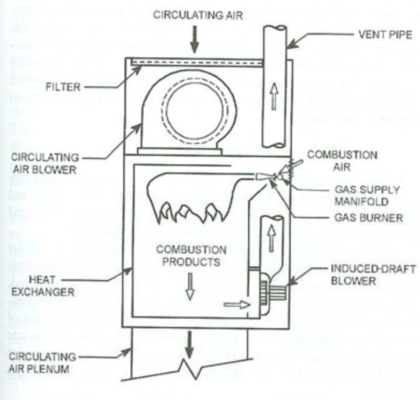

Fig. 3 Downflow (Counterflow) Category I Furnace with Induced-Draft Blower

may be rearranged to allow installation with airflow from left to right or from right to left.

- **Multiposition furnace.** A furnace that can be installed in more than one airflow configuration (e.g., upflow or horizontal; downflow or horizontal; or upflow, downflow, or horizontal) is a multiposition furnace. In some models, a field conversion is necessary to accommodate an alternate installation.
- **Basement or "lowboy" furnace.** The basement furnace (Figure 5) is a variation of the upflow furnace and requires less head room. The blower is located beside the heat exchanger at the bottom. Air enters the top of the cabinet, is drawn down through the blower, is discharged over the heat exchanger, and leaves

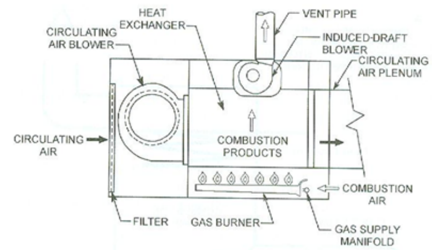

Fig. 4 Horizontal Category I Furnace with Induced-Draft Blower

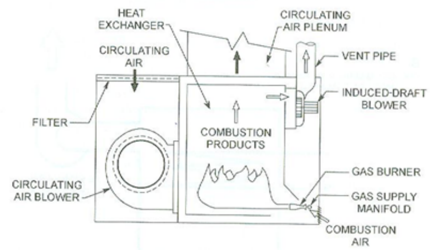

Fig. 5 Basement (Lowboy) Category I Furnace with Induced-Draft Blower

vertically at the top. In recent years, this type of furnace has become less popular due to the advent of short upflow furnaces.

- **Gravity furnace.** These furnaces are no longer available, and they are not common. This furnace has larger air passages through the casing and over the heat exchanger so that the buoyancy force created by the air being warmed circulates the air through the ducts. Wall furnaces that rely on natural convection (gravity) are discussed in Chapter 29.

Combustion System Variations

Gas-fired furnaces use a natural-draft or a fan-assisted combustion system. With a natural-draft furnace, the buoyancy of the hot combustion products carries these products through the heat exchanger, into the draft hood, and up the chimney.

Fan-assisted combustion furnaces have a combustion blower, which may be located either upstream or downstream from the heat exchangers (Figure 6). If the blower is located upstream, blowing the combustion air into the heat exchangers, the system is known as a forced-draft system. If the blower is downstream, the arrangement is known as an induced-draft system. Fan-assisted combustion systems have generally been used with outdoor furnaces; however, with the passage of the 1987 U.S. National Appliance Energy Conservation Act, fan-assisted combustion has become more common for indoor furnaces as well. Fan-assisted combustion furnaces do not require a draft hood, resulting in reduced off-cycle losses and improved efficiency.

Direct vent furnaces may have either natural-draft or fan-assisted combustion. They do not have a draft hood, and they obtain combustion air from outside the structure. Mobile home furnaces must be of the direct vent type.

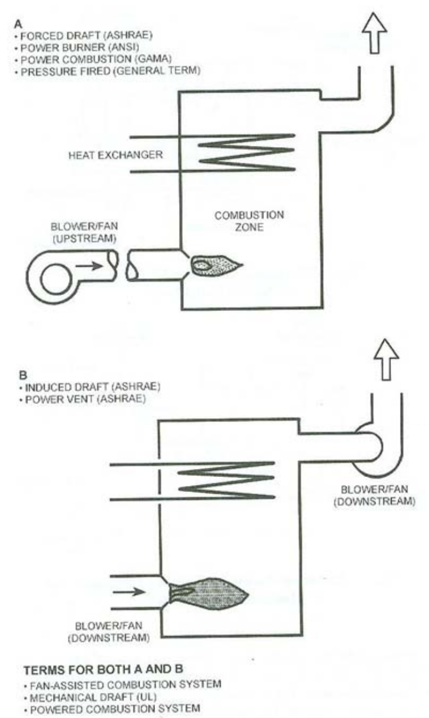

Fig. 6 Terminology Used to Describe Fan-Assisted Combustion

Indoor-Outdoor Furnace Variations

Central system residential furnaces are designed and certified for either indoor or outdoor use. Outdoor furnaces are normally horizontal flow.

The heating-only outdoor furnace is similar to the more common indoor horizontal furnace. The primary difference is that the outdoor furnace is weatherized; the motors and controls are sealed, and the exposed components are made of corrosion-resistant materials such as galvanized or aluminized steel.

A common style of outdoor furnace is the combination package unit. This unit is a combination of an air conditioner and a gas or electric furnace built into a single casing. The design varies, but the most common combination consists of an electric air conditioner coupled with a horizontal gas or electric furnace. The advantage is that much of the interconnecting piping and wiring is included in the unit.

PROPANE FURNACES

Most manufacturers have their furnaces certified for both natural gas and propane. The major difference between natural gas and propane furnaces is the pressure at which the gas is injected from the manifold into the burners. For natural gas, the manifold pressure is usually controlled at 750 to 1000 Pa; for propane, the pressure is usually 2500 to 2700 Pa.

Because of the higher injection pressure and the greater heat content per volume of propane, there are certain physical differences between a natural gas furnace and an propane furnace. One difference is that the pilot and burner orifices must be smaller for propane furnaces. The gas valve regulator spring is also different. Sometimes it is necessary to change burners, but this is not normally required. Manufacturers sell conversion kits containing both the required parts and instructions to convert furnace operation from one gas to the other.

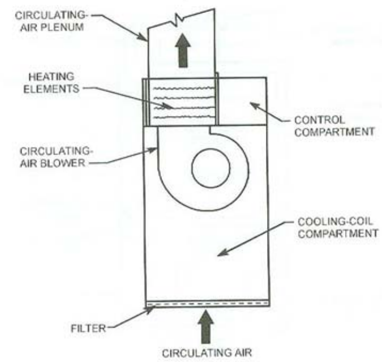

Fig. 7 Electric Forced-Air Furnace

OIL FURNACES

Indoor oil furnaces come in the same configuration as gas furnaces. They are available in upflow, downflow, horizontal, and multiposition lowboy configurations for ducted systems. Oil-fired outdoor furnaces and combination units are not common.

The major differences between oil and gas furnaces are in the combustion system, the heat exchanger, and the barometric draft regulator used in lieu of a draft hood. Ducted system, oil-fired, forced-air furnaces are usually forced-draft and equipped with pressure atomizing burners. The pump pressure and the orifice size of the injection nozzle regulate the firing rate of the furnace. Electric ignition lights the burners. Other furnace controls, such as the blower switch and the limit switch, are similar to those used on gas furnaces.

The heat exchangers of oil-fired furnaces are normally heavy-gage steel formed into a welded assembly. The hot flue products flow through the inside of the heat exchanger into the chimney. The conditioned air flows over the outside of the heat exchanger and into the air supply plenum.

ELECTRIC FURNACES

Electric-powered furnaces come in a variety of configurations and have some similarities to gas- and oil-fired furnaces. However, when a furnace is used with an air conditioner, the cooling coil may be upstream from the blower and heaters. On gas- and oil-fired furnaces, the cooling coil is normally mounted downstream from the blower and heat exchangers.

Figure 7 shows a typical arrangement for an electric forced-air furnace. Air enters the bottom of the furnace and passes through the filter, then flows up through the cooling coil section into the blower. The electric heating elements are immediately above the blower so that the high-velocity air discharging from the blower passes directly through the heating elements.

The furnace casing, air filter, and blower are similar to equivalent gas furnace components. The heating elements are made in modular form, with 5 kW capacity being typical for each module. Electric furnace controls include electric overload protection, contactor, limit switches, and a fan control switch. The overload protection may be either fuses or circuit breakers. The contactor brings the electric heat modules on. The fan control switch and limit switch functions are similar to those of the gas furnace, but one limit switch is usually used for each heating element.

Frequently, electric furnaces are made from modular sections; for example, the coil box, blower section, and electric heat section are made separately and then assembled in the field. Regardless of whether the furnace is made from a single-piece casing or a modular casing, it is generally a multiposition unit. Thus, the same unit may be used for upflow, downflow, or horizontal installations.

When an electric heating appliance is sold without a cooling coil, it is known as an electric furnace. The same appliance is called a fan-coil air handler when it has an air-conditioning coil already installed. When the unit is used as the indoor section of a split heat pump, it is called a heat pump fan-coil air handler. For detailed information on heat pumps, see Chapter 45.

Electric forced-air furnaces are also used with packaged heat pumps and packaged air conditioners.

SYSTEM DESIGN AND EQUIPMENT SELECTION

Warm-Air Furnaces

Two steps are required in selecting a warm-air furnace: (1) determining the required heating capacity of the furnace and (2) selecting a specific furnace to satisfy this requirement.

Heating Capacity. The heating capacity of a warm-air furnace depends on several variables that operate alone or in combination. The first variable is the design heating requirement of the residence. The heat loss of the structure can be calculated using the procedures outlined in Chapter 27 of the 1997 *ASHRAE Handbook—Fundamentals*.

The additional heating required if the furnace is operating on a night setback cycle should also be considered. During the morning recovery period, additional capacity is required to bring the conditioned space temperature up to the desired level. The magnitude of this recovery capacity depends on weather conditions, the magnitude of the night setback, and the time allowed for the furnace to return room air temperature to the desired level. Another consideration similar to night setback concerns structures that require only intermittent heating, such as houses of worship and auditoriums. Chapter 4 of the 1999 *ASHRAE Handbook—Applications* has further information.

A third variable is the influence of internal loads. Normally, the heat gain from internal loads is neglected when selecting a furnace, but if the internal loads are constant, they should be used to reduce the required capacity of the furnace, especially in nonresidential applications.

The energy required for humidification is a fourth variable. The humidification energy depends on the desired level of relative humidity and the rate at which the moisture must be supplied to maintain the specified level. Net moisture requirements must take into account internal gains due to people, equipment, and appliances; losses through migration in exterior surfaces; and air infiltration. Chapter 20 gives details on how to determine humidification requirements.

A fifth variable is the influence of off-peak storage devices. When used in conjunction with a furnace, a storage device decreases the required capacity of the furnace. The storage device can supply the additional capacity required during the morning recovery of a night setback cycle or reduce the daily peak loads to assist in load shedding. Detailed calculations can determine the contribution of storage devices.

The sixth variable is the furnace's capacity to accommodate air conditioning, even if air conditioning is not planned initially. The cabinet should be large enough to accept a cooling coil that satisfies the cooling load. The blower and motor should have sufficient capacity to provide the increased airflow rates typically required in air-conditioning applications. Chapter 9 includes specific design considerations.

Specific Furnace Selection. The second step in the selection of a warm-air furnace is to choose a specific furnace that satisfies the required design capacity. The final decision depends on numerous parameters, the most significant of which is the fuel type. The second step of the furnace selection process is subdivided by fuel types.

Natural Gas Furnaces

Size Selection. Historically, furnaces have been oversized because (1) the calculation procedure was not exact, especially the estimate of air infiltration; (2) a safety factor was added to account for weather conditions that are more severe than the design conditions used to calculate the required furnace capacity; (3) the additional first cost of a slightly larger furnace was considered a good value in view of possible undersizing, which would be expensive to correct; and (4) adequate airflow for cooling was another consideration. Natural gas was relatively inexpensive, so possible inefficiencies due to oversizing were not considered detrimental. The net result was significant oversizing.

Oversizing can increase overall energy use for new houses where vents and ducts are sized to furnace capacity. However, in retrofits (where fixed vent and duct sizes are assumed), oversizing has little effect on overall energy use. In either situation, oversizing may reduce the comfort level due to wide temperature variations in the conditioned space. In a retrofit, if a higher efficiency furnace is selected, the output capacity must match the original equipment's output. Otherwise, additional furnace oversizing results.

Chapter 27 of the 1997 *ASHRAE Handbook—Fundamentals* recommends oversizing new installations by 40%, if a 5 K night setback is prescribed, to obtain a 1 hour recovery.

Performance Criteria. Performance criteria or a consistent definition of efficiency must be used throughout. Some typical efficiencies encountered are (1) steady-state efficiency, (2) utilization efficiency, (3) annual fuel utilization efficiency.

These efficiencies are generally used by the furnace industry in the following manner:

- **Steady-state efficiency (SSE).** This is the efficiency of a furnace when it is operated under equilibrium conditions based on ASHRAE *Standard* 103. It is calculated by measuring the energy input, subtracting the losses for exhaust gases and flue gas condensate (for condensing furnaces only), and then dividing by the fuel input (cabinet loss not included):

$$\text{SSE}(\%) = \frac{\text{Fuel Input} - \text{Flue Loss} - \text{Condensate Loss}}{\text{Fuel Input}} \times 100$$

For furnaces tested under the isolated combustion system (ICS) method and for outdoor furnaces, cabinet heat loss (jacket loss) must also be deducted from the energy input:

$$\text{SSE(ICS)}(\%) = \frac{\text{Fuel Input} - \text{Flue Loss} - \text{Condensate Loss} - \text{Jacket Loss}}{\text{Fuel Input}} \times 100$$

An ICS is a system installed in the building structure but removed from the space it is heating. Locations include garages, attics, and crawl spaces.

A decreased flue temperature corresponds to increased SSE.

- **Utilization efficiency.** This efficiency is obtained from an empirical equation developed by Kelly et al. (1978) with 100% efficiency and deducting losses for exhausted latent and sensible heat, cyclic effects, infiltration, and pilot burner effect.
- **Annual fuel utilization efficiency (AFUE).** This value is the same as utilization efficiency, except that losses from a standing pilot during the nonheating season are deducted. This equation can also be found in Kelly et al. (1978) or ASHRAE *Standard* 103. AFUE is displayed on each furnace produced in accordance with U.S. Federal Trade Commission requirements for appliance labeling found in *Code of Federal Regulations* 16 Part 305.

  The AFUE is determined for residential fan-type furnaces by using ASHRAE *Standard* 103. The test procedure is also presented in *Code of Federal Regulations* Title II, 10 Part 430, Appendix N, in conjunction with the amendments issued by the U.S. Department of Energy in the *Federal Register*. This version of the test method allows the rating of nonweatherized furnaces as indoor combustion systems, ICSs, or both. Weatherized furnaces are rated as outdoor.

Federal law requires manufacturers of furnaces to use AFUE as determined using the isolated combustion system method to rate efficiency. Effective January 1, 1992, all furnaces produced have a minimum AFUE (ICS) level of 78%. Table 1 gives efficiency values for different furnaces.

Annual fuel energy savings may be compared using the following formula:

$$\text{Annual energy reduction (AER)} = \frac{\text{AFUE}_2 - \text{AFUE}_1}{\text{AFUE}_2}$$

where AFUE_2 is greater than AFUE_1. For example, compare items 3 and 2 of Table 1:

$$\text{AER} = \frac{78-69}{78} = \frac{9}{78} = 11.5\%$$

The ASHRAE SP43 work (see the section on System Performance in Chapter 9) confirms that this is a reasonable estimate.

Table 1 Typical Values of Efficiency

	AFUE, %	
Type of Gas Furnace	**Indoor**	**ICS[a]**
1. Natural-draft with standing pilot	64.5	63.9[b]
2. Natural-draft with intermittent ignition	69.0	68.5[b]
3. Natural-draft with intermittent ignition and auto vent damper	78.0	68.5[b]
4. Fan-assisted combustion with standing pilot or intermittent ignition	80.0	78.0
5. Same as 4, except with improved heat transfer	82.0	80.0
6. Direct vent, natural-draft with standing pilot, preheat	66.0	64.5[b]
7. Direct vent, fan-assisted combustion, and intermittent ignition	80.0	78.0
8. Fan-assisted combustion (induced-draft)	80.0	78.0
9. Condensing	90.0	88.0
Type of Oil Furnace	**Indoor**	**ICS[a]**
1. Standard—pre-1992	71.0	69.0[b]
2. Standard—post-1992	80.0	78.0
3. Same as 2, with improved heat transfer	81.0	79.0
4. Same as 3, with automatic vent damper	82.0	80.0
5. Condensing	91.0	89.0

[a]Isolated combustion system (estimate).
[b]Pre-1992 design (see text).

Construction Features and Limitations. Many indoor furnaces have cold-rolled steel heat exchangers. If the furnace is exposed to clean air, and the heat exchanger remains dry, this material has a long life and does not corrode easily. Many deluxe, noncondensing furnaces have a coated heat exchanger to provide extra protection against corrosion. Research by Stickford et al. (1985) indicates that chloride compounds in the condensate of condensing furnaces can cause the heat exchanger to fail unless it is made of specialty steel. A corrosion-resistant heat exchanger must also be used in a furnace certified for use downstream of a cooling coil.

Design Life. Typically, the heat exchangers made of cold-rolled steel have a design life of approximately 15 years. Special coated or alloy heat exchangers, when used for standard applications, have a design life of as much as 20 years. Coated or alloy heat exchangers are recommended for furnace applications in corrosive atmospheres.

Sound Level. This variable must be considered in most applications. Chapter 46 of the 1999 *ASHRAE Handbook—Applications* outlines the procedures to follow in determining acceptable noise levels.

Safety. Because of open-flame combustion, the following safety items need to be considered: (1) the surrounding atmosphere should be free of dust or chemical concentrations; (2) a path for combustion air must be provided for both sealed and open combustion chambers; and (3) the gas piping and vent pipes must be installed according to the NFPA/AGA *National Fuel Gas Code*, local codes, and the manufacturer's instructions.

Applications. Gas furnaces are primarily applied to residential heating. The majority are used in single-family dwellings but are also applicable to apartments, condominiums, and mobile homes.

Performance Versus Cost. These factors must be considered in selection. Included in life-cycle cost determination are initial cost, maintenance, energy consumption, design life, and the price escalation of the fuel. Procedures for establishing operating costs for use in product labeling and audits are available in the United States from the Department of Energy. For residential furnaces, fact sheets provided by manufacturers are available at the point of sale. In addition, AFUE (ICS), fuel, and electrical energy consumption data are listed semiannually in the Gas Appliance Manufacturers Association (GAMA) Directory.

Other Fuels

The design and selection criteria for propane furnaces are identical to those for natural gas furnaces.

The design criteria for oil furnaces are similar to those for natural gas furnaces, except that oil-fired furnaces should be tested in accordance with UL *Standard* 727 and oil burners in accordance with UL *Standard* 296.

Electric Furnaces

The design criteria for electric furnaces are similar to those for natural gas furnaces. The selection criteria are similar, except that an electric furnace does not have the flue loss and combustion air loss of a gas furnace. For this reason, and since the calculations do not account for electrical generation and transmission losses, seasonal efficiency approaches 100% for electric furnaces. Their AFUE ratings are typically 96% to 99% for ICSs.

The design life of electric furnaces is related to the durability of the contactors and the heating elements. The typical design life is approximately 15 years.

Safety primarily concerns proper wiring techniques. Wiring should comply with the *National Electrical Code* (NEC) (NFPA *Standard* 70) and applicable local codes.

TECHNICAL DATA

Detailed technical data on furnaces are available from manufacturers, wholesalers, and dealers. The data are generally tabulated in product specification bulletins printed by the manufacturer for each furnace line. These bulletins usually include performance information, electrical data, blower and air delivery data, control system information, optional equipment information, and dimensions.

Natural Gas Furnaces

Capacity Ratings. ANSI *Standard* Z21.47/CSA 2.3 requires that the heating capacity be marked on the rating plates of commercial furnaces in the United States. The heating capacity of residential furnaces, less than 65 kW input, is required by the Federal Trade Commission and can be found in furnace directories published semiannually by GAMA. Capacity is calculated by multiplying the input by the steady-state efficiency.

Residential gas furnaces with heating capacities ranging from 10 to 50 kW are readily available. Some smaller furnaces are manufactured for special-purpose installations such as mobile homes. Smaller capacity furnaces are becoming common because new homes are better insulated and have lower heat loads than older homes. Larger furnaces are also available, but these are generally considered for commercial use.

Because of the overwhelming popularity of the upflow furnace, or multiposition including upflow, it is available in the greatest number of models and sizes. Downflow furnaces, horizontal furnaces, and various combinations are also available but are generally limited in model type and size.

Residential gas furnaces are available in heating-only and heat-cool models. The difference is that the heat-cool model is designed to operate as the air-handling section of a split-system air conditioner. The heating-only models typically operate with enough airflow to allow a 35 to 55 K air temperature rise through the furnace. This rise provides good comfort conditions for the heating system with a low-noise blower and low electrical energy consumption. Condensing furnaces may be designed for a lower temperature rise (as low as 20 K).

Heat-cool model furnaces have multispeed blowers with a more powerful motor capable of delivering about 55 L/s per kilowatt of air conditioning. Models are generally available in 7, 11, 14, and 18 kW sizes, but all cooling sizes are not available for every furnace size. For example, an 18 kW furnace would be available in models with blowers capable of handling 7 or 11 kW of air conditioning; 35 kW models would be matched to 14 or 18 kW of air conditioning. Controls of the heat-cool furnace models are generally designed to operate the multispeed blower motor at the most appropriate speed for either heating or cooling operation when airflow requirements vary for each mode. This feature provides optimum comfort for year-round operation.

Efficiency Ratings. Currently, gas furnaces have steady-state efficiencies that vary from about 78 to 96%. Natural-draft and fan-assisted combustion furnaces typically range from 78 to 80% efficiency, while condensing furnaces have over 90% steady-state efficiency. Koenig (1978), Gable and Koenig (1977), Hise and Holman (1977), and Bonne et al. (1977) found that oversizing residential gas furnaces with standing pilots reduced the seasonal efficiency of heating systems in new installations with vents and ducts sized according to furnace capacity.

The AFUE of a furnace may be improved by ways other than changing the steady-state efficiency. These improvements generally add more components to the furnace. One method replaces the standing pilot with an intermittent ignition device. Gable and Koenig (1977) and Bonne et al. (1976) indicated that this feature can save as much as 5.9 GJ/year per furnace. For this reason, some jurisdictions require the use of intermittent ignition devices.

Another method of improving AFUE is to take all combustion air from outside the heated space (direct vent) and preheat it. A combustion air preheater incorporated into the vent system draws combustion air through an outer pipe that surrounds the flue pipe. Such systems have been used on mobile home and outdoor furnaces. Annual energy consumption of a direct vent furnace with combustion air preheat may be as much as 9% less than that of a standard furnace of the same design (Bonne et al. 1976). Direct vent without combustion air preheat is not inherently more efficient because the reduction in combustion-induced infiltration is offset by the use of colder combustion air.

An automatic vent damper (thermal or electromechanical) is another device that saves energy on indoor furnaces. This device, which is placed after the draft hood outlet, closes the vent when the furnace is not in operation. It saves energy during the off cycle of the furnace by (1) reducing exfiltration from the house and (2) trapping residual heat from the heat exchanger within the house rather than allowing it to flow up the chimney. These savings approach 11% under ideal conditions, where combustion air is taken from the heated space, which is under thermostat control. However, these savings are much less (estimates vary from 0 to 4%) if combustion air is taken from outside the heated space. The ICS method of determining AFUE gives no credit to vent dampers installed on indoor furnaces because it assumes the use of outdoor combustion air with the furnace installed in an unconditioned space.

The AFUE of fan-assisted combustion furnaces is higher than for standard natural-draft furnaces. Fan-assisted combustion furnaces normally have such a high internal flow resistance that combustion airflow stops when the combustion blower is off. This characteristic results in greater energy savings than those from a vent damper. Computer studies by Gable and Koenig (1977), Bonne et al. (1976), and Chi (1977) have estimated annual energy savings up to 16% for fan-assisted combustion furnaces with electric ignition as compared to natural-draft furnaces with standing pilot.

Control. Externally, the furnace is controlled by a low-voltage room thermostat. Control can be heating-only, combination heat-cool, multistage, or night setback. Chapter 37 of the 1997 *ASHRAE Handbook—Fundamentals* addresses thermostats in more detail. A night setback thermostat can reduce the annual energy consumption of a gas furnace. Dual setback (setting the temperature back during the night and during unoccupied periods in the day) can save even more energy. Gable and Koenig (1977) and Nelson and MacArthur (1978) estimated that energy savings of up to 30% are possible, depending on the degree and length of setback and the geographical location. The percentage of energy savings is greater in regions with mild climates; however, the total energy savings is greatest in cold regions.

Several types of gas valves perform various operating functions within the furnace. The type of valve available relates closely to the type of ignition device used. Two-stage valves, available on some furnaces, operate at full gas input or at a reduced rate and are controlled by either a two-stage thermostat or a software algorithm programmed in the furnace control system. They provide less heat at the reduced input and, therefore, less temperature variation and greater comfort during mild weather conditions when full heat output is not required. Two-stage control is used frequently for zoning applications. Fuel savings with two-stage firing rate systems may not be realized unless both the gas and the combustion air are controlled.

The fan control switch controls the circulating air blower. This switch may be temperature-sensitive and exposed to the circulating airstream within the furnace cabinet, or it may be an electronically operated relay. Blower start-up is typically delayed about 1 min after the start-up of the burners. This delay gives the heat exchangers time to warm up and eliminates the excessive flow of cold air when the blower comes on. Blower shutdown is also delayed several minutes after burner shutdown to remove residual heat from the heat exchangers and to improve the annual efficiency of the furnace. Constant blower operation throughout the heating season has been encouraged to improve air circulation and provide even temperature distribution throughout the house. However, constant blower operation increases electrical energy consumption and overall operating cost in many instances. Electronic motors that provide continuous but variable airflow use less energy. Both strategies may be considered when air filtering performance is important.

The limit switch prevents overheating in the event of severe reduction in circulating airflow. This temperature-sensitive switch is exposed to the circulating airstream and shuts off the gas valve if the temperature of the air leaving the furnace is excessive. The fan control and limit switches are sometimes incorporated in the same housing and are sometimes operated by the same thermostatic element. In the United States, the blocked vent shutoff switch and flame rollout switch have been required on residential furnaces produced since November 1989; they shut off the gas valve if the vent is blocked or when insufficient combustion air is present.

Furnaces using fan-assisted combustion feature a pressure switch to verify the flow of combustion air prior to the opening of the gas valve. The ignition system has a required pilot gas shutoff feature in case the pilot ignition fails.

Propane Furnaces

Most residential natural gas furnaces are also available in a propane version with identical ratings. The technical data for these two furnaces are identical, except for the gas control and the burner and pilot orifice sizes. Orifice sizes on propane furnaces are much smaller because propane has a higher density and may be supplied at a higher manifold pressure. The heating value and relative density of typical gases are listed as follows:

Gas Type	Heating Value, MJ/m^3	Relative Density (Air = 1.0)
Natural	38.4	0.60
Propane	93.1	1.53
Butane	118.3	2.00

As in natural gas furnaces, the ignition systems have a required pilot gas shutoff feature in case the pilot ignition fails. Pilot gas leakage is more critical with propane or butane gas because both are heavier than air and can accumulate to create an explosive mixture within the furnace or furnace enclosure.

Since 1978, ANSI *Standard* Z21.47/CSA 2.3 has required a gas pressure regulator as part of the propane furnace. Prior to that, the pressure regulator was provided only with the propane supply system.

Besides natural and propane, a furnace may be certified for manufactured gas, mixed gas, or propane-air mixtures; however, furnaces with these certifications are not commonly available. Mobile home furnaces are certified as convertible from natural gas to propane.

Oil Furnaces

Oil furnaces are similar to gas furnaces in size, shape, and function, but the heat exchanger, burner, and combustion control are significantly different.

Input ratings are based on the oil flow rate (L/s), and the heating capacity is calculated by the same method as that for gas furnaces. The heating value of oil is 39 MJ/L. Fewer models and sizes are available for oil than are available for gas, but residential furnaces in the range of 19 to 44 kW heating capacity are common. Air delivery ratings are similar to gas furnaces, and both heating-only and heat-cool models are available.

The efficiency of an oil furnace can drop during normal operation if the burner is not maintained and kept clean. In this case, the oil does not atomize sufficiently to allow complete combustion, and energy is lost up the chimney in the form of unburned hydrocarbons. Because most oil furnaces use power burners and electric ignition, the annual efficiency is relatively high.

Oil furnaces are available in upflow, downflow, and horizontal models. The thermostat, fan control switch, and limit switch are similar to those of a gas furnace. Oil flow is controlled by a pump and burner nozzle, which sprays the oil-air mixture into a single-chamber drum-type heat exchanger. The heat exchangers are normally heavy-gage cold-rolled steel. Humidifiers, electronic air cleaners, and night setback thermostats are available as accessories.

Electric Furnaces

Residential electric resistance furnaces are available in heating capacities of 5 to 35 kW. Air-handling capabilities are selected to provide sufficient air to meet the requirement of an air conditioner of a reasonable size to match the furnace. Small furnaces supply about 400 L/s, and large furnaces about 950 L/s.

The only loss associated with an electric resistance furnace is in the cabinet—about 2% of input. Both the steady-state efficiency and the annual fuel utilization efficiency of an electric furnace are greater than 98%, and if the furnace is located within the heated space, the seasonal efficiency is 100%.

Although the efficiency of an electric furnace is high, electricity is a relatively expensive form of energy. The operating cost may be reduced substantially by using an electric heat pump in place of a straight electric resistance furnace. Heat pump systems are discussed in Chapter 8.

Humidifiers and electronic air cleaners are available as accessories for both electric resistance furnaces and heat pumps.

Conventional setback thermostats are recommended to save energy. The electric demand used to recover from the setback, however, may be quite significant. Bullock (1977) and Schade (1978) addressed this problem by using (1) a conventional two-stage setback thermostat with staged supplemental electric heat or (2) solid-state thermostats with programmed logic to inhibit supplemental electric heat from operating during morning recovery. Benton (1983) reported energy savings of up to 30% for these controls, although the recovery time may be extended up to several hours.

Electric furnaces are available in upflow, downflow, or horizontal models. Internal controls include overload fuses or circuit breakers, overheat limit switches, a fan control switch, and a contactor to bring on the heating elements at timed intervals.

COMMERCIAL FURNACES

The basic difference between residential and commercial furnaces is the size and heating capacity of the equipment. The heating capacity of a commercial furnace may range from 44 to over 590 kW. Generally, furnaces with output capacities less than 94 kW are classified as light commercial, and those above 94 kW are considered large commercial equipment. In addition to the difference in capacity, commercial equipment is constructed from material with increased structural strength and has more sophisticated control systems.

EQUIPMENT VARIATIONS

Light commercial heating equipment comes in almost as many flow arrangements and design variations as residential equipment. Some are identical to residential equipment, while others are unique to commercial applications. Some commercial units function as a part of a ducted system, and others operate as unducted space heaters.

Ducted Equipment

Upflow Gas-Fired Commercial Furnaces. These furnaces are available up to 90 kW and supply enough airflow to handle up to 35 kW of air conditioning. They may have high static pressure and belt-driven blowers, and frequently they consist of two standard upflow furnaces tied together in a side-by-side arrangement. They are normally incorporated into a system in conjunction with a commercial split-system air-conditioning unit and are available in either propane or natural gas. Oil-fired units may be available on a limited basis.

Horizontal Gas-Fired Duct Furnaces. Available for built-up light commercial systems, this type of furnace is not equipped with its own blower but is designed for uniform airflow across the entire furnace. Duct furnaces are normally certified for operation either upstream or downstream of an air conditioner cooling coil. If a combination blower and duct furnace is desired, a package called a blower unit heater is available. Duct furnaces and blower unit heaters are available in natural gas, propane, oil, and electric models.

Electric Duct Furnaces. These furnaces are available in a large range of sizes and are suitable for operation in upflow, downflow, or horizontal positions. These units are also used to supply auxiliary heat with the indoor section of a split-heat pump.

Combination Package Units. The most common commercial furnace is the combination package unit, sometimes known as a combination rooftop unit. These are available as air-conditioning units with propane and natural gas furnaces, electric resistance heaters, or heat pumps. Combination oil heat/electric cool units are not commonly available. Combination units come in a full range of sizes covering air-conditioning ratings from 18 to 180 kW with matched furnaces supplying heat-to-cool ratios of approximately 1.5 to 1.

Combination units of 50 kW and under are available as single-zone units. The entire unit must be in either heating mode or cooling mode. All air delivered by the unit is at the same temperature. Frequently, the heating function is staged so that the system operates at reduced heat output when the load is small.

Large combination units in the 50 to 180 kW range are available as single-zone units, as are small units; however, they are also available as multizone units. A multizone unit supplies conditioned air to several different zones of a building in response to individual thermostats controlling those zones. These units are capable of supplying heating to one or more zones at the same time that cooling is supplied to other zones.

Large combination units are normally available only in a curbed configuration; that is, the units are mounted on a rooftop over a curbed opening in the roof. The supply and return air enters through the bottom of the unit. Smaller units may be available for either curbed or uncurbed mounting. In either case, the unit is usually connected to ductwork within the building to distribute the conditioned air.

Unducted Heaters

Three types of commercial heating equipment are used as unducted space heaters. One is the **unit heater**, which is available from about 7 to 94 kW. These heaters are normally mounted from ceiling hangers and blow air across the heat exchanger into the heated space. Natural gas, propane, and electric unit heaters are available. The second unducted heater used in commercial heating is the **infrared heater**. These units are mounted from ceiling hangers and transmit heat downward by radiation. Both gas and electric infrared heaters are available.

Finally, **floor (standing) furnaces** (Figure 8) are used as large area heaters and are available in capacities ranging from 60 to 590 kW. Floor furnaces direct heated air through nozzles for task heating or use air circulators to heat large industrial spaces. Residential floor-suspended furnaces are described in Chapter 29.

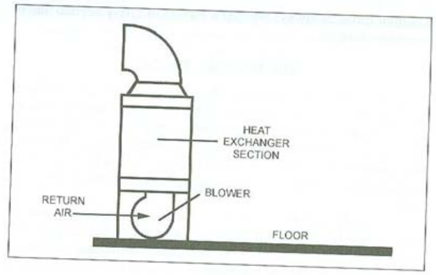

Fig. 8 Standing Floor Furnace

SYSTEM DESIGN AND EQUIPMENT SELECTION

The procedure for design and selection of a commercial furnace is similar to that for a residential furnace. First, the design capacity of the heating system must be determined, considering heat loss from the structure, recovery load, internal load, humidification, off-peak storage, waste heat recovery, and backup capacity. Because most commercial buildings use setback during weekends, evenings, or other long periods of inactivity, the recovery load is important, as are internal loads and waste heat recovery.

Selection criteria differ from those for a residential furnace in some respects and are identical in others. Sizing criteria are essentially the same, and it is recommended that the furnace be oversized 30% above total load. Because combination units must be sized accurately for the cooling load, it is possible that the smallest gas-fired capacity available will be larger than the 30% value. This is especially true for the warmer climates of the United States.

Efficiency of commercial units is about the same as for residential units. Two-stage gas valves are frequently used with commercial furnaces, but the efficiency of a two-stage system may be lower than for a single-stage system. At a reduced firing rate, the excess combustion airflow through the burners increases, decreasing the steady-state operating efficiency of the furnace. Multistage furnaces with multistage thermostats and controls are commonly used to provide more uniform distribution of heat within the building.

The design life of commercial heating and cooling equipment is about 20 years. Most gas furnace heat exchangers are either coated steel or stainless steel. Because most commercial furnaces are made for outdoor application, the cabinets are made from corrosion-resistant coated steel (e.g., galvanized or aluminized). Blowers are usually belt-driven and capable of delivering air at high static pressure.

The noise level of commercial heating equipment is important with some products and less important with others. Unit heaters, for example, are used primarily in industrial applications where noise is less important. Most other commercial equipment is used in schools, office buildings, and other commercial buildings where noise level is important. In general, the larger the furnace, the more air it handles, and the more noise it generates. However, commercial systems with longer and larger ductwork result in more sound attenuation. The net result is that quality commercial heating systems produce about the same noise level in the heated space as do residential systems.

Safety requirements are the same for light commercial systems as they are for residential systems. Above 117 kW gas input, the ANSI *Standard* Z21.47/CSA 2.3 requirements for gas controls are more stringent. A large percentage of commercial heating systems are located on rooftops or some other location outside a building.

Outdoor furnaces always provide a margin of safety beyond that of an indoor furnace.

TECHNICAL DATA

Technical data for commercial furnaces are supplied by the manufacturer. Furnaces are available with heat outputs ranging from 45 to more than 590 kW. For heating-only commercial heaters, the airflow is set to supply air with a 47 K temperature rise. Heat-cool combination units supply air equal to about 54 L/s per kilowatt of cooling capacity. Heat-to-cool ratios are generally held at about 1.5 to 1.

The steady-state efficiency for commercial furnaces is about the same as that for residential furnaces. The 1992 U.S. Energy Policy and Conservation Act (EPCA) prescribes minimum efficiency requirements for commercial furnaces based on ASHRAE *Standard* 90.1. Some efficiency improvement components, such as intermittent ignition devices, are common in commercial furnaces.

GENERAL CONSIDERATIONS

INSTALLATION PRACTICES

Installation requirements call for a forced-air heating system to meet two basic criteria: (1) the system must be safe, and (2) it must provide comfort for the occupants of the conditioned space.

Indoor furnaces are sometimes installed as isolated combustion systems (ICSs): a furnace is installed indoors, and all combustion and ventilation air is admitted through grilles or is ducted from outdoors and does not interact with air in the conditioned space. Examples of ICS installations include interior enclosures with air from an attic or ducted from outdoors, exterior enclosures with air from outdoors through grilles, or enclosures in garages or carports attached to the building (NFPA *Standard* 54). This type of installation presents special considerations in determining efficiency.

Generally, the following three categories of installation guidelines must be followed to ensure the safe operation of a heating system: (1) the equipment manufacturer's installation instructions, (2) local installation code requirements, and (3) national installation code requirements. While local code requirements may or may not be available, the other two are always available. Depending on the type of fuel being used, one of the following U.S. national code requirements will apply:

• NFPA 54-99 (also AGA Z223.1-99)	*National Fuel Gas Code*
• NFPA 70-99	*National Electrical Code*
• NFPA 31-97	Standard for the Installation of Oil-Burning Equipment

Comparable Canadian standards are

• CAN/CGA-B149.1-M95	*Natural Gas Installation Code*
• CAN/CGA-B149.2-M95	*Propane Installation Code*
• CSA C22.1-98	*Canadian Electrical Code*
• CAN/CSA B139-M91	*Installation Code for Oil Burning Equipment*

An additional source is the *International Fuel Gas Code* (IFGC) (ICC 1997). These regulations provide complete information about construction materials, gas line sizes, flue pipe sizes, wiring sizes, and so forth.

Proper design of the air distribution system is necessary for both comfort and safety. Chapter 32 of the 1997 *ASHRAE Handbook—Fundamentals*, Chapter 1 of the 1999 *ASHRAE Handbook—Applications*, and Chapter 9 of this volume provide information on the design of ductwork for forced-air heating systems. Forced-air furnaces provide design airflow at a static pressure as low as 30 Pa for a residential unit to above 250 Pa for a commercial unit. The air distribution system must handle the required volumetric flow rate within the pressure limits of the equipment. If the system is a combined heating-cooling installation, the air distribution system must meet the cooling requirement because more air is required for cooling than for heating. It is also important to include the pressure drop of the cooling coil. The Air-Conditioning and Refrigeration Institute (ARI) maximum allowable pressure drop for residential cooling coils is 75 Pa.

AGENCY LISTINGS

The construction and performance of furnaces is regulated by several agencies.

The Gas Appliance Manufacturers Association (GAMA), in cooperation with its industry members, sponsors a certification program relating to gas- and oil-fired residential furnaces and boilers. This program uses an independent laboratory to verify the furnace and boiler manufacturers' certified AFUEs and heating capacities, as determined by testing in accordance with the U.S. Department of Energy's *Uniform Test Method for Measuring the Energy Consumption of Furnaces and Boilers* (CFR Title II, 10 Part 30, Subpart B, Appendix N). Gas and oil furnaces with input ratings less than 66 kW and gas and oil boilers with input ratings less than 88 kW are currently included in the program.

Also included in the program is the semiannual publication of the GAMA consumers directory, which identifies certified products and lists the input rating, certified heating capacity, and AFUE for each furnace. Participating manufacturers are entitled to use the GAMA Certification Symbol (seal). These directories are published semiannually and distributed to the reference departments of public libraries in the United States.

ANSI *Standard* Z21.47/CSA 2.3, Gas-Fired Central Furnaces (CSA America is secretariat), gives minimum construction, safety, and performance requirements for gas furnaces. The CSA maintains laboratories to certify furnaces and operates a factory inspection service. Furnaces tested and found to be in compliance are listed in the CSA Directory and carry the Seals of Certification. Underwriters Laboratories (UL) and other approved laboratories can also test and certify equipment in accordance with ANSI *Standard* Z21.47/CSA 2.3.

Gas furnaces may be certified for standard, alcove, closet, or outdoor installation. Standard installation requires clearance between the furnace and combustible material of at least 150 mm. Furnaces certified for alcove or closet installation can be installed with reduced clearance, as listed. Furnaces certified for either sidewall venting or outdoor installation must operate properly in a 50 km/h wind. Construction materials must be able to withstand natural elements without degradation of performance and structure. Horizontal furnaces are normally certified for installation on combustible floors and for attic installation and are so marked, in which case they may be installed with point or line contact between the jacket and combustible constructions. Upflow and downflow furnaces are normally certified for alcove or closet installation. Gas furnaces may be listed to burn natural gas, mixed gas, manufactured gas, propane, or propane-air mixtures. A furnace must be equipped and certified for the specific gas to be used because different burners and controls, as well as orifice changes, may be required.

Sometimes oil burners and control packages are sold separately; however, they are normally sold as part of the furnace package. Pressure-type or rotary burners should bear the Underwriters Laboratory label showing compliance with UL *Standard* 296. In addition, the complete furnace should bear markings indicating compliance with UL *Standard* 727. Vaporizing burner furnaces should also be listed under UL *Standard* 727.

Underwriters Laboratories *Standard* 1995 gives requirements for the listing and labeling of electric furnaces and heat pumps.

The following list summarizes important standards issued by the International Approval Service, Underwriters Laboratories, the Canadian Gas Association, and the Canadian Standards Association that apply to space-heating equipment:

ANSI/ASHRAE 103-1993	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers
ANSI Z21.66-96/CGA 6.14-M96	Automatic Vent Damper Devices for Use with Gas-Fired Appliances
ANSI Z83.4-91 (R 1998)	Direct Gas-Fired Makeup Air Heaters
ANSI Z83.6-90 (R 1998)	Gas-Fired Infrared Heaters
ANSI Z83.8-96/CGA 2.6-M96	Gas-Fired Duct Furnaces and Unit Heaters
ANSI Z21.47-98/CSA 2.3-M98	Gas-Fired Central Furnaces
ANSI/UL 296-94	Oil Burners
ANSI/UL 307A-95	Liquid Fuel-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles
UL 307B-95	Gas-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles
UL 727-94	Oil-Fired Central Furnaces
UL 1995-95 • CAN/CSA C22.2 No. 236	Heating and Cooling Equipment
CGA 3.2-1976	Industrial and Commercial Gas-Fired Package Furnaces
CAN1-3.7-77 (R 1986)	Direct Gas-Fired Non-Recirculating Makeup Air Heaters
CAN/CGA-2.5-M86 (R 1996)	Gas-Fired Gravity and Fan Type Vented Wall Furnaces
CAN/CGA-2.16-M81 (R 1996)	Gas-Fired Infra-Red Radiant Heaters
CAN/CGA-2.19-M81 (R 1996)	Gas-Fired Gravity and Fan Type Direct Vent Wall Furnaces
CSA-B140.4-1974 (R 1998)	Oil-Fired Warm Air Furnaces

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Chi, J. 1977. DEPAF—A computer model for design and performance analysis of furnaces. AICHE-ASME Heat Transfer Conference, Salt Lake City, UT (August).

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Schade, G.R. 1978. Saving energy by night setback of a residential heat pump system. *ASHRAE Transactions* 84(1):786-98.

Stickford, G.H., B. Hindin, S.G. Talbert, A.K. Agrawal, and M.J. Murphy. 1985. Technology development for corrosion-resistant condensing heat exchangers. GRI-85-0282. Battelle Columbus Laboratories to Gas Research Institute. Available from National Technical Information Service, Springfield, VA.

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McQuiston, F.C. and Spitler, J.D. 1992. *Cooling and heating load calculation manual*. ASHRAE.

Appendix I. List of Biomass Furnace Manufacturers

Advanced Alternative Energy Corp.

1207 N. 1800 Road
Lawrence, KS 66049
1-785-842-1943
http://aaecorp.com
(U.S., Canada, Europe, Asia)

A E & E – Von Roll, Inc.
302 Research Drive, Suite 130
Norcross, GA 30092
1-770-613-9788
www.aee-vonroll.com
(U.S., Canada, Mexico, & S. America)

Alpha American Co.
10 Industrial Blvd.
Palisade, MN 56469
1-800-358-0060
www.yukon-eagle.com
(U.S. and Canada)

Alternative Green Energy Systems, Inc.
20,201 Clark Graham
Quebec, Canada H9X 3T5
1-514-695-0686
(Worldwide)

American Energy Systems
150 Michigan St.
Hutchison, MN 55350
1-800-495-3196
www.magnumheat.com
(U.S. and Canada)

Big M Manufacturing
928 E. 1090 N. Road
Taylorville, IL 62568
1-217-824-9372
(U.S.)

Biomass Combustion Systems, Inc.
67 Millbrook St., Suite 505
Worcester, MA 01606
1-508-798-5970
http://www.biomasscombustion.com
(U.S. and Canada)

Bixby Energy
9300 75th Avenue North
Minneapolis, MN 55428
1-877-500-2800
www.bixbyenergy.com
(U.S. and Canada)

Braymo Energy Corporation
Box 123
Torrington, Alberta, Canada T0M 2B0
1-877-327-2966
(U.S. and Canada)

Burns Best
P. O. Box 680
Spooner, WI 54801
1-877-983-4328
www.burnsbest.com
(U.S. and Canada)

Central Boiler
20502 160th Street
Greenbush, MN 56726
1-800-248-4681
www.centralboiler.com
(U.S. and Canada)

Chiptec Wood Energy System
54 Echo Place, Unit 1
Williston, VT 05495
1-802-658-0956
http://www.chiptec.com/
(U.S., Canada, Europe, S. America)

Dectra Corporation
3425 33rd Avenue NE
St. Anthony, MN 55418
1-612-781-3585
www.garn.com
(U.S., Canada, and Mexico)

Detroit Stoker Company
1510 E. First Street, P. O. Box 732
Monroe, MI 48161
1-800-stoker-4
www.detroitstoker.com
(Worldwide)

Energy King
P. O. Box 27
Chippewa Falls, WI 54729
1-877-720-1794
www.EnergyKing.com

Energy Products of Idaho
4006 Industrial Ave.
Coeur d'Alene, ID 83815
1-208-765-1611
www.energyproducts.com
(Worldwide)

Energy Unlimited, Inc.
P. O. Box 7
Dodgeville, WI 53533
1-608-935-9119
www.energyunlimitedinc.com
(U.S. and Canada)

Golden Grain Corn Stoves
P. O. Box 5000
Sterling, CO 80751
1-800-634-6097
www.goldengrainstove.com/prod_info
(U. S.)

Grove Wood Heat, Inc.
P. O. Box 25
York, P.E. I., Canada C0A 1P0
1-902-672-2090
grovewoodheat@pei.sympatico.ca
(Canada)

Hawken Energy, Inc.
980 Industrial Park Drive, P. O. Box 351
Shelby, MI 49455
1-800-LOG-BURN
www.hawkenenergy.com
(U.S.)

Heatmor, Inc.
Box 787
Warroad, MN 56763
1-800-834-7552
http://www.heatmor.com
(U.S. and Canada)

Heat Source1
2201 Ridgeview Drive
Beatrice, NE 68310
1-888-628-3533
www.heatsource1.com
(U.S. and Canada)

Ja-Ran Enterprises, Inc.
3541 Babcock Rd.,
Lexington, MI 48450
1-810-359-7985
ranoy@ja-ran.com
(U.S. and Canada)

LDJ Manufacturing
1833 Highway 163
Pella, IA 50219
1-866-535-7667
www.cornheat.com
(U.S.)

LMF Manufacturing
601 Woods Ave,
Lock Haven, PA 17745
1-570-748-7080
www.americasheat.com
(U.S.)

L. R. Equipment Corp.
4064 Lyle Road
Beaverton, MI 48612
989-435-9052
www.lrequipment.com
(U.S. and Canada)

McBurney Corporation
P. O. Box 1827
Norcross, GA 30091
1-770-925-7100
www.mcburney.com
(Worldwide)

Messersmith Manufacturing, Inc.
2612 F Road
Bark River, MI 49807
1-906-466-9010
(U. S.)
www.burnchips.com

Meyer Manufacturing Corporation
P. O. Box 405
Dorchester, WI 54425
1-800-325-9103
www.meyermfg.com
(U.S.)

Mitch Hart, Mfg., Inc.
46304 Jeffrey Street
Hartford, SD 57033
1-605-528-4700
www.KernelBurner.com
(U. S.)

Nesco, Inc.
1011 Volunteer Drive,
Cookeville, TN 38506
1-931-372-0130
www.amaizablaze.com
(U.S. and Canada)

Northwest Manufacturing
600 Polk Avenue SW
Red Lake Falls, MN 56750
800-932-3629
www.woodmaster.com
(U.S. and Canada)

Pinnacle Stove Sales
1089 Caribou Highway 97 N
Quesnel, British Columbia
Canada V2J 243
866-967-9777
www.pinnaclestove.com
(U.S. and Canada)

Pro-Fab Industries, Inc.
Box 112
Arborg, Manitoba, Canada R0C0A0
1-888-933-4440
www.cozeburn.com
(U. S. and Canada)

Ryte Heating Systems
Box 30, R. R. 2
Morris, Manitoba, Canada
1-204-0746-8351
(U.S. and Canada)

SolaGen Inc.
33993 Lawrence Road
Deer Island, OR 97054
1-503-366-4210
solageninc.com
(U.S. and Canada)

Vidir Biomass, Inc.
Box 428
???????????????
877-746-8833
www.vidirbiomass.com

Year-A-Round Corporation
P. O. Box 2075
Mankato, MN 56001
1-800-418-9390
www.year-a-round.com
(U.S. and Canada)

Zilkha Biomass Energy LLC
1001 McKinney, Ste 1900,
Houston, TX 77002
713-979-9961
www.zilkabiomass.com

Appendix J. 2011 Block Diagram

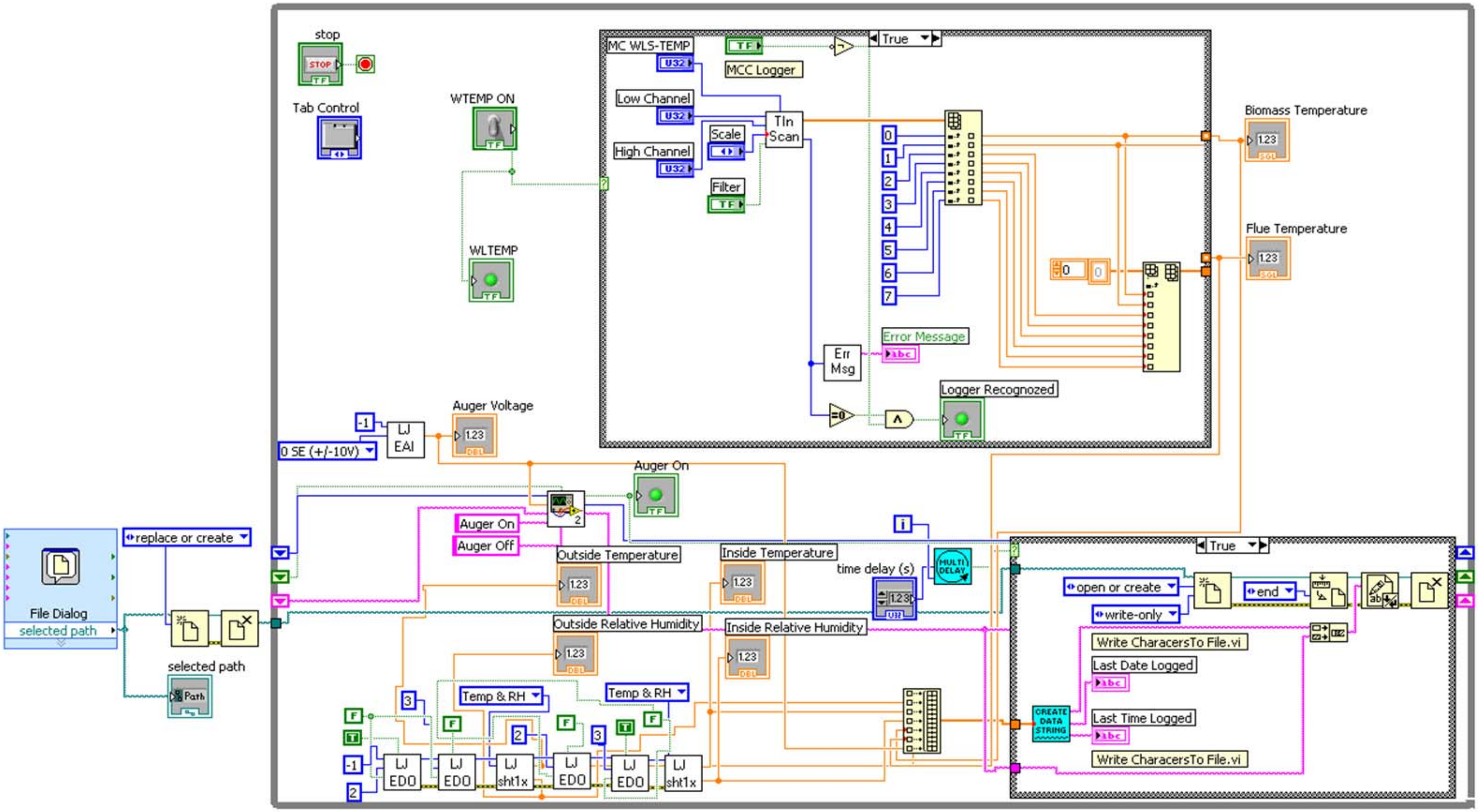

Figure J.1. 2011 LabView Program Block Diagram.

Appendix K. Biomass Blends Thermal Images

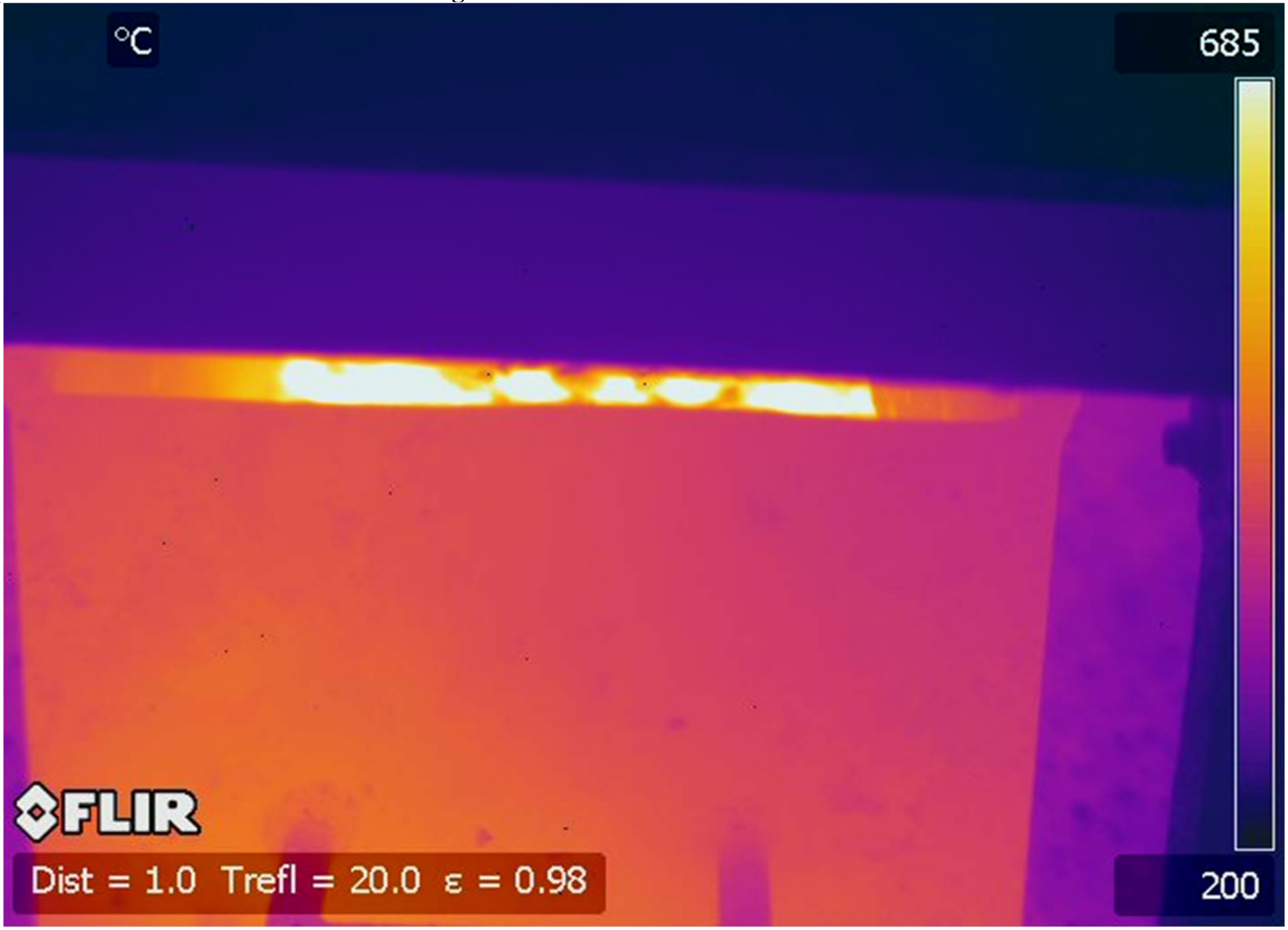

Figure K.1. Corn/Wood Biomass Blend.

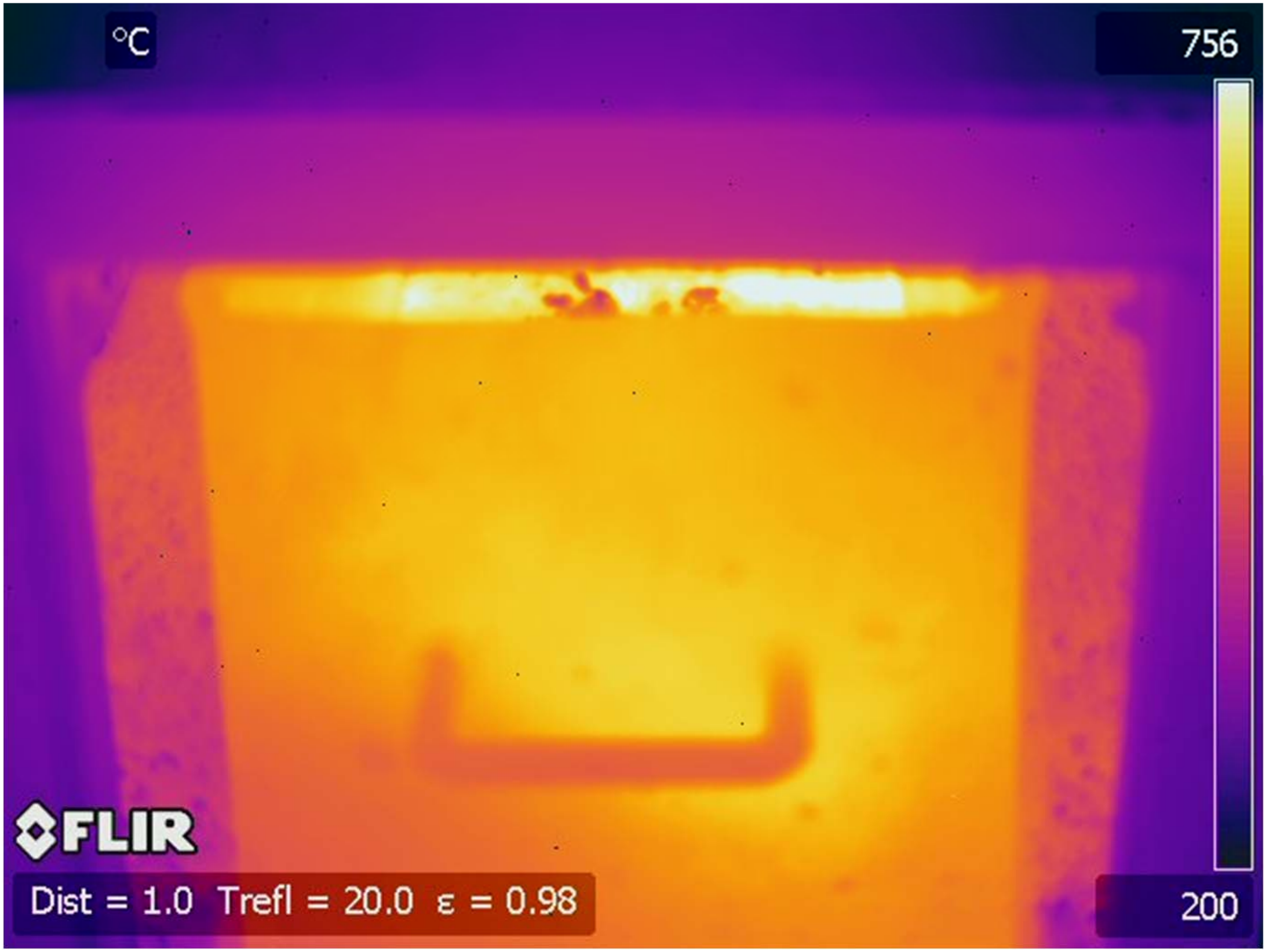

Figure K.2. Corn/DDGPs Blend.

Appendix L. CD of Raw Data

Raw Data is available at Biological Systems Engineering.