

Biofuel Gasification for Residential Heating

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Introduction

The purpose of this project was to develop two natural draft components to be attached to a biomass residential gasification heating unit. The biomass heater is purposed to supply a heating fluid to a storage tank for hydronic radiant heating or other similar intermediate fluid heating systems. Ideally, the system will accommodate a large range of biomass fuels and efficiently provide heat with minimal polluting byproducts.

Our original project was to design a complete gasifier boiler. However, through our collaboration with project sponsor Tod duBois, the scope of our senior project has narrowed on a specific subsystem of the gasifier. With the help of our advisor, Dr. Andrew Davol, we chose to design, build, and test a natural draft nozzle. This particular component is explained below in our report, and specific design schematics can be found in the detailed analysis below.

Background

Gasification is the process of converting organic carbonaceous materials into carbon dioxide, hydrogen, and carbon monoxide. The primary combustion of biomass releases approximately 40 percent of the chemical energy as heat, while the remaining energy is released as carbonaceous gas, often simply referred to as smoke. The secondary combustion of these carbonaceous gases (syngas or producer gas) can result in thermal efficiencies of up to 93 percent. Gasification differs from simple wood burning in that the producer gas is used to do work rather than exhausted to atmosphere. The producer gas can either be used immediately in the case of heating, or filtered and used as a clean burning fuel in an internal combustion engine, generator, or other gas powered machinery.

The producer gas that follows the gasification process differs from the gases exhausted from simple wood burning. The gasification process relies heavily on the air to fuel ratio (AFR). For combustion reactions, the ideal AFR is the stoichiometric rate measured as a mass ratio. Combustion under ideal stoichiometric rates will result in the complete combustion of all fuels and no residual oxygen. AFR numbers below stoichiometric are considered rich and will have unburnt hydrocarbons. AFR numbers above stoichiometric are considered lean and will have residual oxygen but ideally, no unburnt hydrocarbons. Gasification aims to produce rich gas, that is to say an AFR below stoichiometric. This rich syngas contains combustible hydrocarbons without the presence of oxygen. When air is reintroduced to this gas, a second combustion reaction may occur.

The burning of any biomass fuel follows five thermal processes: drying, pyrolysis, oxidation, cracking and reduction. In the drying stage, water is driven out from the biomass fuel at a temperature between 212-300°F. This process is important because any water remaining in the fuel during later processes will extract energy for vaporization causing a lower efficiency. Pyrolysis is the process of converting dry biomass to charcoal and tar gasses in the absence of oxygen at temperatures between 400-900°F. During pyrolysis, the carbon, hydrogen, and oxygen atoms of biomass are fragmented into volatiles (tar gasses) and fixed carbon chains (charcoal). These tar gases are comprised mostly of carbon monoxide and hydrogen gas, both of which are fuels with very good combustion characteristics. The charcoal produced is later used in the reduction reaction. Combustion and cracking follow pyrolysis, which is a largely exothermic process occurring at 1450-2200°F. During combustion and cracking, charcoal, air, and tar gases are converted to water, carbon dioxide, cracked tar, and reactive charcoal. Reduction is the final stage of the gasification process. In this stage, carbon, water, and carbon dioxide are converted to hydrogen gas and carbon monoxide in an endothermic

reaction at approximately 1200-1650°F. The carbon monoxide and hydrogen gas mixture is referred to as producer gas and can be later consumed in a second combustion/reduction reaction if oxygen is reintroduced.

Gasification is a mature science that has been used in practice for over 200 years. Today, gasification is used in two ways - either as an efficient heating process or as a means to convert biofuel directly into syngas to produce electricity. The gasification of biofuels for electric energy production requires clean burning, molecularly consistent syngas. The syngas used in internal combustion engines or generators has to have very little tar, ash, and byproducts to allow for proper, prolonged engine use. The gasification process and syngas produced for heating units can be more crude because all gases are ultimately exhausted to atmosphere, rather than fed into an engine. However, if the combustion of the producer gas is not optimized, unburnt fuels can pass through the system lowering the efficiency and polluting the atmosphere. Gasification for electrical use operates at lower temperatures and requires careful monitoring, while gasification for thermal use attempts to operate at the greatest possible temperature to ensure complete combustion of all fuels. Gasification for electrical use is similar to thermal uses, except the gases are usually filtered and cooled before sent to a generator. Gasification for thermal use, usually combusts the syngas as close to the primary solid fuel reaction as possible in order to capitalize on the heat of the primary reaction.

Regardless of the end purpose of the gasification process, standards set by government agencies, such as the United States Environmental Protection Agency (EPA), need to be taken into consideration when designing a system that combusts fuel and exhausts gases to the atmosphere. In the United States, exhaust emissions are strictly regulated by the federal government for the public welfare. According to the National Energy Technology Laboratory (NETL), the United States Environmental Protection Agency was required by Congress in 1963 to establish standards “for any pollutants which affect public health and welfare.” The emissions of any gasification process are required to hold to the standards established by the EPA. Updated and adjusted to meet modern technology, the EPA revises the standards every five years. In the last revision released on February 3, 2015, the government agency declared the limit for particulate matter (PM) contained in emissions of wood burning appliances to be 0.32 pounds per million BTU heat output. The EPA also issued a statement that by 2020, the government will be enforcing that all emissions must not exceed a maximum of 0.10 pounds of PM per million BTU heat output. These regulations will set the standards for gasification emissions.

The efficiency and performance of a gasification boiler system is highly dependent on its ability to sustain high temperatures. An insulating layer may be used to retain a gasifier’s high temperatures in order for proper combustion and safety regulations to be met. There are a multitude of insulating materials that can be used for this requirement, but some options are expensive. For a cost effective system, the use of refractories is the best option to insulate a gasifier. Refractories are commonly used to insulate and protect furnaces from chemical, thermal, and mechanical damage. There are three classes of refractories: acid, neutral, and basic. For vessels experiencing fumes at elevated temperatures such as gasifiers, a basic refractory is ideal because of its stability in alkaline (non-acidic) environments. Basic refractories are resistant to fatigue caused by alkaline slag and conserve heat at high temperatures. Some refractories are preshaped, such as bricks and tiles, while others have the ability to be casted into specific shapes.

To prevent heat losses, we used insulating refractories around high temperature components. Because our sponsor, Tod duBois, already possessed castable refractories, we will use these to help in the construction of

our nozzle. Basic refractories are selected based on their heat capacities and chemical reactions at high temperatures. When selecting our refractory material, the temperature limits, thermal conductivity, and thermal expansion of the material were considered. Research also showed the relationship between high temperatures and refractory durability. For example, lightweight chamotte and kaolin bricks are often used for application temperatures up to 2500°F. However, these bricks also have disadvantages, because their porosity allows the material to expand on their heated surface, weakening their mechanical strength and structure. Therefore, heat capacities are not the only consideration. In conclusion, heat capacities and overall refractory selection also requires consideration of refractories' chemical characteristics as well as cost.

When designing a gasifier system, it is crucial to have a biomass fuel that is readily available, clean burning, and sustainable. It is difficult to design a universal gasifier that runs off any form of biomass because biomass fuels have different chemical structures. If a biomass fuel, such as wheat straw or corn stalks, produces a higher level of ash content, the gasifier will experience slagging. Slagging is the process in which a glass-like by-product remains after the gasification process of a fuel. Slag is usually a mixture of oxides and silicon dioxide which can cause permanent damage to the system. Therefore, it is ideal to use a fuel source with a lower ash content, such as corn cobs or alfalfa seed straw. Fuels with a lower ash content prevent slagging and combust more efficiently. To minimize slagging, a fuel source should have an ash content below 5% for optimal use.

When selecting a fuel source, one also needs to take into consideration the moisture content of the fuel. It is desirable to have fuels with minimal amount of water content in order to decrease the length of the drying process. Dried fuels also extract less heat from the combustion reaction. The origin of the fuel source also has an effect on the gasification process. For example, if wood was collected near the ocean, it would contain chlorine, causing an increase in ash content. The potassium and phosphorus present in bark and sawdust will also increase the ash production reducing the efficiency. The only biomass fuel that does not require special attention is coal, but its price, availability and its non-renewable source makes it a non-ideal fuel source.

Of all the biofuels available, wood is most commonly used because it is inexpensive, largely available, causes minimal slagging, burns clean, and has a well-studied combustion reaction. Figure 1 shows the typical heating value of common wood species and Figure 2 shows the molecular analysis of common dry wood species. Both of these figures were provided by Richard C. Hill of the University of Maine's Patent of a stick-wood fired furnace. See Appendix A for more on the chemical reactions of biofuel combustions.

SPECIES	WT/CORD ²	BTU/LB	BTU/TON	BTU/CORD ²
White Ash	4034	8920	17,840,000	35,983,280
Fir Balsam Bark		9100	18,200,000	36,400,000
Yellow Birch	4121	8650	17,300,000	35,655,300
Yellow Birch Bark		9870	19,740,000	
White Cedar	2236	8400	16,800,000	18,782,400
Elm Bark		7600	15,200,000	
Eastern Hemlock	2722	8620	17,240,000	23,463,640
Eastern Hemlock Bark		8890	17,780,000	
Red Maple	3460	8580	17,160,000	29,686,800
Red Maple Bark		8190	16,380,000	
Oak (white)	4433	8810	17,620,000	39,063,540
Pine (yellow)	3917	9610	19,220,000	37,642,370
Pine (white)	2352	9000	18,000,000	21,168,000
Pine (white) Bark		8930	17,860,000	
Poplar	2187	8920	17,840,000	19,516,960
Poplar Bark		8810	17,620,000	

Figure 1. Typical Heating Value of Oven Dry Wood

	Percent by Weight					
	Carbon, C	Hydrogen, H ₂	Sulfur, S	Oxygen, O ₂	N ₂	Ash
<u>HARDWOODS</u>						
Ash, white	49.73	6.93	- -	43.03	- -	0.30
Beech	51.64	6.26	- -	41.45	- -	0.65
Birch, white	49.77	6.49	- -	43.45	- -	0.29
Elm	50.35	6.57	- -	42.34	- -	0.74
Hickory	49.67	6.49	- -	43.11	- -	0.73
Maple	50.64	6.02	- -	41.74	0.25	1.35
Oak, black	48.76	6.09	- -	44.98	- -	0.15
red	49.49	6.62	- -	43.74	- -	0.15
white	50.44	6.59	- -	42.73	- -	0.24
Poplar	51.64	6.26	- -	41.45	- -	0.65
<u>SOFTWOODS</u>						
Cedar, white	48.80	6.37	- -	44.46	- -	0.37
Cypress	54.98	6.54	- -	38.08	- -	0.40
Fir, Douglas	52.30	6.30	- -	40.50	- -	0.80
Hemlock, western	50.40	5.80	0.1	41.40	0.1	2.20
Pine pitch	59.00	7.19	- -	32.68	- -	1.13
white	52.55	6.08	- -	41.25	- -	0.12
yellow	52.60	7.02	- -	40.07	- -	0.31
Redwood	53.50	5.90	- -	40.03	0.1	0.20

Figure 2. Typical Molecular Analysis of Dry Wood

A few expensive wood boiler gasifiers exist on the market today. Most wood boiler gasifiers on the market use an insulated firebrick firebox with a connecting refractory secondary burn chamber directly above or below the firebox, such as the ProFab Empyre Elite XT Heater schematic in Figure 3.

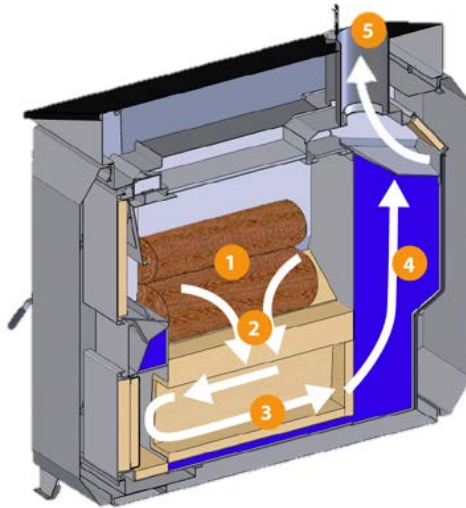


Figure 3. ProFab Empyre Elite XT Heater Schematic

Similar to many wood gasification boilers, the ProFab Empyre Elite XT Heater is started by placing small kindling in the firebox (1) through a horizontal door. A blower is turned on, creating a draft pulling atmospheric air through the primary chamber (1) to secondary combustion (3) and then to the chimney exhaust (5). Paper is placed under the kindling and then ignited with a match. Once the fire is established, the fire is stirred to create a bed of coals over the nozzle (2). Larger logs are then loaded into the firebox, allowing for burn times up to six hours. The gas and smoke created from the combustion of the wood fuel travels through the fire nozzle (2) and into the secondary burn chamber (3). The smoke is pulled by the blower through the firebox into the refractory lined nozzle and secondary burn chamber. Air is reintroduced in specific ratios in the secondary burn chamber and a second combustion/reduction reaction of the gas occurs in the secondary burn chamber (3). Gases are then expelled at temperatures up to 2,000°F to the heat exchanger (4). The heat of the combusted gas is exchanged with water and then exhausted through the chimney. ProFab suggests that the exhausted gases can be as low as 350°F during optimal operation. The ProFab Empyre Elite XT Heater is comparable to our biofuel boiler prototype which will be described in more detail in the subsequent sections. The Empyre Heater retails for \$7,995 and requires a 120 volt, 15-amp electrical input to power the blower and monitoring systems.

Wood boilers are typically integrated into residential heating in a configuration similar to that in Figure 4. The wood boiler supplies hot water to the buffer tank, where diffusers and buoyancy effects keep the cold water from mixing with the hot water. The hot water is then fed into a mixing valve, where colder return water mixes with the hot water to achieve the desired temperature for the piping and the residents ideal comfort temperature. Once passed through the piping of the house, the cold water is returned to the buffer tank then fed back to the wood boiler and the cycle continues.

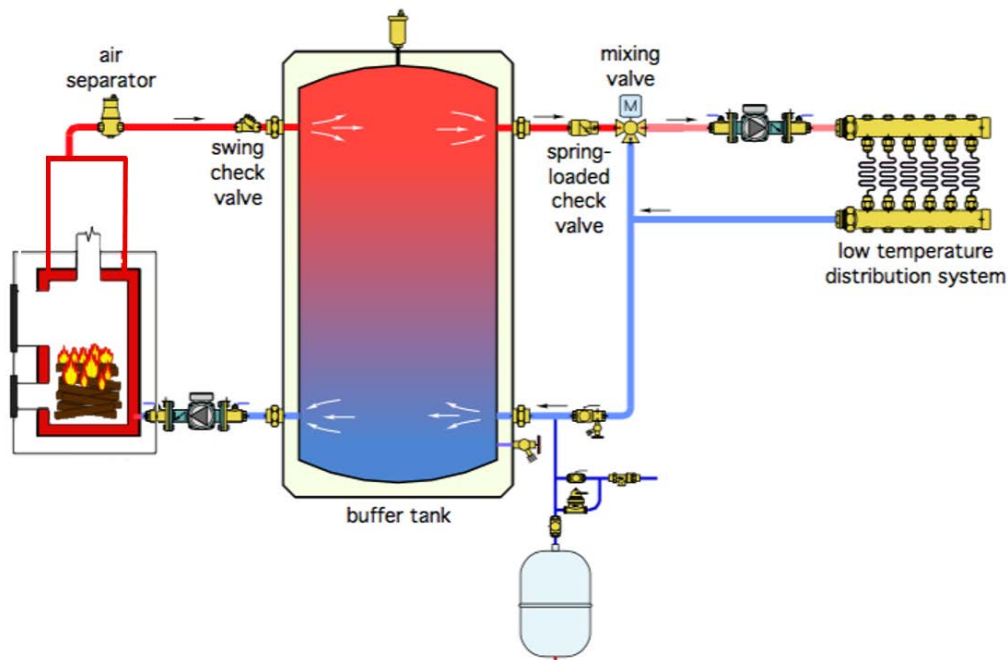


Figure 4. Standard Wood Boiler Hydronic Heating System

Objective

Our sponsor, Tod duBois, asked us to take his current wood gasifier design and create a nozzle and heat exchanger that will be placed within his system to create a natural draft. Mr. duBois originally emphasized that our nozzle eliminates the need for electrical resources to deduct an electrical cost of the system. This nozzle has been designed to deliver an overall heating load of at least 20,000 BTU/hr to the water storage tank. The design has four main functions: provide a natural draft moving gas from primary to exhaust; provide a vacuum pressure, to create suction through the heat exchanger pipes; preheat secondary combustion air via helical pipes, and provide sustainable temperature for autoignition combustion of syngas.

The initial scope of the project involved designing a new gasifier system from scratch, but after further research and discussion with Mr. duBois and Professor Davol, we reached an agreement that our project needed to focus on designing the nozzle and heat exchanger portion of Mr. duBois' prototype. Consequently, the nozzle portion of the gasifier is constrained by the prototype that Mr. duBois has already built. The nozzle outer diameter had to fit within an 8" cylinder.

With our nozzle design incorporated into 'Tod duBois' pre-existing prototype, displayed in Appendix D, we were able to satisfy the engineering specifications that were assigned for the complete gasifier system in Table 1. We attempted to keep the total nozzle cost as low as possible so the entire gasifier would be under \$2000.

Table 1. Original Engineering Specifications

Spec. #	Parameter Description	Requirement or Target (units)	Tolerance	Risk	Compliance
1	Price	\$2,000	Max	High	A, S
2	Heat Output	20,000 BTU/hr	Min	Medium	A, T
3	Size	4'x4'	Max	Low	A, I
4	Weight	4,000 lbs	Max	Low	A, S
5	Safety	140 °F (External)	Max	Medium	A, T
6	Material	Withstand 2500°F	Min	Low	A,I
7	Run Time	3 Hours	Max	Medium	A,T

A = Analysis, T = Test, S = Similarity to Existing Designs, I= Inspection

Management Plan

For the analysis of the nozzle design, Bryan took the lead in finding the appropriate temperatures, and pressures needed for secondary combustion. Since the analysis of the nozzle was one of the most comprehensive parts of our project, Bryan mainly be focused on the fluid dynamics analysis, while assisting other teammates when needed. Bryan performed extensive research to determine the effects of pressure, temperature, and flowrate on the shape of the nozzle. Because Bryan focused mainly on the nozzle design, Courtney proceeded to work on the analysis pertaining to the heat exchanger of the system.

While all members of the team worked on the analysis portion of the project, each member had their own assigned tasks. Jeron took the lead for manufacturing the nozzle and heat exchanger. He reached out to the Cal Poly machine shops to gain a better insight for possible manufacturing routes. With the help of the machine shops on campus, Sangha Energy, and team members, Jeron ensured the manufacturing was completed.

Courtney researched testing methods to verify the validity of our design. The testing portion of our project began when the nozzle manufacturing was complete. With Courtney taking the lead of testing she reached out to various professors on campus for advice, such as Dr. Westphal and Dr. Thorncroft. The team decided that the main focus of testing was to measure the flow rate and the pressure drop discussed later in the testing section.

Basic Quantitative Analysis Principles

Our system utilizes the laws of fluid dynamics to create a draft and minimize the use of fans or blowers. Gases flow from high to low pressure, therefore we needed a vacuum to force the woodgas through the

secondary combustion chamber. Our project utilizes the Venturi effect through the design of our converging-diverging nozzle. The Venturi effect states that when a gas flows through a pipe with decreasing diameter, the velocity of the gas will increase and the static pressure will decrease. In our design, the gas flows through the inlet of our nozzle, where the cross sectional area decreases until it reaches the throat diameter. Therefore, the gas will see a drop in pressure and an increase in velocity at the throat. The gases also increase in temperature during combustion, occurring slightly after the throat of the nozzle. This increase in temperature is caused by the chemical reactions found in Appendix A.

The heat transfer analysis of the preheated air was modeled using basic conduction and convection principles. The thermal resistance network of the heat transfer was used to determine the proper positioning of the heat exchanger piping that coils around the nozzle. In determining the proper placement, we assured the thermal equivalence resistance system is one directional.

Analysis

The overall shape of our nozzle was largely analytically based. However, some limiting pre-existing prototype dimensions also defined aspects of the nozzle design. For example, the inner diameter of the keg shell heat exchanger provided the maximum outer diameter of our nozzle shape. The accuracy of the manufactured parts limited the distance the preheat coils could be to the nozzle inner wall.

To start the analysis of the nozzle design, we first needed the properties of the wood gas entering the nozzle. These calculations started with the sponsor's required heating loads - a maximum of 100,000 BTU/hr. After a first run of calculations, 100,000 BTU/hr natural draft system seemed improbable, so a second 20,000 BTU/hr iteration was completed. We assumed a gasifier using dry white pine wood, with a heating value of 9,000 BTU/lb or 19,8405 BTU/kg, would produce 40,000 BTU/hr for a fuel consumption rate of 2 kg/hr. Assuming an overall efficiency of 50%, the useful heat gain would be 20,000 BTU/hr.

As mentioned earlier, gasifiers operate with a combustion air rate less than stoichiometric. This fraction of the stoichiometric rate is called the equivalence ratio (ER). An equivalence ratio between 0.20 - 0.33 is recommended for the gasification process. An ER of 0.20 will produce rich, energetic, but also tarry gas. An ER of 0.33 produces hot, but low energy density gas. Gasifiers that produce gases for internal combustion engines use cooler gases with higher energy densities and thus, lower ERs. Gasifier systems using moist fuel or with poor thermal efficiency require a higher ER. For our system, with the main objective being high heat output. With a wide range of fuel, we are looking for hot gas that can cause autoignition in the secondary chamber. For these reasons, we have established an ER of 0.30.

The stoichiometric rate for complete combustion of dry wood is 6.5 kg air per 1.0 kg fuel. With an ER of 0.30, the air needed for primary combustion is 3.93 kg air/hr. With a fuel consumption rate of 2 kg/hr the total wood gas entering the inlet of the nozzle is 5.93 kg/hr. The remaining secondary air required for the stoichiometric reaction is 1.0 less than the ER, or 9.1 kg/hr. However, the stoichiometric rate is an ideal scenario. Due to gas phase kinetics, excess air must be provided to ensure proper mixing of the air and fuel, and to ensure complete combustion. Too much air, however, will dilute our exhaust gas and decrease our heat exchanger efficiency. We have established a required 50% excess air in our design which is common for natural gas combustion applications. The resulting secondary air that needs to be introduced at the nozzle choke is 13.65 kg/hr.

Here lies our problem: 13.65 kg/hr of air needs to be introduced under vacuum pressure with only 5.93 kg/hr of wood gas. Furthermore, in an effort to design out the blower we have no means to drive the flow other than the buoyancy driven stack effect.

To pull large flow rates of secondary combustion air with a relatively small wood gas flow rate, we need either a small choke diameter, large inlet air pipes, or some available vacuum pressure available throughout the entire nozzle. We limited the choke diameter to 2 inches to ensure ease of manufacturability and to limit the losses at extreme velocities from friction. The only other source drawing air in for secondary combustion is with a blower.

The large change in density from the combusted gas to the outlet can provide a natural draft, pulling both wood gas and combustion air through the nozzle. However, this stack effect is relatively weak at low chimney heights. As seen in Figure 5, in order to get a substantial draft, either a tall chimney or a large flue gas temperature to atmospheric temperature difference is needed. In this nozzle design, we have a large temperature difference but a limited height. With a 1000°C temperature flue gas and a meter-tall chimney exhausting to atmosphere, a draft of 9.55 Pa is produced.

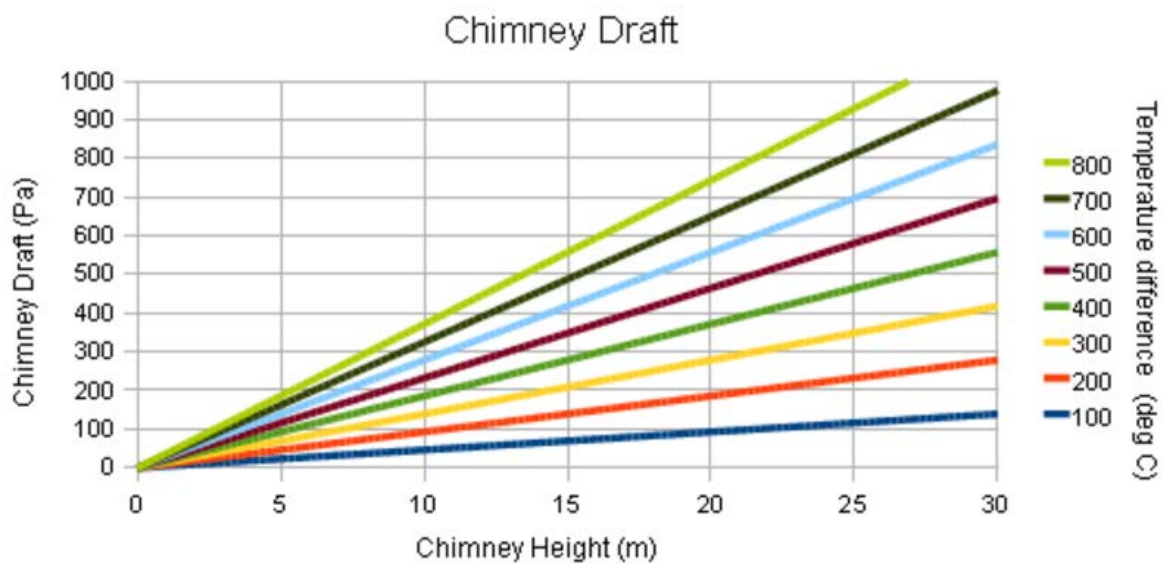


Figure 5. Chimney Draft vs Chimney Height over Varying Temperatures

Along with providing a stack effect draft and drawing in combustion air, our nozzle also serves as the secondary combustion chamber. Because there is no pilot flame in our combustion chamber, high temperatures in the nozzle are needed for autoignition. The autoignition temperature for carbon monoxide and hydrogen gas is about 609°C and 500°C, respectively. Our heavily insulating refractory layer will retain the temperatures from primary reaction needed for secondary combustion. It was also suggested to us that the height of the nozzle would need to be tall enough to allow ample time for mixing and combustion. Upon further research, we have found the ignition time delay is in the range of 50-500 microsecond which provides plenty time for ignition as the gas flows at an average velocity of approximately 6 m/s. The reaction time

decreases at higher pressures and temperatures, which would only benefit the ignition timing as the systems heats to steady state temperature.

Assuming incompressible, inviscid, and steady flow of the woodgas, were able to use the ideal Bernoulli's equation. The assumption of incompressible flow is reasonable as the wood gas flow reaches a Mach number of 0.03 at the choke, which is much less than the $Ma = 0.3$ suggested limit for incompressible flow. The flow of the wood gas can be considered inviscid because the length to diameter ratio is small and the surface roughness of the refractory is rather smooth. The steady flow assumption is valid for times after startup when the refractory has reached a steady temperature. This is usually 15-20 minutes. With an 8-inch inlet diameter, a 2-inch choke diameter, and an inlet velocity of 0.7925 m/s, the available vacuum pressure at the choke is 80.913 Pa. Including the 9.55 Pa from the effects of buoyancy mentioned earlier, the total vacuum pressure available is 90.459 Pa.

The same assumptions allowing us to use Bernoulli's equation for the woodgas hold for the air entering the nozzle, except viscous effects must be accounted for. Our preheated air pipes have small diameters and long lengths. The riveted steel pipes also have a relatively high surface roughness, making the inviscid assumption invalid. To account for the friction losses, we created an excel tool program that calculates head loss at different air inlet configurations and choke diameter. With a friction factor of 0.02 and a pipe length of 11.5 feet, the head loss is 77.365 Pa.

With an 8-inch inlet, a 2 inch choke, and three 0.5 inch air inlet pipes, our incoming air flow rate is 11.42 m³/hr, which is 30% greater than the required 11.42 m³/hr. This 30% extra flow rate was chosen as our design point and provides a cushion for error. Also, any small increases in excess air will have very little effect on the overall performance.

These calculations were based off the assumption that the gasifier produces gas at the necessary flow rate. In most gasification systems, a blower provides a pressure drop across the reaction bell which yields the flow rate of woodgas. Unfortunately, the buoyancy effect of the gasses is not enough to produce the needed 4-6 inches of water pressure typically found in gasifiers to draw the gasses through the combustion hearth and sustain the needed flow rate. It should be noted that this vacuum pressure is needed for the solid fuel gasification hearth and is not necessary in our nozzle design. However, there is not enough stack pressure to ensure these flow rates. Therefore, a light duty fan is needed to draw the air from the combustion chamber to exhaust and ensure proper flow rates. This fan will be placed at the inlet of the air pipes. We have specified a San Ace 40 fan to provide the needed flow rate. This blower provides 29.3 cfm or approximately 49.8 m³/hr and is discussed in more detail in the following sections.

The heat transfer measured across the refractory to the preheated air was performed in various ways. Each of the methods aimed to design a tubing system to heat atmospheric air (20°C) to about 593 °C through conduction and convection. In all models, radiation was negligible and the stainless steel tubing of the heat exchanger was treated as thin-walled piping.

The thermal resistances of the system included the convection of the syngas on the flume walls, the conduction transmitted through the refractory, and the convection of the preheated air through the heat exchanger coils. Theoretically, this method is accurate. However, the fluid air moving through the coiled

pipe follows a spiraling path and causes a centripetal force toward the refractory of the inner wall of the piping. The air in the piping is also experiencing mixing, requiring a convection correction factor, determined in complex heat transfer analysis.

In our first attempt, we modeled the flow across the nozzle as 1-D heat transfer as convection from the syngas, conduction through the refractory, and convection through the preheated air pipes. Using iteration cycles in Excel, we formulated a 10,128 cell matrix iterating the nozzle diameter, height of the nozzle, and distance between the flume and heat exchanger pipes. A screenshot of the excel file is located in Appendix G. These calculations concluded a maximum heat transfer of 33.16 BTU/sec. We also realized that the thermodynamic properties (such as dynamic viscosity, density, and Prandtl number) of wood gas could not be determined through Engineering Equation Solver (EES) or any other database within our research. However, through this simplified model, we concluded the height of the nozzle, the size of the nozzle diameter, and the distance between the flume of the nozzle and the heat exchanger piping against the overall heat transfer.

Our second method assumed a flume wall temperature of 1000°C and modeled only refractory conduction and air convection. Using an EES file, we were able to calculate the minimum length of the pipe and the thickness of the refractory layer separating the pipe heat exchanger from the wood gas flowing through the nozzle. From the iterative techniques shown in Appendix G, the output temperature of the air flowing from the heat exchanger will be about 444°C with a length of 3.5 m of heat exchanger piping and a refractory thickness of 2.54 cm (1 inch). We decided to use three individual ½ inch heat exchanger pipes with a pitch of about 8.79 mm (0.346 in), resulting in about 12 coils to reach our final desired air temperature. This calculation is also based on the assumption of a nozzle length equal to 1.5 meters. The pitch of our coils is not a sensitive parameter, however it ensures no pipe interference.

In reality, the heat transfer analysis of our nozzle should be performed through computational fluid dynamics. The total heat transfer will vary due to mixing non uniform surface temperature of the heat exchanger piping. With a deeper study of computational fluid dynamics and heat transfer, further modeling and analysis would be used to improve the accuracy of the data.

Cost Analysis

One of the largest tasks of this project was to create an entire system under \$2000. Because we are focused on the nozzle and heat exchanger, our overall cost much less than the entire gasifier budget. The Bill of Materials listed below, Table 2, shows the overall cost of the nozzle and heat exchanger design.

Table 2. Bill of Materials

Part	Quantity	Supplier	Price
Kast-O-Lite 26 LI Castable Refractory (55 lb Bag)	1	High Temp Tools	\$65.00
Flex Tubing Corrugated Stainless Steel (164 Feet, 1/2" Diameter)	1	Duda Diesel	\$175.00
8"X48" Concrete Form Tube	1	Home Depot	\$7.40
10"X48" Concrete Form Tube	1	Home Depot	\$9.97
Sanyo Denki DC Blower 40X28mm/12VDC/18.3 W 29.3CFM	1	Mouser Electronics	\$22.20
Polyethylene Foam Cylinder	1	Foam Factory	\$25.99
Rutland - Refractory Cement ½ Gallon	1	Ace Hardware	\$17.27
Unique Goods 1803BKW DC Motor Speed Controller	1	Amazon	\$6.99
6"X24" PVC Pipe	1	Home Depot	\$9.96
Total	9	-	\$339.78

Material Selection

For our system we used a variety of materials, that were each selected for specific characteristics. Since the casted nozzle shape was designed using hand chisels and files, we used a material that was able to withstand these forces. Thus, we decided to use a high density foam. With a high density foam, we were able to get a rigid structure with a narrow throat diameter that was able to stand alone during casting. High density foam also had the added benefit of a higher surface finish. A finer surface finish produced a smoother refractory wall and less frictional drag on the woodgas.

With castable refractories we were able to mold components at a cheaper cost than machining parts. As previously stated, refractories were chosen for their high thermal conductivity.

For the pipes casted in the heat exchanger we used corrugated stainless steel. Stainless steel is its high melting temperature of 2500°F allowing it to withstand the high combustion temperatures. Another added benefit to using stainless steel is that it is non-corrosive. Since the gasifier will be operating outdoors, it is crucial that it will be able to withstand the elements. Corrugated stainless steel pipes are also rigid, while still being flexible. This allows us to form the proper shape.

Research was also done on efficient, inexpensive, and low power consumption blowers. The Sans Ace 40 blower was chosen, as it can provide 4.42" water, 29.3 cfm, and 18.3 watts at maximum output. The performance curves can be seen in Figure 6. The fan was connected to a simple PWM controller which can pulse a DC voltage at high frequency. This allows for operation at some range below maximum - usually between 10-100% duty cycle. A PWM controller will also use less power when set to a duty cycle below 100%.

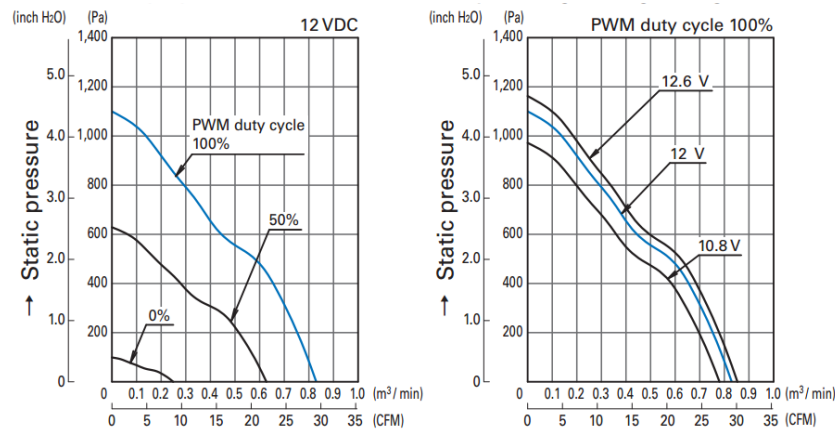


Figure 6: PWM Duty Cycle and Operating Voltage Range of the Sans Ace 40 Blower

Manufacturing

Our manufacturing process produced two key components for our sponsor's complete design: the converging-diverging nozzle and the heat exchanger. As previously described, these components were designed to achieve the pressure, temperature, and flow rate needed for the secondary combustion of the woodgas. To create the nozzles' unique shape, we created a converging-diverging form out of foam. Because the nozzle and heat exchanger are sitting inside keg frames, as seen in Figure 7 below, we used cardboard cylindrical forms with a wax coating as outer shells. We poured refractory around the foam nozzle and heat exchanger and set them aside for hardening. After 24 hours of setting and curing, we removed the templates and sanded the final finish on our nozzle and heat exchanger. Proceeding are the detailed steps of the manufacturing process.

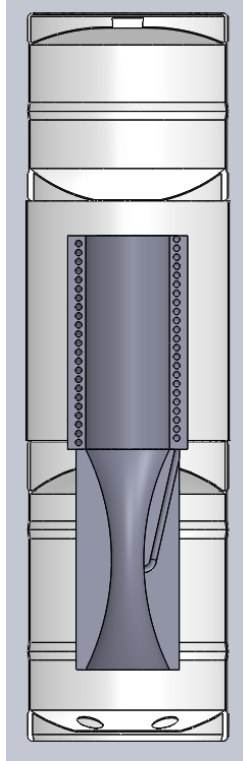


Figure 7. Nozzle and Heat Exchanger Placement

As previously described, we selected a material for the male form of our nozzle. We needed a material that could be easily removed from the final refractory casting. Its composition is adaptable for shaping and easily fragmented for deconstruction. We were able to obtain a square stock piece of high density foam from the Cal Poly Mechanical Engineering Department and proceeded into the shaping process of the nozzle design.

To create the nozzles' shape, we drew its curved profile on a sheet of plywood as a template. The plywood was cut along the drawn edge using a bandsaw and was used to check the shape of the foam as it was sanded.

To rotate our nozzle in the shaping process, we glued wooden dowels on both ends of the foam. The dowels served as fixtures and created an axis of rotation parallel to the ground. Progress of the shaping can be seen below in Figure 8.



Figure 8. Nozzle Shaping Process

After finishing the nozzle shaping, we removed the dowels from the high density foam using a handsaw. Once the dowels were cut off, we sanded both ends of the nozzle to ensure that the nozzle would sit flat during the refractory pouring process. We then added three wooden dowels that were later used as place holders for our preheated air pipes at the throat of the nozzle. Once the dowels were secured in place, we took three 1.5 foot segments of the $\frac{1}{2}$ inch corrugated stainless steel pipe and attached them to the dowels. The corrugated pipe was curved upwards to allow for the pipes exit at the top of the nozzle. The nozzle was set aside until the completion of the heat exchanger.

Next, we began construction of the heat exchanger. We were given a supply of $\frac{1}{2}$ inch corrugated stainless steel pipe that we wrapped into a circular formation to create the heat exchanger. We proceeded to cut three pipes each 6.5 feet in length, and formed them to a 6.5 inches in diameter. The pipes were later placed around a 6-inch inner form. To correctly position the pipes, we took a 6 inch PVC pipe and layered wooden spacers to the outside diameter. While wrapping the piping we needed to ensure that it would stay in place, so throughout the process zip ties were used to help with the positioning. The wrapping process continued until the final shape of the heat exchanger was formed and can be seen below in Figure 9.



Figure 9. Heat Exchanger Form

Upon the completion of the nozzle form, we poured the refractory. We casted two separate cylinders, one with the nozzle in the center, and one with the heat exchanger casted into the refractory. For the nozzle casting, we obtained an 8-inch diameter cardboard form tube, in which the foam nozzle form was placed. The cardboard form was cut to 1" above our desired height to prevent an overflow of refractory. We then placed the nozzle inside the cardboard tube, and centered the nozzle. We then mixed refractory and began pouring refractory, patting down the sides and tops to prevent air bubbles.

After the nozzle pouring was complete, we moved onto preparing the heat exchanger for casting. To do so we obtained a 10-inch diameter cardboard concrete form and a 6-inch PVC pipe. Our heat exchanger piping was centered in the 10-inch cardboard form, with the 6-inch PVC pipe centered inside the piping. The cardboard form needed slits cut for the entering and exit pipes to be outside of the refractory. After the heat exchanger had been properly secured within the 10-inch cardboard form, we were able to pour refractory between the form and the PVC pipe, coating the heat exchanger. Both the nozzle and heat exchanger pouring can be seen in Figure 10 below.



Figure 10. Refractory Pouring

While curing for 48 hours, the refractory was wrapped in a plastic tarp to retain its moisture. Once the refractory finished curing, we removed the forms. The foam was picked out using a small chisel to help prevent any damage. The removal of the PVC piping was more difficult than planned since the PVC stuck to the refractory. With this complication, we had to take a hand saw and cut away at the PVC until it had a straight cut down one side. Once this cut was made, we pulled the PVC from the heat exchanger. Finally, we removed the outer cardboard form.

Once the forms were removed, we used premixed refractory cement to fill in any cracks or low spots of the nozzle and heat exchanger. We also sanded certain parts of the nozzle to create a smoother surface.

To assemble both components together for testing, we used a 1 ½ inch steel pipe as a header for the three inlet air pipes of our heat exchanger. We sealed the ends with premixed refractory to ensure no leaks. The Sans Ace 40 blower was mounted to the end opposing the three inlet air pipes and connected to our PWM

controller. The PWM controller was connected to a power source provided by the Cal Poly Electrical Engineering Department.



Figure 11. Heat Exchanger and Nozzle Finished Design

Design Verification

Our team performed two sets of tests. The first test measured the change in air temperature between the inlet and outlet of our heat exchanger. The second provided measurements of the total head loss experienced by the air flowing through the corrugated piping. The thermal test validated our design. However, the flow test results proved our design calculations were inadequate representations of the actual head loss. Therefore, the results were used to theorize design revisions to reduce the head loss.

Thermal Test

The thermal test was performed using two thermocouples. The first thermocouple was located inside of the heat exchanger's inner wall. The second was placed at the exit of the corrugated piping. Because the air flowing into the corrugated piping was atmospheric, the recorded temperature at the inlet of the heat exchanger was at a constant ambient temperature.



Figure 12. Thermal Test Setup with Propane Burner and Thermocouples

Figure 12 shows our testing apparatus and the setting of our data collection. For safety purposes, the test was performed outdoors. The outdoors setting is an uncontrolled environment; therefore, it was non-ideal for our controlled experiment. However, the ambient temperature was particularly low on this day, forcing our heat exchanger to work in realistic conditions.

The propane burner was placed directly below the heat exchanger's inner surface. The burner was controlled with a valve located next to the spout of the gas tank. We used this control to vary the amount of gas provided to the flame, consequently changing the temperature inside the inner wall of heat exchanger. With the ability to control the temperature of the flue gasses inside the heat exchanger, we were able to analyze the effectiveness of the heat exchanger in different conditions. Unfortunately, the burner never allowed us to reach our expected operating temperature of 1800°F. However, our results at lower temperatures provided sufficient data to analyze the performance of the heat exchanger.

We performed two trials in which we analyzed the heat transfer between the hot gas temperature and the air exiting the heat exchanger coils. The trials measured the change in air temperature for average flue gas temperatures of 408°F and 788°F. The raw data collected in this test can be found in Appendix J. Using the Excel and EES files created for our design, we were able to graph our expected results with our actual results. This graph is shown below in Figure 13.

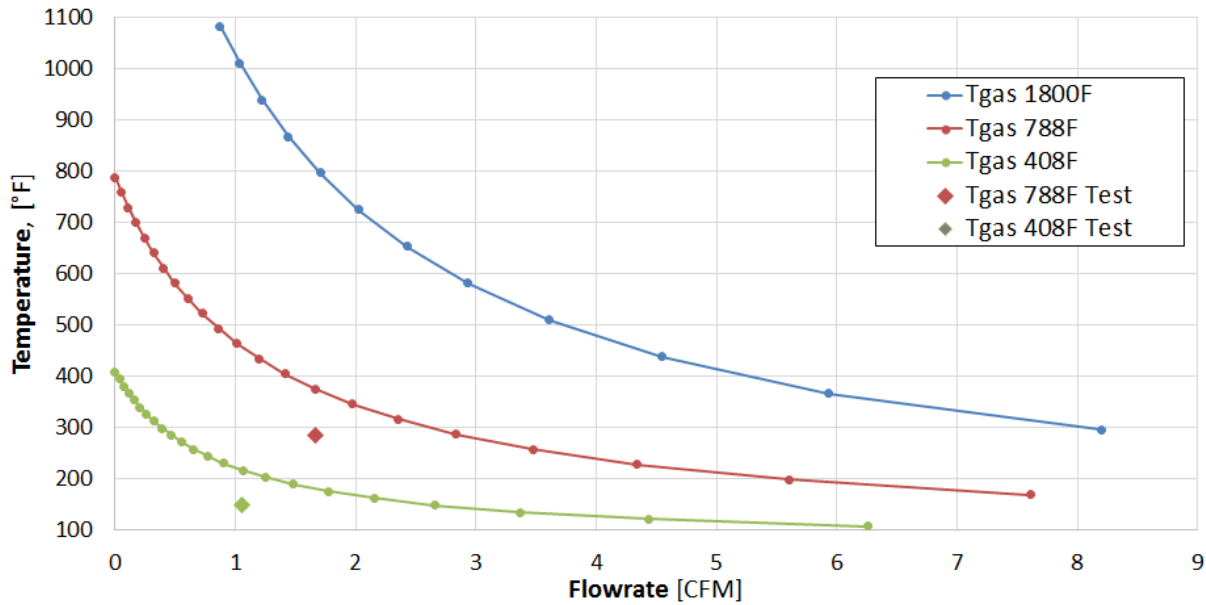


Figure 13. Thermal Test Results

Figure 13 is a graph of the calculated system outlet temperature for a range of flow rates and gas temperature. Since we could not test our heat exchanger at the actual operating point of 1800°F we tested at lower temperatures and developed a correlation between our testing temperature and the operating temperature. Figure 13 shows the two temperatures and flow rates we tested. Ideally, our test point would lie on the system curve. Since both test points lie below the calculated curve, our incoming air temperature will be less than calculated. This could be due to a number of reasons: pipes spaced too far from inner wall, thermal conductivity of refractory was lower than published, or the resistance calculated from our 1-D thermal resistance calculations were not a good approximation. However, after correlating test points to the operating point, we have found the air will come in at a minimum of 520°F degrees when operating at 1800°F. When this air is mixed with woodgas at a ratio of 1:1.58, the heat exchanger will provide the necessary 1110°F needed for autoignition.

Flow Test

Our second test was designed to calculate the total head loss of the corrugated piping. In our supporting analysis, many assumptions were made such as incompressible, one dimensional, steady state flow. Viscous friction losses were a major concern with the design and manufacturing of the heat exchanger and nozzle. To overcome the losses in the pipes, we placed a fan at the air inlet to provide the necessary pressure. However, the flow rate is dependent on the pressure loss seen in the corrugated pipes. For this reason, we designed our test to specify the threshold of pressure loss a fan must meet in order to ensure the needed air flow.

For our experiment, we used the PWM controlled fan to calculate the total head loss in our air system. Our experiment consisted of a small desk fan, a DC power supply, a San Ace 40 blower, and an Alnor anemometer.

To replicate the woodgas through the nozzle, we placed a fan, operating at constant speed, at the nozzle's entrance. Our flow rate through the corrugated piping was created using a blower, located at the entrance of our three corrugated pipes. However, to ensure the flow from the blower was distributed evenly between the

pipes, a 1.5” steel tube, encompassing the three 0.5” pipes, was used as a flow hood. Therefore, the San Ace blower was located at the entrance of the steel tube. The blower providing the air flow was powered by the DC power supply. Lastly, the anemometer, measuring the volumetric flow rate of the air, was placed at the exit of the nozzle to read the overall flow rates.

Initially, the flow test was designed to measure the head loss of the air system operating at different levels of fan power. However, we found that the San Ace fan, specified at a maximum pressure loss of 4.42 inches of water, did not have an operating range large enough to graph the head loss as a function of fan power. Therefore, the flow test was used to calculate the head loss of the pipes. Our results from the flow test can be seen below in Table 3.

The head loss was measured using Bernoulli’s equation for non-ideal circumstances. Bernoulli’s equation is displayed below.

$$\frac{P_1}{\rho g} + \frac{u_1^2}{\rho g} + z_1 = \frac{P_2}{\rho g} + \frac{u_2^2}{\rho g} + z_2 + h_L$$

Where P is pressure, g is the acceleration due to gravity, u is velocity of the air, z is the elevation with respect to a ground reference, ρ is density of air, and h_L is the head loss. Because the inlet and outlet of the pipes were at equal elevation and the velocity is considered constant in the pipe, the equation was simplified to:

$$\frac{P_1}{\rho g} - \frac{P_2}{\rho g} = h_L$$

Head loss is also equal to:

$$h_L = f \frac{L}{D} \frac{v^2}{\rho g} = \frac{16LQ^2}{\pi^2 D^5}$$

Where L is the length of the pipe, Q is the flowrate, and D is the diameter of pipe. Thus the friction losses are heavily dependent on diameter of the piping. A pipe with a diameter twice as large will have 32 times less friction losses. It is recommended to use a larger pipe diameter to reduce friction losses.

The following table summarizes our results and our initial test goals. The results reflect the San Ace blower’s insufficient ability to overcome the total head loss in the corrugated piping. Therefore, we recommend a fan with a larger pressure capacity. More specific details on this recommendation may be found in the “Future Iterations” section below.

Table 3. Flow Test Results

Blower at Max. Capacity	
Q _{air} Needed [cfm]	6.73
Q _{air} Supplied [cfm]	0.69
Fan Pressure Capacity [in. of H ₂ O]	4.15
Head Loss [in. of H ₂ O]	4.03

Maintenance and Repair Considerations

It is crucial to ensure that the system is properly maintained for the entirety of its life. For safety reasons it is advised that any maintenance to the gasifier is done after the system is turned off, and has cooled down.

One maintenance concern is to make sure that the gasifier is cleaned of any ash or remaining particles. A buildup of the residual flue particles can cause for an obstruction of flow in the system. To avoid buildup one should remove the ash catcher at the base of the main burning chamber and clean it out at least once every two runs. It is also advised to clear out the piping from any ash or sediments. If these particles are not cleared out the pipes will have a greater head loss.

A gasifier owner should also check monthly to ensure that there is no slag build up. One of the most susceptible places for slag to build up is in the nozzle and the hoses. A visual inspection will allow for the best results when looking for slag build up. Any minor build-up of slag can be treated by carefully using a chisel to chip away the slag. It is necessary to note that possibly chipping away at the refractory may develop further issues. Slag inspection should occur at least once a month to help reduce the amount of buildup in the system.

When maintaining a gasifier one should also look out for any cracks in the refractory. Any cracks in the refractory can cause a leakage of gas, which would minimize the effectiveness of the design and leak poisonous gases. We advise using a premixed refractory cement to fill in any cracks that may form. When using the premixed refractory cement, it is crucial to have the system sit for at least 24 hours to allow for it to cure and dry properly.

Concept Design Hazard Identification Checklist

When designing any new product, it is necessary to ensure that the product is safe for both the user and the environment. We have started a preliminary checklist listed in Table 4 below, including possible hazards that could arise in the use of a gasifier.

Table 4. Hazard Identification Checklist

Hazard	How to avoid hazards
High temperatures	Warning labels and insulation around high temperature components.
Backfire	Locking mechanism for lid and air sealed from primary combustion to exhaust
Poisonous Gas	Proper installation and sealed connections from combustion to exhaust
Pressurized Water/Steam	Locking mechanism to prevent the gasifier from being opened while running.
Carbon Monoxide Poison	Make sure the gasifier is not running in an enclosed area, such as a house or shed. Keep outside

Safety Considerations

As with any form of machinery, there are always safety actions that should be taken into consideration while building and operating the system. One of the biggest safety considerations is to be cautious of the high temperatures. Therefore, it would be advised to keep children and animals out of reach of the heating unit.

With the high heat of the system, the user should also take into consideration the time it takes for the system to cool down prior to any maintenance or repairs. With the extreme heats, the system will take time to cool down, and allow for a safe environment to work in.

When assembling the gasifier, the manufacturer should take caution while moving some of the subsystems. Depending on the nozzle size, the weight of the refractories can cause possible injury.

Future Iterations

Because viscous friction losses were greater than expected, some adjustments need to be made to our current design. To begin, a blower with a greater pressure head is needed to overcome the head loss. However, a bigger blower would draw more power, working adversely towards our low power consumption goal. A better alternative would be to use smooth piping of a larger diameter. In our design, we used corrugated stainless steel piping, because it could withstand the high temperatures and was made readily available by our sponsor. However, the piping network doesn't need to be permanent if the shape could be cast into place and the piping removed. A smooth piping network that could be dissolved or burned out would be a better alternative. Unfortunately, research into such materials yielded limited results.

Another possible design iteration uses a jet pump, similar to the design seen in Figure 14 below, as a replacement for the blower. Jet pumps (commonly known as an ejector, injectors, or venturi pumps) use high pressure fluid to pump low pressure fluid. These pumps are used in industry to prime pumps or when high pressure exhaust steam is available. They are usually inexpensive and reliable, as there are no moving parts.

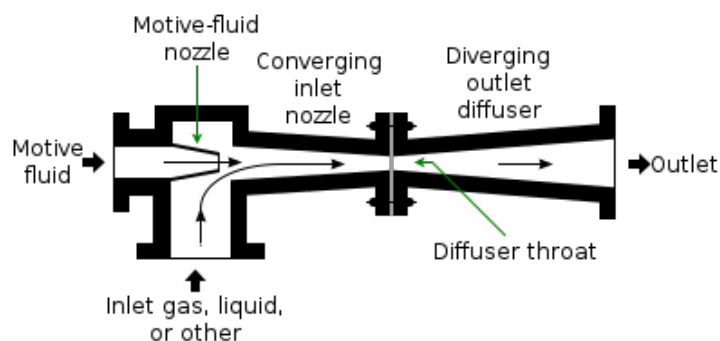


Figure 14. Jet Pump Section View

In our gasifier design, high pressure steam can be provided by the heat generated by primary combustion. A small section of the water jacket, as seen in Appendix D, can be sectioned off as the steam source. A jet pump can be placed on the exhaust end of the gasifier where steam would be the motive fluid and flue gases would be the inlet gas as seen in Figure 14. The low pressure created at the inlet would be sufficient to draw air into initial combustion chamber and across the hearth. The low exhaust pressure could also draw woodgas and secondary air through the nozzle. A jet pump's performance is dependent on geometry and fluid properties so calculating the needed flow rate of steam and pressure is not possible without manufacture

specifications. Most manufacturers do not publish their specifications as most are uniquely designed for specific uses. However, jet pumps are capable of reaching vacuum pressures as low as a tenth of a psi. For this design we only need about 10 inches of water total from primary combustion to exhaust, so a relatively small jet pump utilizing a small amount of steam would suffice. For small applications jet pumps range in price from \$20-\$60.

Summary

Over the last three quarters, we designed, built, and tested two components of a wood gasifier. In this process, we have calculated the dimensions needed to achieve the necessary flow rates, pressures, and temperatures for autoignition. In our building phase, we successfully constructed both the nozzle and heat exchanger designs using the castable refractory. We tested the effectiveness of our design, and based on our measurements, we detailed future iterations for a more successful design. To commence our project, we presented our work at the Senior Project Expo, held on December 1, 2016.

Lastly, we would each like to thank all of the people who donated their time and assistance to help us with this project. We would like to thank our sponsor, Tod duBois, for making his shop and resources available to us. To the Cal Poly Engineering Faculty, thank you for aiding us in our analysis, allowing us the use of laboratory equipment, and allowing us to raid your office hours. Thank you to the Mustang 60 and Hangar shop techs for donating some of your expertise. And lastly, thank you Dr. Davol for keeping us on track and advising us through the roadblocks.

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Appendices

A - Gasification Chemical Reactions and Equilibrium Constants

B - Gantt Chart

C - Concept Designs

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E - FMEA

F - Gas Analysis and Fluid Dynamic Hand Calculations

G - Fluid Dynamics Excel Model

H - EES Calculations and Results for the Second Iteration of the Heat Transfer Analysis

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J - Biomass Producer Gas Composition

K - Thermal Test Results

Appendix A - Gasification Chemical Reactions and Equilibrium Constants

Designation	Mechanism	ΔH (kJ/mol)	
Oxidation	$C_{(s)} + O_2 \rightleftharpoons CO_2$	-392,5	(1)
	$C_{(s)} + \frac{1}{2} O_2 \rightleftharpoons CO$	-110,5	(2)
Boudouard	$C_{(s)} + CO_2 \rightleftharpoons 2 CO$	172,0	(3)
Water Gas: primary secondary	$C_{(s)} + H_2O \rightleftharpoons CO + H_2$	131,4	(4)
	$C_{(s)} + 2 H_2O \rightleftharpoons CO_2 + 2 H_2$	90,4	(5)
Methanation	$C_{(s)} + 2 H_2 \rightleftharpoons CH_4$	-74,6 [†]	(6)
Water-gas shift	$CO + H_2O \rightleftharpoons CO_2 + H_2$	-41,0	(7)
Steam Reforming	$CH_4 + H_2O \rightleftharpoons CO + 3 H_2$	205,9 [†]	(8)
	$CH_4 + 2 H_2O \rightleftharpoons CO_2 + 4 H_2$	164,7 [†]	(9)
	$C_n H_m + n H_2O \rightleftharpoons n CO + (n + m/2) H_2$	210,1 ^{†‡}	(10)
	$C_n H_m + n/2 H_2O \rightleftharpoons n/2 CO + (m-n) H_2 + n/2 CH_4$	4,2 ^{†‡}	(11)
CO ₂ Reforming	$CH_4 + CO_2 \rightleftharpoons 2 CO + 2 H_2$	247,0 [†]	(12)
	$C_n H_m + n CO_2 \rightleftharpoons 2n CO + m/2 H_2$	292,4 ^{†‡}	(13)
	$C_n H_m + n/4 CO_2 \rightleftharpoons n/2 CO + (m-3n/2) H_2 + (3n/4) CH_4$	45,3 ^{†‡}	(14)
H ₂ Reforming	$CO + 3 H_2 \rightleftharpoons CH_4 + H_2O$	-205,9 [†]	(15)

Over a period of time these two reaction rates will tend to reach a common value and the gas composition will have reached a state of equilibrium. Under these circumstances

$$K_p = \frac{k_r}{k_f} = \frac{[CO_2] \cdot [H_2]}{[CO] \cdot [H_2O]}$$

where K_p is the temperature dependant equilibrium constant for the CO shift reaction. Assuming ideal gases this can also be expressed as

$$K_p = \frac{P_{CO_2} \cdot P_{H_2}}{P_{CO} \cdot P_{H_2O}} = \frac{v_{CO_2} \cdot v_{H_2}}{v_{CO} \cdot v_{H_2O}} \quad (2-10)$$

where P_{CO} is the partial pressure and v_{CO} is the volume fraction $\frac{P_{CO}}{P}$ of CO in the gas, and so on.

Similarly the equilibrium constants for the other reactions can be expressed as

$$K_p = \frac{P_{CO}^2}{P_{CO_2}} = \frac{(v_{CO})^2}{v_{CO_2}} \cdot P \quad (2-11)$$

for the Boudouard reaction (2-4),

$$K_p = \frac{P_{CO} \cdot P_{H_2}}{P_{H_2O}} = \frac{v_{CO} \cdot v_{H_2}}{v_{H_2O}} \cdot P \quad (2-12)$$

for the water gas reaction (2-5), and

$$K_p = \frac{P_{CO} \cdot P_{H_2}^3}{P_{CH_4} \cdot P_{H_2O}} = \frac{v_{CO} \cdot v_{H_2}^3}{v_{CH_4} \cdot v_{H_2O}} \cdot P^2 \quad (2-13)$$

for the reforming reaction (2-8), where P is the total absolute pressure of the gas.

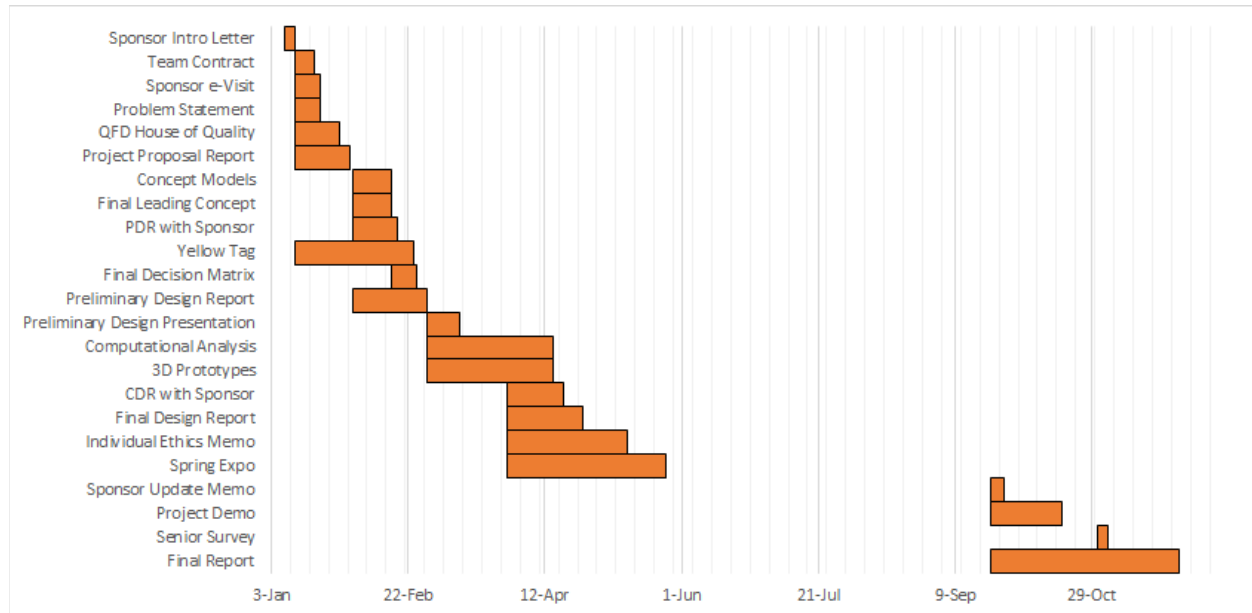
The temperature dependency of these equilibrium constants can be derived from fundamental data but are usually expressed as a correlation of the type

$$\ln (K_{p,T}) = \ln (K_{p,T0}) + f(T)$$

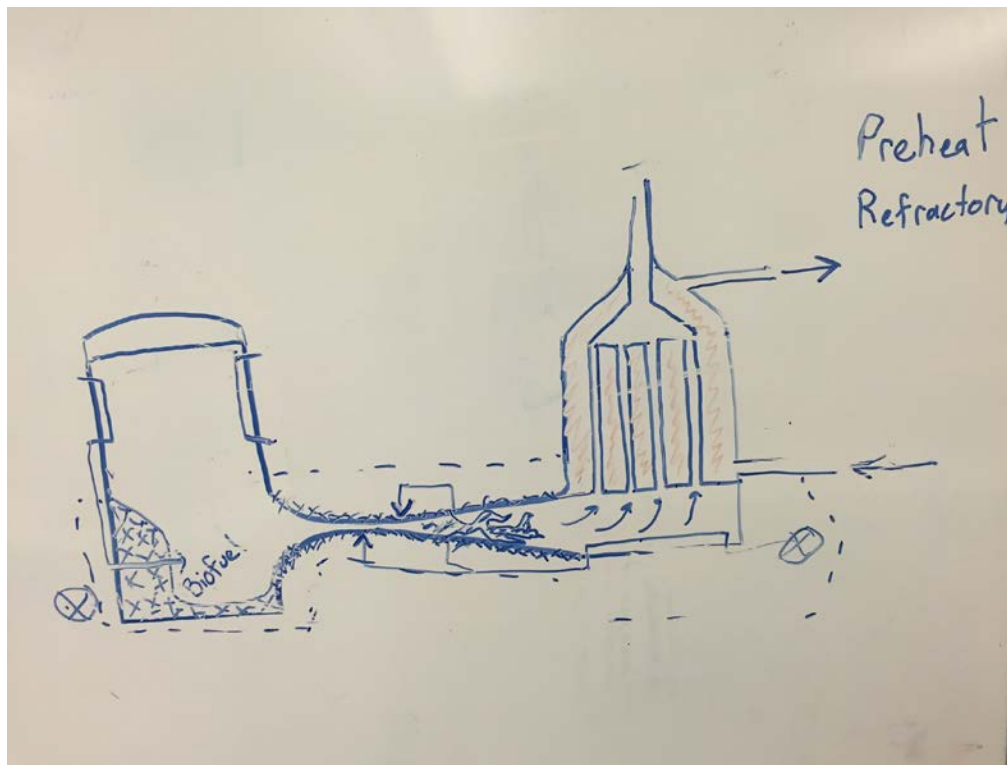
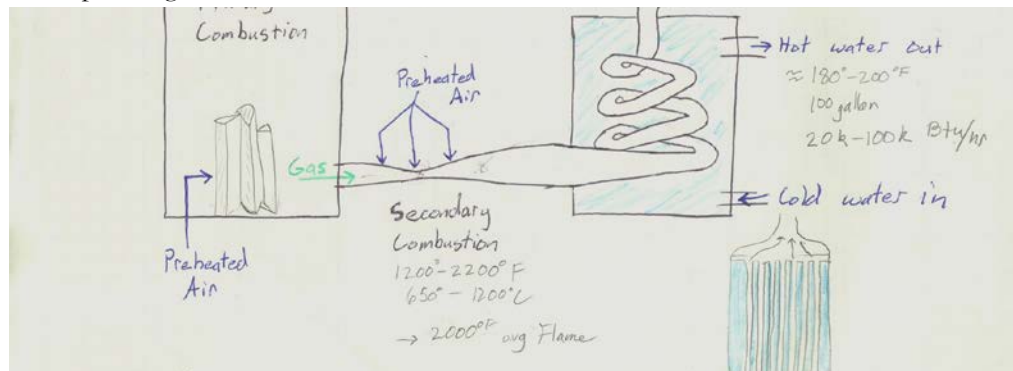
Appendix B - Gantt Chart

TASK	START DATE	DURATION	END DATE
Sponsor Intro Letter	8-Jan	4	12-Jan
Team Contract	12-Jan	7	19-Jan
Sponsor Visit	12-Jan	9	21-Jan
Problem Statement	12-Jan	9	21-Jan
QFD House of Quality	12-Jan	16	28-Jan
Project Proposal Report	12-Jan	20	2-Feb
Concept Models	2-Feb	14	16-Feb
Final Leading Concept	2-Feb	14	16-Feb
PDR with Sponsor	2-Feb	16	18-Feb
Yellow Tag	12-Jan	43	3-Mar
Final Decision Matrix	16-Feb	9	25-Feb
Preliminary Design Report	2-Feb	27	29-Feb
Preliminary Design Presentation	29-Feb	12	11-Mar
Computational Analysis	29-Feb	46	15-Apr
3D Prototypes	29-Feb	46	15-Apr
CDR with Sponsor	29-Mar	21	12-May
Final Design Report	29-Mar	28	3-May
Individual Ethics Memo	29-Mar	44	12-May
Spring Expo	29-Mar	58	26-May
Sponsor Update Memo	22-Sep	5	27-Sep
Project Demo	22-Sep	26	18-Oct
Senior Survey	31-Oct	4	3-Nov
Final Report	22-Sep	69	12-Dec

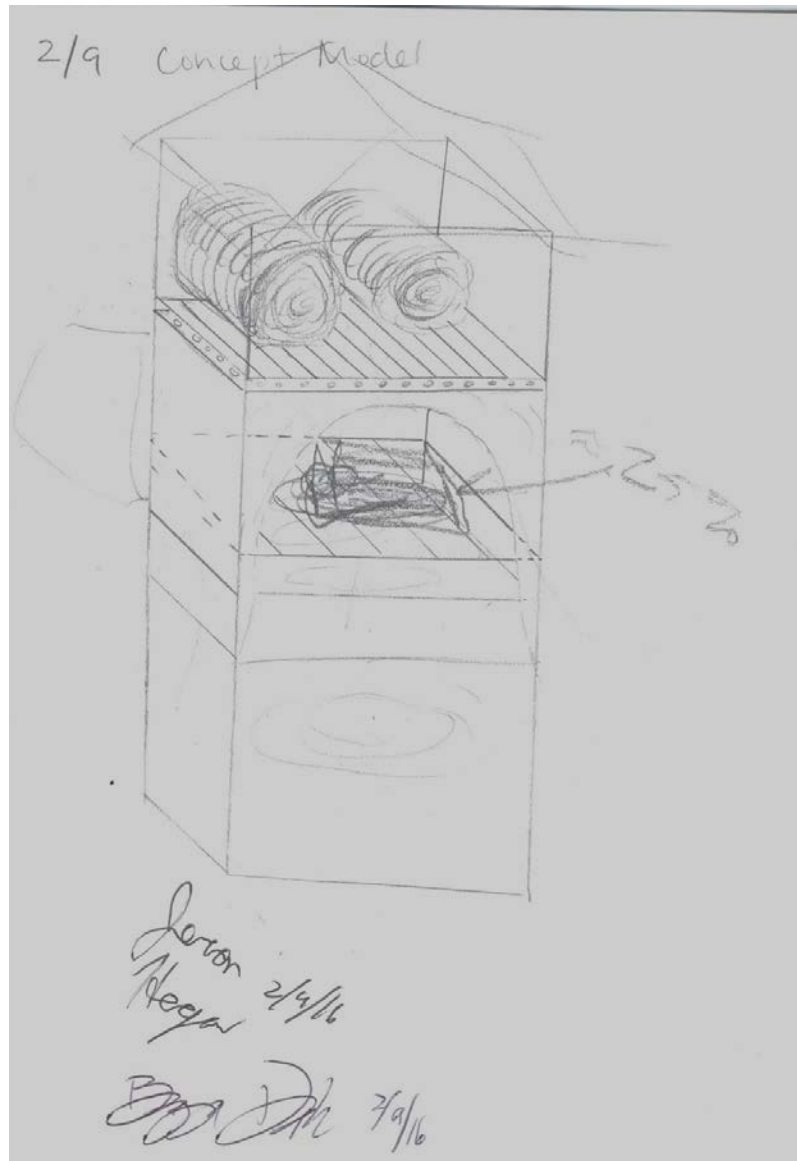
Appendix B - Gantt Chart



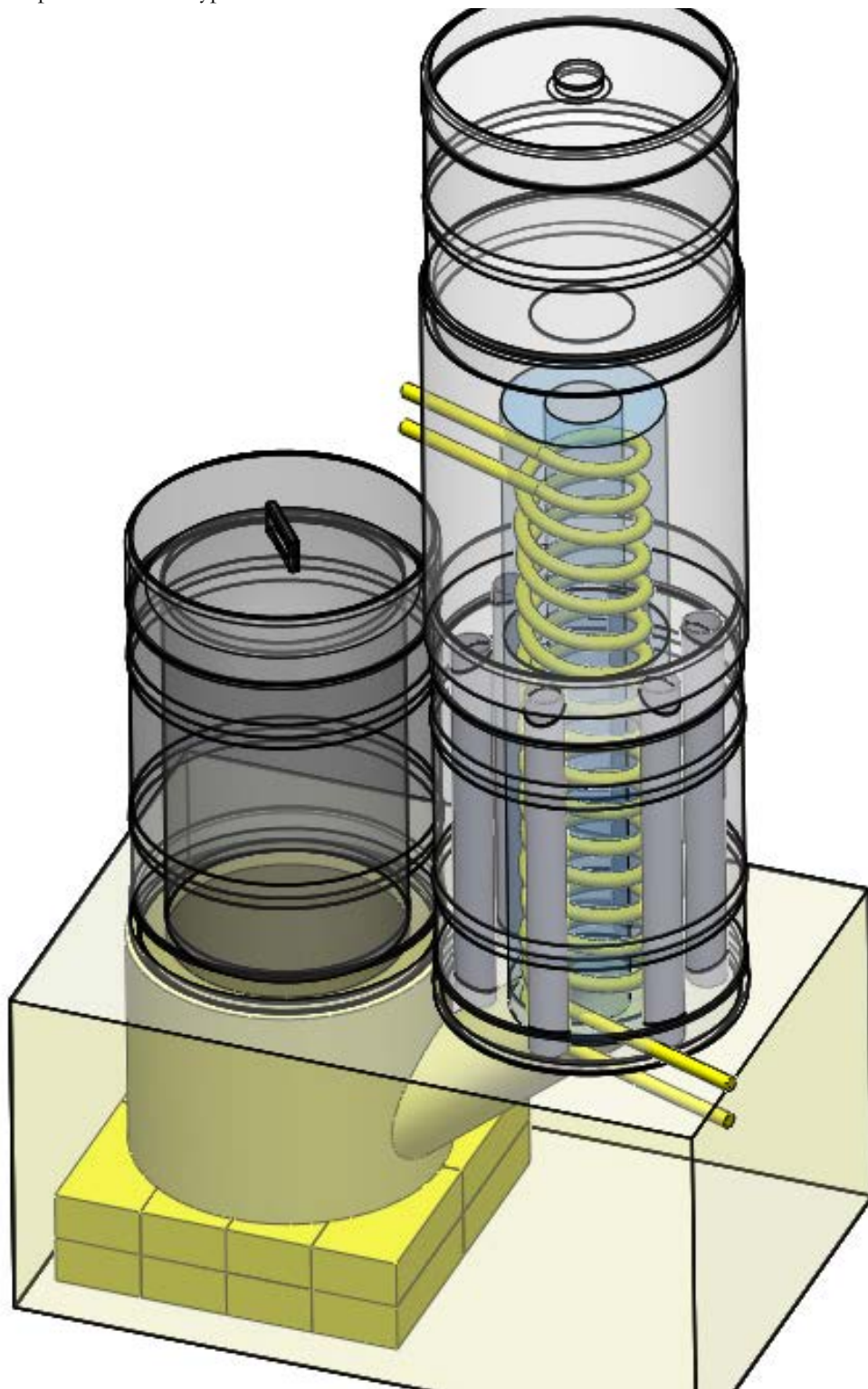
Appendix C - Concept Designs



Appendix C - Concept Designs



Appendix D - Sponsor's Prototype



Potential Failure Mode and Effect Analysis (Design FMEA)												
System								FMEA Number:				
Subsystem								Page 1 of 1				
Component								Prepared By:				
Model Year(s)/Vehicle(s):			Key Date:					FMEA Date (Orig.)	(Rev.)			
Core Team:												
								Action Results				
Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurrence	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Actions Taken	Severity	Occurrence	Criticality
System must meet the required temperature to obtain secondary combustion	Doesn't reach required temperature	Smoke comes out of the system rather than the clean air desired	7	Preheated air is not being introduced in the correct location	4	28	Redesign nozzle shape to allow for temperature to rise to desired shape that will maximize temperature	Bryan 5/3/2016				
Pressure difference to create a suction	No suction created	Harmful gases could not be sucked out of the system causing a build up, similar to a pressure vessel	6	Wrong dimensioning of the nozzle making it so that a vacuum is not possible	6	36	Ensure that the pressure drop is substantial enough to create a natural vacuum	Bryan 5/3/2016				
Preheated air in cooled piping	Improper location for the pipes to allow for maximum heat transfer	No production of the preheated air	3	Location of the pipes is too far/close to the heat source	5	15	Move the heat exchanger closer to the heat source to gain desired preheated air	Jeron 5/29/2016				
Creating heat exchanger within the refractory mold	The placement of the piping may be moved due to the weight of the refractory being poured on it	Heat exchanger no longer in the proper location	9	Improper placement and holding of the pipes	8	72	Create strong supports for the piping to help with the placement	Jeron 5/29/2016				
Refractory mold stability	The mold for the refractory has to be held together enough to allow for the	Wrong shape of the nozzle/casting	3	Weak supports	1	3	Use a more rigid form to help with the formation	Jeron 5/29/2016				
Burning of fuel properly	Creation of carbon monoxide that is contained in one area	Ensure that the gasifier is ran outside, since it is purposefully creating carbon monoxide	1	Placement in an enclosed area	1	1	Keep the gasifier outside	Jeron 5/29/2016				
Refractory strength	Refractory crumbles, or doesn't gain the desired shape	No cylindrical structure	1	Not allowing the refractory enough time to harden	1	1	Let the mold stay in the cardboard structure for longer than the required time	Jeron 5/29/2016				
Pressure Testing	Pressure measure testing not sufficient with our low pressure drop	Invalid data gathered	3	Poor measuring tools	2	6	Insure that the testing devices work prior to testing on the amchine	Courtney 5/29/2016				

Fluids Hand Calcs

- Heating goal $\sim 100,000 \text{ BTU/hr}$
- white pine wood $\sim 9,000 \text{ BTU/lb}$

$$9,000 \text{ BTU/lb} \times \frac{2.2045 \text{ lb}}{1 \text{ kg}} \times 10 \text{ kg/hr} = 198,405 \frac{\text{BTU}}{\text{hr}}$$

assuming: gasification & thermal efficiency $\sim 50\%$

$$E_{\text{useful}} = \eta E_{\text{fuel}} = (0.50)(198,405 \frac{\text{BTU}}{\text{hr}})$$

$$E_{\text{useful}} = 99,202.5 \frac{\text{BTU}}{\text{hr}}$$

so 10 kg/hr wood consumption reaches our goal

stoichiometric rate: $r = 6.5 \text{ kg air per } 1 \text{ kg fuel}$

declare $ER = 0.30$

$$\dot{m}_{\text{air primary}} = ER(r)(\dot{m}_{\text{fuel}}) = (0.30)\left(\frac{6.5 \text{ kg air}}{1 \text{ kg fuel}}\right)\left(10 \frac{\text{kg fuel}}{\text{hr}}\right)$$

$$\dot{m}_{\text{air}} = 19.5 \frac{\text{kg air}}{\text{hr}}$$

$$\dot{m}_{\text{woodgas}} = \dot{m}_{\text{fuel}} + \dot{m}_{\text{air}} = 10 \frac{\text{kg}}{\text{hr}} + 19.5 \frac{\text{kg}}{\text{hr}}$$

$$\dot{m}_{\text{woodgas}} = 29.5 \text{ kg/hr}$$

$$\dot{m}_{\text{air secondary}} = (1-ER)r(\dot{m}_{\text{fuel}}) = (1-0.30)\left(6.5 \frac{\text{kg fuel}}{\text{kg air}}\right)\left(10 \frac{\text{kg}}{\text{hr}}\right)$$

$$\dot{m}_{\text{air secondary ideal}} = 45.5 \text{ kg air/hr}$$

include
50% excess air

$$\dot{m}_{\text{air}} = 45.5 \frac{\text{kg air}}{\text{hr}} (1 + 0.50) = 68.25 \frac{\text{kg}}{\text{hr}}$$

1/4

2/4

assume

- incompressible flow
- steady state
- inviscid woodgas flow (but head loss in air pipes)
- velocity inlet to air pipes = 0 m/s
- 8.5' inlet air pipes (4 pipes in total)
- ρ_0 evaluated at 500°C
- STP for atmospheric conditions
- $P_1 = 101,325 \text{ Pa}$ - STD ATM

Woodgas Analysis
 assuming incompressible (Valid because $Ma \approx 0.0327$)

$$\dot{V}_1 = A_1 V_1 ; \quad V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{m}}{\rho A}$$

$$V_1 = \frac{\frac{29.5 \frac{\text{kg}}{\text{hr}}}{3600 \frac{\text{s}}{\text{hr}}} \frac{1}{0.95 \frac{\text{kg}}{\text{m}^3}}}{\frac{\pi}{4} (0.2032 \text{ m})^2} = 0.7925 \text{ m/s}$$

$$8'' = 0.2032 \text{ m}$$

$$1.5'' = 0.03810 \text{ m}$$

$$A_1 V_1 = A_2 V_2$$

$$V_2 = \frac{A_1 V_1}{A_2} = \frac{D_1 V_1}{D_2} = \frac{(0.2032 \text{ m})(0.7925 \text{ m/s})}{0.03810 \text{ m}}$$

$$V_2 = 22.543 \text{ m/s}$$

$$P_1 + \frac{1}{2} \rho V_1^2 + \rho g h^0 = P_2 + \frac{1}{2} \rho V_2^2 + \rho g h_2^0$$

assume: incompressible, steady, inviscid, negligible height difference,
 and atmosphere pressure at inlet

$$P_2 = P_1 + \frac{1}{2} \rho V_1^2 - \frac{1}{2} \rho V_2^2$$

$$P_2 = 101,135 \text{ Pa} + \frac{1}{2} (0.95 \frac{\text{kg}}{\text{m}^3}) (0.7925 \frac{\text{m}}{\text{s}})^2 - \frac{1}{2} (0.95 \frac{\text{kg}}{\text{m}^3}) (22.543 \frac{\text{m}}{\text{s}})^2$$

$$P_2 = 101,234 \text{ Pa}$$

Air Analysis

$$\dot{V}_0 = A v_0 ; \quad v_0 = \frac{\dot{V}_0}{A} = \frac{(68.25 \frac{\text{kg}}{\text{hr}}) (\frac{1 \text{ hr}}{3600 \text{ s}}) \times \frac{1}{4} \text{ inlets}}{\frac{\pi}{4} (0.0254 \text{ m})^2}$$

$$1'' = 0.0254 \text{ m}$$

$$v_0 = 7.7689 \text{ m/s}$$

$$H_L = f f \frac{L}{D} \frac{\rho v^2}{2} = (0.0206) \frac{8.5 \text{ ft}}{\frac{1}{12} \text{ ft}} \frac{(1.204 \frac{\text{kg}}{\text{m}^3}) (7.7689 \text{ m/s})^2}{2}$$

$$H_L = 76.34 \text{ Pa}$$

calculated
flowrate

$$P_A + \frac{1}{2} \rho v_A^2 + \rho g h = P_0 + \frac{1}{2} \rho v_0^2 + \rho g h + H_L$$

$$v_0 = \sqrt{(P_A - P_0 - H_L) \frac{2}{\rho}} = \sqrt{(101,325 \text{ Pa} - 101,234 - 76.34) (\frac{2}{0.4565})}$$

$$\rho(500^\circ\text{C}) = 0.4565$$

$$v_0 = 7.8638 \text{ m/s}$$

$$\dot{V} = A v = \frac{\pi}{4} (0.0254 \text{ m})^2 7.8638 \text{ m/s} (\frac{3600 \text{ s}}{1 \text{ hr}})$$

$$\dot{V}_{\text{calc}} = 57.379 \text{ m}^3/\text{hr}$$

$$\dot{V}_{\text{needed}} = 56.686 \text{ m}^3/\text{hr}$$

So a 8" inlet, 1.5" choke, & 4-1" inlet air pipes
will provide sufficient secondary flowrate

Appendix F. Gas Analysis and Fluid Dynamic Calculations

LengthAccelerometer	25	in				
	2.083333333	ft				
Tsyngas	1800	F	HARD			
Tair	1100	F	HARD			
					Total Heat Transfer	
Nozzle Convection					Total Resistance	C18+C23+C38
Spec.Vol. Syngas	1.053	m^3/kg	HARD		Q (Heat Transfer)	(C3-C4)/H6
densitysyngas	0.059285812	lbm/ft^3				
Vol. flowratesyngas	30	m^3/hr	HARD			
	17.65734	ft^3/min				
Mass flowratesyngas	1.046829739	lbm/min				
Dyn. Viscosity syngas (Tsyngas)	1.0022E-06	lbf*s/ft^2	HARD			
Reynolds Number	$4*C9/(PI()*D*C10)$					
Nusselt Number	4.36		HARD			
Therm. Cond. Syngas (Tsyngas)	0.0924	W/m-K	HARD			
	0.053423832	Btu/hr-ft^2-F				
Conv. Coeff. (hsyngas)	$C16*C14/D$					
Conv.Resistancesyngas	$(C17*pi()*D^2/4)^{-1}$					
Refractory Conduction						
Therm. Cond. Refractory	5.7	Btu-in/hr-ft^2-F	HARD			
	0.474999998	Btu/hr-ft-F				
Cond.Resistance refractory	$\ln(1+2*d/D)/(2*pi()*L*C22)$					
Preheated Air Conduction						
Pipe Diameter	0.5	in	HARD			
	0.04167	ft				
Pipe Area	0.00136	ft^2				
Dyn. Viscosityair (TAIR)	2.572E-05	lbm/hr-s	HARD (EES)			
Therm.Cond. Air (TAIR)	0.035148	Btu/hr-ft-R	HARD (EES)			
Mass flowrateair	73.125	kg/hr	HARD			
	0.04478	lbm/s				
Reynolds Number	53204.5					
Prandtl Number	0.7003	HARD				
Friction Factor	0.02066					
Nusselt Number	109.35					

Appendix F. Gas Analysis and Heat Transfer Calculations for First Design Iteration

Nozzle Diameter (D) [in]	D [ft]	Length [L] [ft]	Distance (d) [ft]	Conv. Coeff. (h _{syngas})	Conv.Resistance _{syngas}	Cond.Resistance _{refractory}	Conv.Resistance _{air}	Total Resistance	Q (Heat Transfer)	Maximum HT
1	0.083333333	2.5	0.0625	2.79513489	65.59486452	0.122806063	7.950868273	73.66853886	9.502020956	33.167489
1.1	0.091666667	2.5	0.0625	2.541031718	59.63169502	0.11528866	7.950868273	67.69785195	10.3400622	
1.2	0.1	2.5	0.0625	2.329279075	54.6623871	0.108685097	7.950868273	62.72194047	11.160369	
1.3	0.108333333	2.5	0.0625	2.150103762	50.45758809	0.102831537	7.950868273	58.5112879	11.96350354	
1.4	0.116666667	2.5	0.0625	1.996524922	46.85347466	0.097602322	7.950868273	54.90194525	12.75000361	
1.5	0.125	2.5	0.0625	1.86342326	43.72990968	0.0928992	7.950868273	51.77367715	13.52038407	
1.6	0.133333333	2.5	0.0625	1.746959306	40.99679033	0.088644074	7.950868273	49.03630267	14.27513825	
1.7	0.141666667	2.5	0.0625	1.644169694	38.58521442	0.084773972	7.950868273	46.62085667	15.01473911	
1.8	0.15	2.5	0.0625	1.552852717	36.4415914	0.081237482	7.950868273	44.47369715	15.73964039	
1.9	0.158333333	2.5	0.0625	1.471123626	34.52361291	0.077992161	7.950868273	42.55247334	16.45027762	
2	0.166666667	2.5	0.0625	1.397567445	32.79743226	0.075002627	7.950868273	40.82330316	17.1470691	
2.1	0.175	2.5	0.0625	1.331016614	31.23564977	0.072239122	7.950868273	39.25875717	17.83041672	
2.2	0.183333333	2.5	0.0625	1.270515859	29.81584751	0.069676421	7.950868273	37.8363922	18.50070684	
2.3	0.191666667	2.5	0.0625	1.215276039	28.51950631	0.067292981	7.950868273	36.53766757	19.15831104	
2.4	0.2	2.5	0.0625	1.164639538	27.33119355	0.06507029	7.950868273	35.34713211	19.80358683	
2.5	0.208333333	2.5	0.0625	1.118053956	26.23794581	0.062992338	7.950868273	34.25180642	20.43687832	
2.6	0.216666667	2.5	0.0625	1.075051881	25.22879405	0.061045206	7.950868273	33.24070752	21.05851686	
2.7	0.225	2.5	0.0625	1.035235145	24.29439427	0.05921673	7.950868273	32.30447927	21.66882166	
2.8	0.233333333	2.5	0.0625	0.998262461	23.42673733	0.057496229	7.950868273	31.43510183	22.26810028	
2.9	0.241666667	2.5	0.0625	0.963839617	22.6189188	0.055874282	7.950868273	30.62566135	22.85664926	
3	0.25	2.5	0.0625	0.93171163	21.86495484	0.054342549	7.950868273	29.87016566	23.43475453	
3.1	0.258333333	2.5	0.0625	0.901656416	21.15963372	0.052893613	7.950868273	29.1633956	24.00269192	
3.2	0.266666667	2.5	0.0625	0.873479653	20.49839516	0.051520861	7.950868273	28.5007843	24.56072762	
3.3	0.275	2.5	0.0625	0.847010573	19.87723167	0.05021837	7.950868273	27.87831832	25.10911857	
3.4	0.283333333	2.5	0.0625	0.822098497	19.29260721	0.048980825	7.950868273	27.29245631	25.64811287	
3.5	0.291666667	2.5	0.0625	0.798609969	18.74138986	0.047803436	7.950868273	26.74006157	26.17795019	
3.6	0.3	2.5	0.0625	0.776426358	18.2207957	0.04668188	7.950868273	26.21834585	26.69886208	
3.7	0.308333333	2.5	0.0625	0.755441862	17.72834176	0.04561224	7.950868273	25.72482227	27.21107235	
3.8	0.316666667	2.5	0.0625	0.735561813	17.26180645	0.04459096	7.950868273	25.25726569	27.71479735	
3.9	0.325	2.5	0.0625	0.716701254	16.81919603	0.043614807	7.950868273	24.81367911	28.21024633	
4	0.333333333	2.5	0.0625	0.698783723	16.39871613	0.04268083	7.950868273	24.39226523	28.6976217	
4.1	0.341666667	2.5	0.0625	0.681740217	15.99874744	0.041786331	7.950868273	23.99140205	29.17711931	
4.2	0.35	2.5	0.0625	0.665508307	15.61782489	0.040928841	7.950868273	23.609622	29.64892873	
4.3	0.358333333	2.5	0.0625	0.65003137	15.25461966	0.040106094	7.950868273	23.24559402	30.11323347	
4.4	0.366666667	2.5	0.0625	0.63525793	14.90792375	0.039316004	7.950868273	22.89810803	30.57021126	
4.5	0.375	2.5	0.0625	0.621141087	14.57663656	0.038556652	7.950868273	22.56606148	31.02003424	
4.6	0.383333333	2.5	0.0625	0.60763802	14.25975316	0.037826267	7.950868273	22.2484477	31.46286921	
4.7	0.391666667	2.5	0.0625	0.594709551	13.95635415	0.037123213	7.950868273	21.94434564	31.8988778	
4.8	0.4	2.5	0.0625	0.582319769	13.66559678	0.036445975	7.950868273	21.65291102	32.32821671	

Appendix G - Fluid Dynamics Excel Model

Parameter	Value	Units	Assumptions/Notes
Inlet Diameter	8 inch		
	0.203200406	meters	
Choke Diameter	2.5 inch		
	0.063500127	meters	
Inlet Air Pipe Diameter	0.5 inch		
	0.012700025	meters	
Number of inlet nozzles	3		
Stoichiometric Rate	6.5	kg air/kg fuel	
Equivalence Ratio	0.3		
CO fraction	0.2	Volume Fraction	Gas % by Volume
H2 fraction	0.2	Volume Fraction	
N2 fraction	0.52	Volume Fraction	
CO2 fraction	0.06	Volume Fraction	
Methane	0.02	Volume Fraction	
Total Gas %	100	% - check	
Grams per mole	23.5335832	g/mol	
Density	0.978874532	kg/m ³ - 20C	
	0.328534064	kg/m ³ - 600C	
	0.225302622	kg/m ³ - 1000C	
Woodgas specific volume	1.021581385	m ³ /kg - 20C	
	3.043824399	m ³ /kg - 600C	
	4.438474754	m ³ /kg - 1000C	
BTU Goal	20000	BTU/hr	
Efficiency (thermal and gasification)	50	%	
Needed BTU of fuel	40000	BTU/hr	
Fuel heating value	9000	BTU/lb	White Pine
	19841.58	BTU/kg	White Pine
Fuel consumption	2.015968486	kg/hour	
	0.000559991	kg/sec	
Air to primary	3.931138548	kg air/hr	Typical air provided with a 4" vac
	0.001091983	kg air/sec	
Total woodgas	5.947107035	kg woodgas/hour	
	0.001651974	kg woodgas/sec	
	0.005028319	m ³ woodgas/sec	Evaluated at density of 600C of woodgas

Appendix G - Fluid Dynamics Excel Model

Parameter	Value	Units	Assumptions/Notes
Air Specific Volume	3.607503608	m ³ /kg	Evaluated at 1000 C - Heated gasses leaving nozzle
	2.190580504	m ³ /kg	Evaluated at 500 C - Preheated air entering choke
	1.481481481	m ³ /kg	Evaluated at 250 C - Average air temperature through piping
	0.830564784	m ³ /kg	Evaluated at 20 C - Local atmospheric air temperature
Excess Air Percentage	50	%	
Secondary Air w/excess air	13.75898492	kg air/hr	
	0.00382194	kg air/sec	
	0.003174369	m ³ air/sec	Evaluated at 20C of air
	11.42772834	m ³ air/hr	
	6.726118028	CFM	Evaluated at 20C of air
			Assuming incompressible
Woodgas Velocity at nozzle inlet	0.155054303	m/s	
Woodgas Velocity at nozzle choke	1.58775606	m/s	
C _p	1.142	KJ/(Kg*K)	Evaluated at 1000C
C _v	0.855	KJ/(Kg*K)	Evaluated at 1000C
k = cp/cv	1.335672515		
R	286.9	J/(Kg*K)	
c	578.3921507	m/s	Evaluated at 600C of woodgas
Ma # of air at choke	0.00274512	Ma	Less than ~0.3 for incompressible flow
			Head Losses - evaluated at cold side
Equivalent Roughness	0.9		Riveted steel
Friction Factor	0.1258		
length of preheated air pipes	8.5	feet	
length/diameter	204		
velocity	8.352896339	m/s	Evaluated at 20C of air
head loss	1077.907474		Evaluated at 20C of air
Nozzle height from air inlet to exit	1	meter	
Stack Effect Draft Pressure	9.601021278	Pa	Evaluated at 1000C flue gas and 20C atmospheric
Vacuum pressure at nozzle inlet	4.4	in H2O	
	1094.896	Pa	
Local atmospheric pressure (stp)	101325	Pa	
Pressure of woodgas at nozzle inlet	100230.104	Pa	Atmospheric minus any available nozzle vacuum
Pressure of woodgas at nozzle choke	100220.0928	Pa	Woodgas at constant 600C - no heat loss/gain from inlet to choke
Pressure Difference from inlet to choke	10.01118414	Pa	
Calculated Flowrate at solved pressure	0.004133277	m ³ air/sec	Woodgas at constant 600C - no heat loss/gain from inlet to choke
	0.247996597	m ³ air/min	
	14.87979581	m ³ air/hr	
	8.75791716	CFM	
Needed Flowrate	0.003174369	m ³ air/sec	
	0.190462139	m ³ air/min	
	11.42772834	m ³ air/hr	
	6.726106957	CFM	
Percent Difference in flowrate	-30.20781881	%	Difference of calculated vs needed

Appendix H - EES Calculations and Results for the Second Iteration of the Heat Transfer Analysis

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EES Ver. 9.925: #0552: for use only by students and faculty, Mechanical Engineering, Dept. Cal Poly State University

Heat Transfer Calculations

$$k = k[\text{Air}, T = T_a]$$

$$h = 134.35 \cdot \frac{k}{0.0127 \text{ [m]}}$$

$$q = 0.019084 \cdot 1032 \cdot [T_a - 293.15 \text{ [K]}]$$

$$T_a = [982.22 + 273.15] \cdot 1 \text{ [K]} - \frac{q}{4 \cdot 3.5 \text{ [m]}} \cdot \frac{0.015 \text{ [m]}}{0.654 \text{ [W/m-K]} \cdot 0.0254 \text{ [m]}} + \frac{1}{h \cdot \pi \cdot 0.0254}$$

SOLUTION

Unit Settings: SI K kPa kJ mass deg

$$h = 556.8$$

$$q = 8347$$

$$k = 0.05264$$

$$T_a = 717$$

4 potential unit problems were detected.

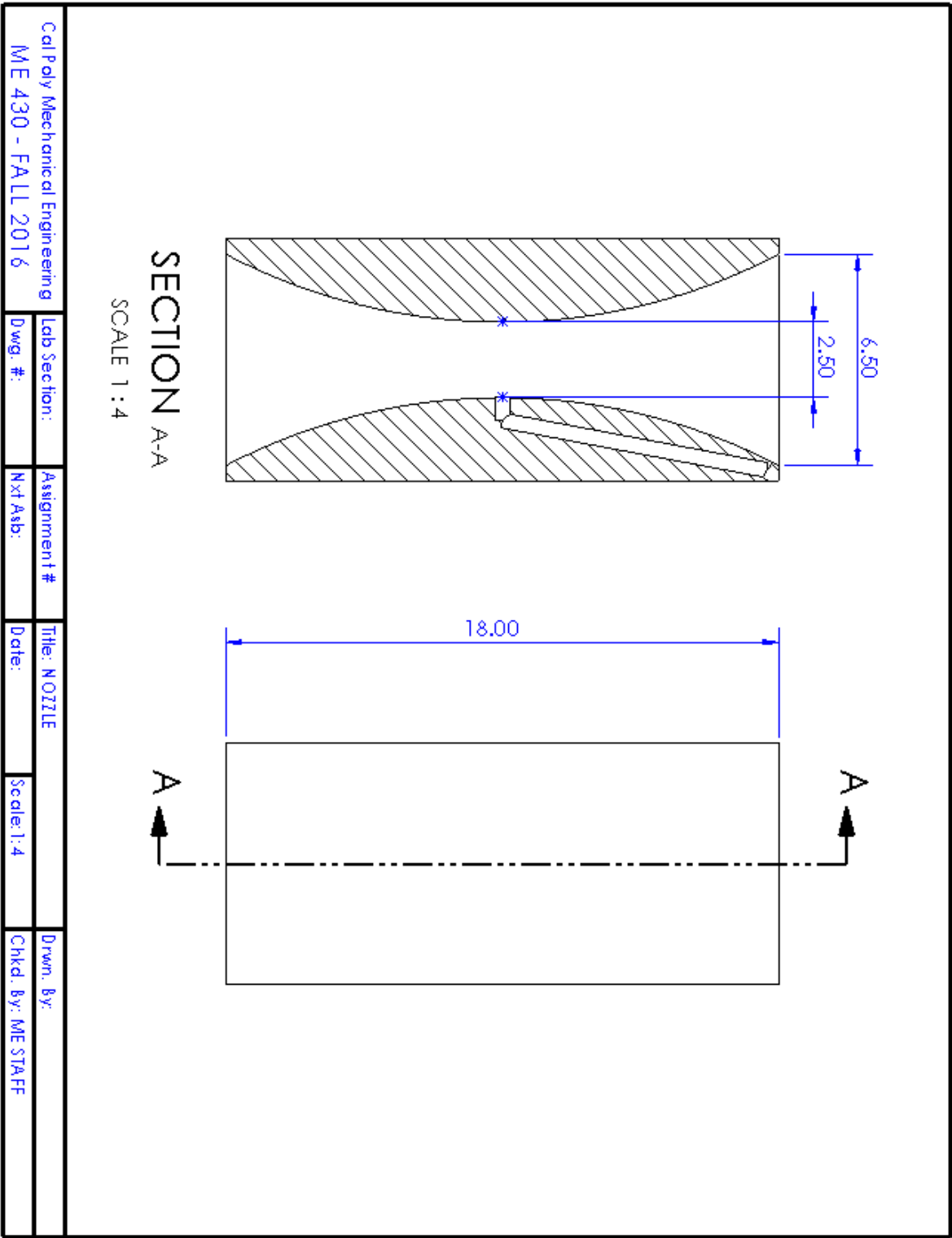


Table 4-2. Typical Properties of Producer Gas from Biomass^a

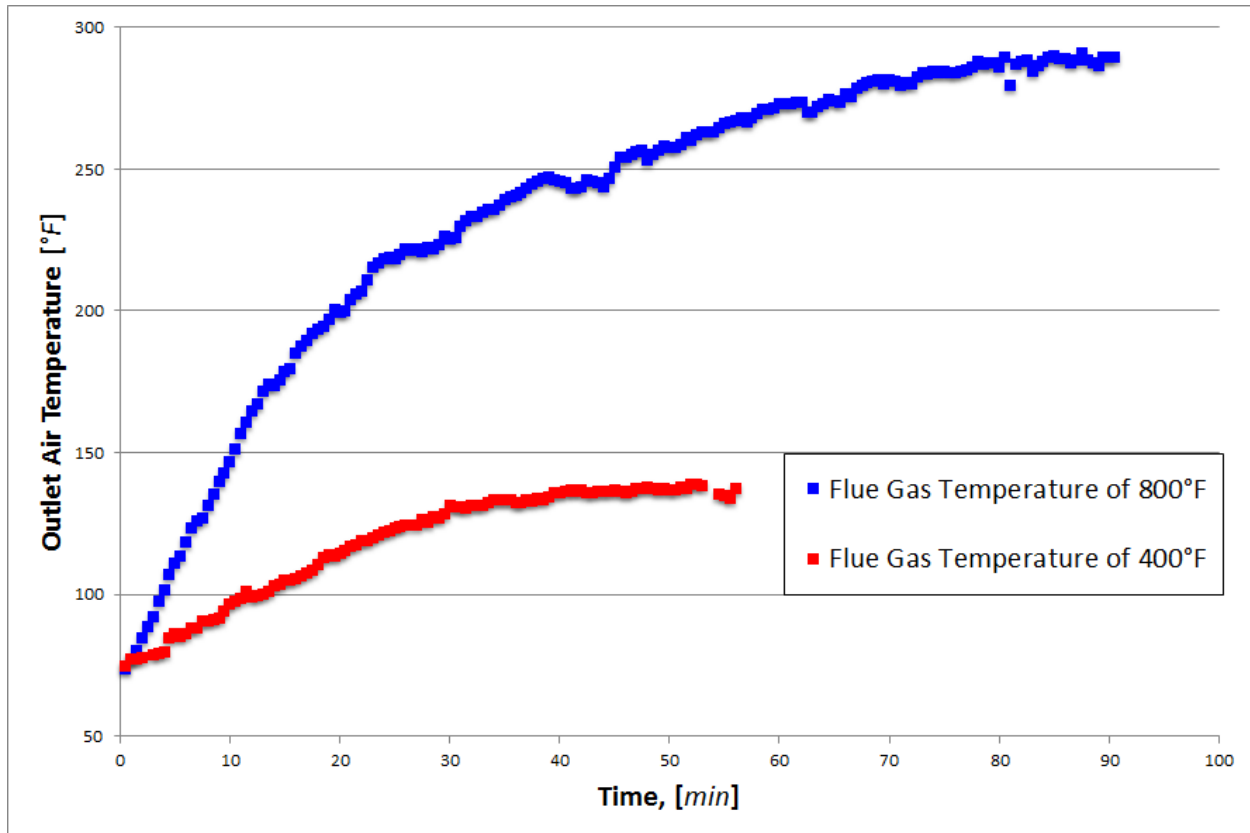
Compound	Symbol	Gas (vol.%)	Dry Gas (vol.%)
Carbon monoxide	CO	21.0	22.1
Carbon dioxide	CO ₂	9.7	10.2
Hydrogen	H ₂	14.5	15.2
Water (v)	H ₂ O	4.8	—
Methane	CH ₄	1.6	1.7
Nitrogen	N ₂	48.4	50.8
Gas High Heating Value:			
Generator gas (wet basis) ^b		5506 kJ/Nm ³ (135.4 Btu/scf)	
Generator gas (dry basis) ^b		5800 kJ/Nm ³ (142.5 Btu/scf)	
Air Ratio Required for			
Gasification:		2.38 kg wood/kg air (lb/lb)	
Air Ratio Required for			
Gas Combustion:		1.15 kg wood/kg air (lb/lb)	

^aThese values are based on ash- and moisture-free biomass with the composition given in Table 4-1. The wet-gas composition is the most important property of the gas for mass and energy balances, but the dry-gas composition is usually reported because of the difficulty in measuring moisture. The heating value of the gas is usually calculated from the gas composition, using a value of 13,400 kJ/Nm³ (330 Btu/scf) for H₂ and CO, and 41,900 kJ/Nm³ (1030 Btu/scf) for methane.

^bThese are typical values for downdraft air gasifiers, but they can vary between 4880 and 7320 kJ/Nm³ (120-180 Btu/scf), depending on variables such as gasifier heat loss, biomass moisture content, and char removal at the grate.

Source: Modified from data in Reed 1981.

Appendix K - Thermal Test Results



This graph of our thermal test shows two tests - one with the inner wall heat exchanger temperature at 400F and one at 800F. The blower was set to 1.0 CFM for each test. The outlet air temperature through the heat exchanger was measured in 30 second intervals until a steady state temperature was reached.