COMBINDED HEAT AND POWER FROM COTTON GIN TRASH THROUGH FLUIDIZED BED GASIFICATION: EVALUATION OF A HEAT EXCHANGER FOR WASTE HEAT CAPTURE

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Abstract

Every year, cotton gins accumulate tons of cotton gin trash that must be disposed of properly. However, gin trash is a waste biomass with a heating value of about 7000 Btu/lb that can be used as fuel for a fluidized bed gasification system to supply cotton gins with heat and power. In this study, a counter flow, finned heat pipe heat exchanger was designed and evaluated for a pilot scale gasifier to capture waste heat. To acquire preliminary data, exhaust gases from a propane burner were used as a surrogate for hot syngas. Ambient air was used as the cooling agent since heated air is used to dry cotton. Three flow rates of 50, 75 and 100 cfm were selected to evaluate the overall heat transfer between exhaust gas (heat released) and cooling air (heat captured). Results from this study revealed that as air flow rate increased, the temperature differential between the inlet and outlet decreased but the heat rate captured increased. For air flow rates of 50, 75 and 100 cfm, average heat capture was 119, 128, and 137 Btu/min, respectively. Conclusions from this study were used to operate the heat exchanger during gasification for validation. In addition, the design of the heat exchanger demonstrated that waste heat can be captured for cotton drying with minimal losses.

Introduction

In 2016, cotton gins in the United States ginned over 17 million bales of cotton (USDA, 2016). Depending on harvest method, an estimate of the amount of cotton gin trash (CGT) can be calculated that gets accumulated at a cotton gin. A picker, stripper with a cleaner, or stripper without a cleaner typically have about 150, 400, or 800 pounds of CGT per bale, respectively (Parnell et. al., 1977). The number of bales processed varies by gin; some cotton gins process up to or over 100,000 bales in a given season which results in thousands of tons of accumulated CGT (fig. 1). In most cases, the CGT is a waste product that cost the gin \$20 - \$50 per ton to dispose of properly. However, CGT is a biomass that has a heating value of 7000 Btu/lb (LePori et. al., 1981) and can be utilized as fuel for a fluidized bed gasification (FBG) system for combined heat and power generation for a cotton gin.

Figure 1. Piles of cotton gin trash near a cotton gin.

In order for cotton to be properly ginned, the incoming seed cotton is recommended to be dried to a moisture content (wet basis) of about 6% - 7% (Anthony and Mayfield, 1994), however, gins may dry the lint to as low as 4%. Generally, heated air is used to pneumatically convey the seed cotton throughout the gin's individual components (dryers, pre-cleaners, gin stands, etc.). A blower coupled with a natural gas or propane burner supplies the heated air, generally set to a maximum temperature of about 300°F to 350°F to prevent cotton fibers from becoming too brittle and becoming damaged. On average, gins require 0.2 MCF per bale of thermal energy to dry cotton (TCGA, 2016), however, this value can vary by up to a factor of 30 depending on the moisture content of the incoming seed cotton.

Thermal energy conversion processes, such as combustion and gasification, provide a substantial potential of waste heat energy due to the elevated reaction temperatures. In order to justify implementing equipment for heat capture, there must be a need for the thermal energy. Waste heat captured from the FBG system has the potential to either partially or fully supply the thermal energy to dry cotton. Sources of waste heat from the gasification system include utilizing exhaust gases from the engine/generator and cooling the hot syngas through a heat exchanger.

A counter flow, finned heat pipe heat exchanger (HPHE) was designed for a pilot scale FBG system fueled by cotton gin trash located at Texas A&M University. In this study, the HPHE was evaluated with exhaust gases from a propane burner as a surrogate for syngas to acquire preliminary data before gasification tests. Ambient air was used as the cooling agent since heated air will be used to supply the thermal energy for drying cotton. The objective of this study was to evaluate the heat exchanger's overall heat transfer between the air and exhaust gases by varying the flow rate of ambient, cooling air. Conclusions from this study were used to operate the HPHE during gasification to capture waste heat from the hot syngas. In addition, results from this study were used to aid the design of a HPHE for a larger FBG system for cotton drying.

Materials and Methods

Heat Pipe Heat Exchanger Design

A counter flow, finned heat pipe heat exchanger (HPHE), shown in figure 2, was designed to maximize the convective heat transfer between the hot exhaust gases and cooling air. The inner and outer tubes conveyed the exhaust gases and cooling air, respectively. Threaded pipe caps with bored holes were screwed to the outer tube to contain the air while allowing the inner tube to pass through. Two air inlets were constructed to increase turbulence of the air and increase contact between the air and fins. Three holes were drilled near both ends in the outer tube for the air inlets and outlet. Threaded couplings were welded to the holes such that threaded pipe could be connected.

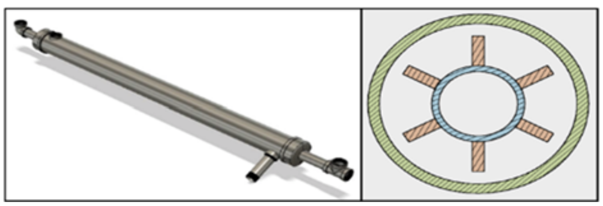

Figure 2. Final design of the finned HPHE (left) and sectional view of the inner pipe, outer pipe, and fins (right)

Extended surfaces, or fins, were welded on the outer surface of the inner pipe. The geometry of the fins play a significant role by increasing the effective surface area for convective heat transfer. A fin model was developed (Bergman et. al., 2011) as a basis for the design of the heat exchanger based upon the estimated heat transfer. Heat transfer was optimized by varying the dimensions of the fins in the HPHE through an iterative process.

Experimental Set Up

Heat exchanger experiments were conducted using a propane burner to produce exhaust gases as a surrogate for syngas. A compressor blower was used to supply air through the propane burner and HPHE while radial fans were used to supply ambient air. Orifice meters were used to measure the flow rates of air supplied by the compressor and fans. Thermocouples were placed at the exhaust gas inlet, exhaust gas outlet, and air outlet of the heat exchanger. A schematic of the set up can be seen in figure 3.

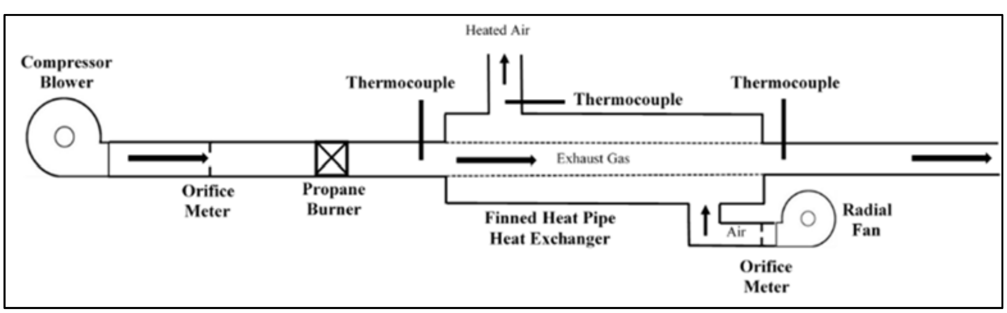

Figure 3. Set-up of heat exchanger experiments.

Data Collection and Evaluation

Barometric pressure, relative humidity, and ambient temperature were recorded from the National Weather Service's website (National Weather Service, 2017). Temperatures were measured with K-Type thermocouple probes connected to Jenco temperature displays (Model 765, Jenco Quality Instruments, San Diego, CA). The compressor blower was a Sutorbilt positive displacement compressor (Type L, Gardner Denver, Quincy, Illinois), while the radial fans were heavy duty blowers (Model HP33P, Blowers LLC, Elmhurst, IL). Volumetric flow rates were measured with orifice meters that were calibrated with a laminar flow element (Model Z50MC2-2, Meriam Process Technologies, Cleveland OH). Differential pressures were measured with Magnehelic gauges (Series 2000, Dwyer Instruments, Michigan City, IN).

Volumetric flow rates supplied by both the compressor blower and radial fans were calculated using equation 1. Mass flow rate was calculated by multiplying the volumetric flow rate and density of the air. The mass flow rate of the exhaust gases was assumed to be equal to that of the air supplied by the blower since the addition of propane, by mass, was negligible. The mass flow rate of exhaust gas was held constant and similar to that expected of the syngas from gasification throughout all tests.

$$V = 5.976 K {D_o}^2 \sqrt{\frac{\Delta P}{\rho_{air}}} \tag{1}$$

where
V = volumetric flow rate, cfm
K = dimensionless orifice meter constant
D_o = orifice diameter, in.
ΔP = pressure differential, in. water gauge
ρ_{air} = density of air, lb/ft^3.

Heat transfer rate was calculated using equation 2. Specific heats for the exhaust gas and air were estimated by taking the average temperature between the inlets and outlets. Specific heat of the exhaust gases was assumed to be equivalent to air at the elevated temperatures. Heat captured and heat released were defined as the heat transfer of the air and exhaust gases, respectively.

$$Q = \dot{m} c_p \Delta T \tag{2}$$

where
Q = heat rate, Btu/min
$\dot{m}$ = mass flow rate, lb/min
c_p = specific heat, Btu/lb-°F
ΔT = differential temperature, °F.

The HPHE was evaluated by varying the flow rate of ambient air. Three flow rates of 50, 75, and 100 cfm were selected due to the limited capacity of the fans. The inlet temperature of exhaust gases was maintained at 1000°F ± 50°F with a constant flow rate throughout each test. Each flow rate was tested for one hour in which data was collected every three minutes. Before each test was initiated, the inlet and outlet temperatures of the exhaust gases were allowed to stabilize for ten minutes. Three replicates were performed for each flow rate, resulting in a total of 9 tests. For each test, average heat rates (heat captured and heat released) were calculated once the system was in equilibrium and maintained a consistent rate.

Statistical Analysis

Tests performed in this study were a single-factor experiment with three levels of the factor (air flow rate) and three replicates. A randomized complete block design was performed to minimize experimental variability of the atmospheric air properties. Each block represented each replicate. An analysis of variance (ANOVA) single factor test was performed with a selected significance level (α) of 0.05.

Results and Discussion

All 9 heat exchanger tests revealed a similar trend between the heat captured and heat released rates. A sample plot can be seen in figure 4, which was the results for a flow rate of 75 cfm for the first replicate. The first 30 minutes revealed that the two heat rates approach each other, where the heat released eventually reaches heat captured. The reason for this trend is that heat was being stored within the HPHE prior to the initiation of a test. This is a typical relationship for a counter flow heat exchanger. Fluctuations in the heat released were a consequence of adjusting the propane pressure to maintain the exhaust gas inlet temperature. Over time, the two heat rates become relatively equal to each other, which was expected since all of the heat from the exhaust gases in the inner pipe is transferred to the cooling air in the outer pipe. The HPHE was insulated with ultra-high temperature ceramic fiber in which heat loss from the cooling air was almost negligible. The purpose of this study was to evaluate the heat exchanger during equilibrium, therefore, the heat rate distributions within the first 30 minutes were not further analyzed.

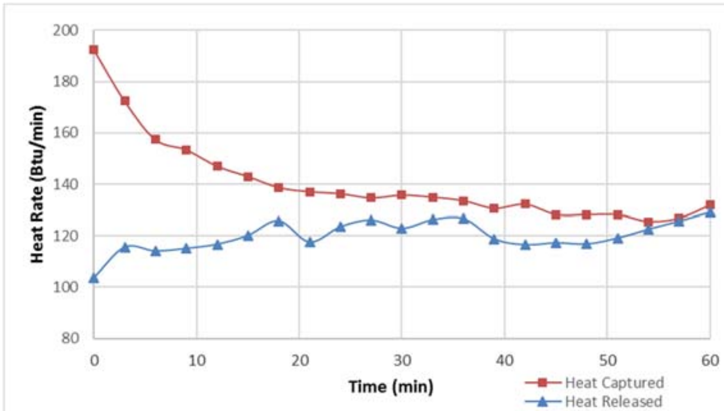

Figure 4. Heat rate results from an air flow rate of 75 cfm.

For each test, the system took approximately 30 minutes to achieve equilibrium where the heat rates stabilized. Average heat rates were calculated only from the values that were relatively consistent. In addition, temperature differential between the inlets and outlets of the cooling air were calculated and averaged. The results from all three replicates are summarized in table 1. As air flow rate increased, the temperature differential of the air decreased, i.e. outlet air temperature decreased. This is due to the residence time of the air flowing through the HPHE; longer residence time resulted in higher outlet temperature. Ambient air temperatures for all tests ranged between 69°F and 76°F. At air flow rates of 50, 75 and 100 cfm, the average heat captured rate was 119, 128 and 137 Btu/min, respectively. However, as air flow rate increased, heat rate captured by the air also increased even though the temperature differential decreased. From this observation, it appears that flow rate of air outweighs temperature differential for the flow rates selected in this study. These conclusions were supported by the data of heat released of the exhaust gases. As air flow rate was increased, heat released was also increased.

Table 1. Summary of HPHE results from all nine tests.

Flow Rate [cfm]	Avg. Air Temp. Differential (Std. Dev.) [°F]	Avg. Heat Captured (Std. Dev.) [Btu/min]	Avg. Heat Released (Std. Dev.) [Btu/min]
50	133 (3.0)	119 (3.6)	118 (5.6)
75	96 (2.2)	128 (2.5)	125 (4.4)
100	76 (1.6)	137 (2.3)	130 (5.2)

Statistical Results

The results of the ANOVA test revealed that the three air flow rates selected in this study significantly affect the heat captured by the air. The calculated p-value was 1.65E-46, which was much less than the selected significance level of 0.05. Therefore, the null hypothesis was rejected in which the means of heat captured rates between the three flow rates were not equal.

Summary

A counter flow, finned HPHE was designed and constructed for a pilot-scale FBG system, fueled by CGT, to capture waste heat for cotton drying. Exhaust gases from a propane burner were used as a surrogate for the hot syngas while ambient air was used as the cooling agent. Three air flow rates of 50, 75 and 100 cfm were selected to evaluate the overall heat transfer between the heat released by the exhaust gases and heat captured by the air. As air flow rate was increased, the air temperature differential decreased but the overall heat captured rate increased. Since the objective was to capture as much waste heat as possible, an air flow rate of 100 cfm was selected when operating the HPHE during gasification. Results from this study were used to aid the design of a larger FBG system for cotton drying at a cotton gin. The design of the HPHE demonstrated that almost all of the waste heat can be captured for cotton drying with minimal heat losses.

Acknowledgments

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