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Recovery of water from boiler flue gas

Christopher Samuelson
Lehigh University

Samuelson,
Christopher

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RECOVERY OF WATER FROM BOILER FLUE GAS

by

CHRISTOPHER SAMUELSON

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Dr. Edward K. Levy

Thesis Advisor

Dr. Herman F. Nied

Chairperson of Department

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Abstract

New methods are needed to recover water and heat from boiler flue gas in order to reduce overall plant water intake, increase efficiency, and reduce harmful emissions. This project involved design and testing of a condensing heat exchanger for use in pilot scale testing with an oil or coal fired boiler. The heat exchanger design described in this report consists of vertical cylindrical tubes in cross-flow, with cooling water on the inside of the tubes and the flue gas flowing on the outside of the tubes. The heat exchanger was split into stages so that acid can be condensed in the early stages, while the water will be condensed in the later stages. Testing was done using flue gas provided by Lehigh University's steam generation plant, which burns No. 6 fuel oil. The results from six tests show that 80% of the total moisture present in the flue gas was condensed and recovered. The sulfuric acid measurements showed that 80% of the sulfuric acid mist was condensed from the flue gas. The sulfuric acid was collected mostly in the early stages of the heat exchanger, while the water condensed separately in the later stages of the heat exchanger.

1.0 Introduction

Every fossil fuel power plant needs a massive amount of water to operate. Some plants take in up to 50 million gallons of water each day [5]. For some power plants, it can be difficult to obtain enough cooling water for its processes. Other plants are looking to limit their impact on the environment and the local marine life. Yet power plants release a large amount of water vapor into the atmosphere through the stack. The flue gas is a potential source for obtaining much needed cooling water for the plant.

Water is a natural product of combustion. Depending on the fuel being burned, the flue gas can contain up to 20% moisture (by volume). If a power plant can recover and reuse a significant portion of this moisture, it can lower its total cooling water intake requirement. The most practical way to recover this water is to use a condensing heat exchanger. In doing so, the plant can also recover lost heat. A plant's overall thermal efficiency directly depends on how much heat it releases to the atmosphere. In addition, harmful acids exist in the flue gas which are normally released through the stack. Some of these acids can conceivably be condensed out along with the water. The benefits of condensing the flue gas are threefold:

1. Recover Water
2. Recover Heat
3. Reduce emissions

The goal of this project is to create a new and cost effective design for a condensing heat exchanger, to be used in coal fired power plants. The heat exchanger will recover moisture from the flue gas, improve efficiency, and reduce harmful emissions.

2.0 Theory

2.1 Sources of Water

The water that exists in flue gas comes from three sources. First, water is a product of combustion. Any fossil fuel has hydrogen, and the hydrogen is oxidized to form water. Second, water is present in the fuel. Coal can consist of up to 40% moisture, and this moisture evaporates during combustion. Third, moisture is present in the air used for combustion (humidity). Figure 2.1 shows the total moisture content of the flue gas produced by different fuels.

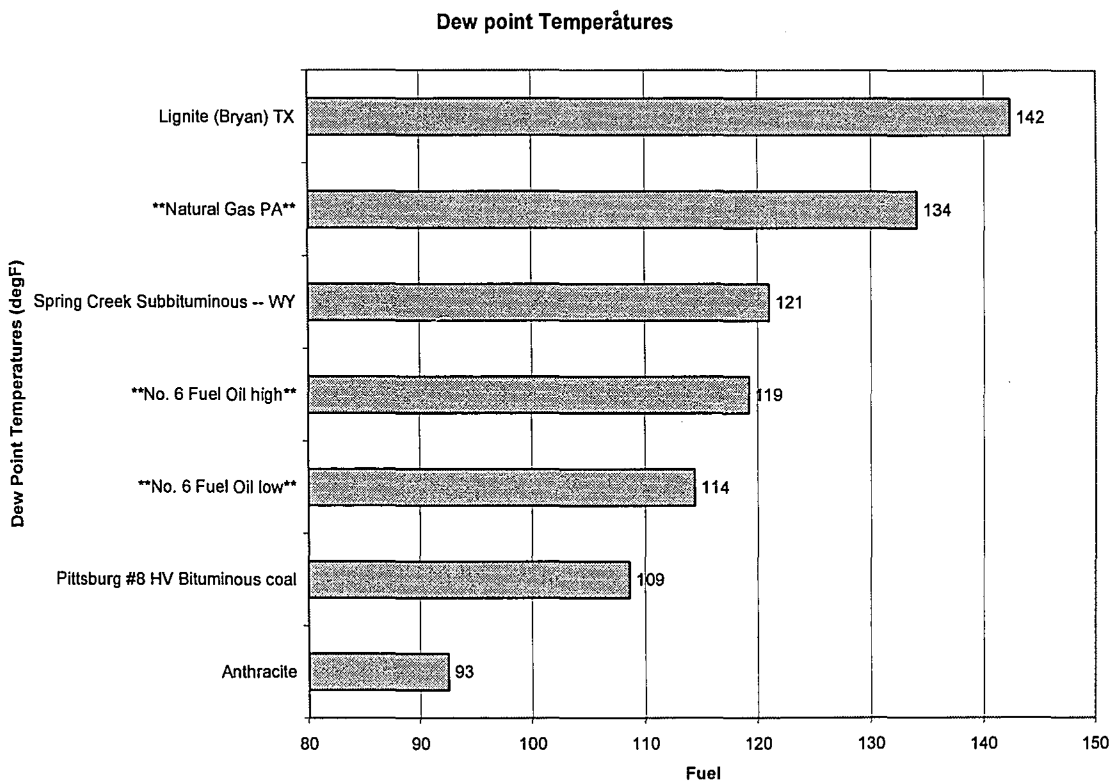

Figure 2.1. Flue Gas Moisture Content by Fuel

It can be seen from the figure that plants which burn a high moisture coal (such as a lignite or subbituminous coal) or natural gas, can release a significant amount of water through the stack.

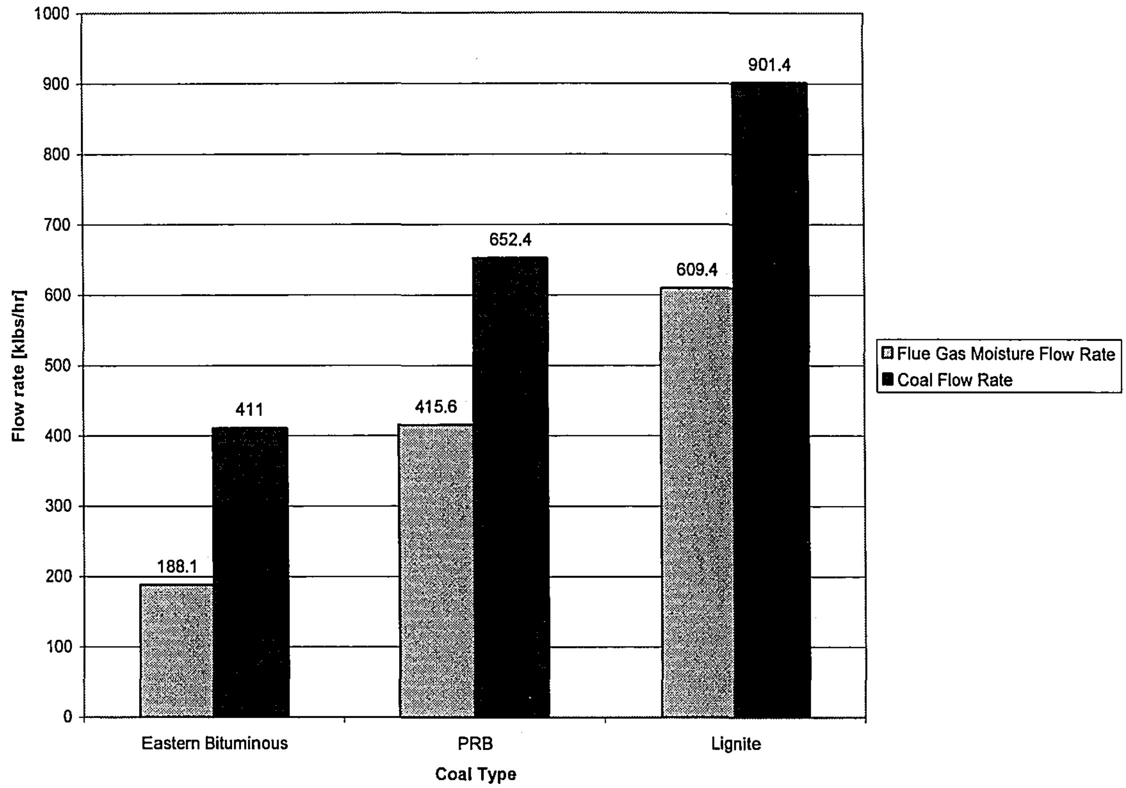

Figure 2.2. Typical Coal and Flue Gas Moisture Flow Rates for a 600 MW Power Plant [3]

Figure 2.2 shows typical coal and flue gas moisture flow rates for a 600 MW power plant.

In comparison, a typical cooling tower in a 600 MW unit has an evaporation rate of about 1.6 million lbs/hr. So, if even 80% of the water in the flue gas can be recovered, then a lignite plant could substitute 30% of its total cooling tower water requirements with recycled water from the flue gas (20% for PRB).

2.2 *Psychrometrics*

2.2.1 Water

In order to recover the water from the flue gas, a condensing heat exchanger will be used to cool the gas-vapor mixture to a temperature below the dew point of the water. Once water begins to condense out, the gas-vapor mixture is said to be saturated. This means that the gas is holding the maximum amount of moisture possible at that temperature and pressure. The dew point of water varies according to the volumetric percentage of water in the flue gas. Figure 2.3 shows how the dew point of water varies according to the water content.

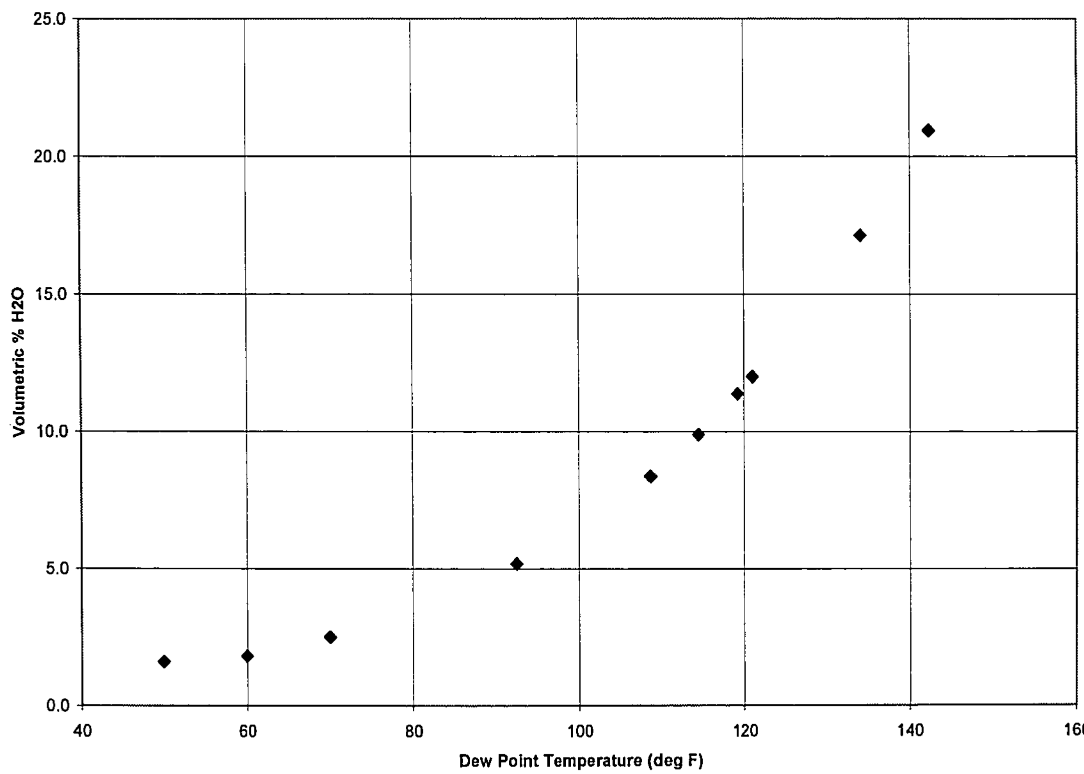

Figure 2.3. Dew Point of Water as a Function of Volumetric Percentage of Water in Flue Gas (Atmospheric pressure) [3]

Using this principle, the amount of water to be removed can be achieved only if the flue gas can be cooled to a certain temperature. For example, a plant burning No. 6 Fuel oil with a flue gas moisture content of 10% by volume wants to recover water. In order to condense 80% of the total moisture by volume, the outlet flue gas must have a moisture content of 2% by volume. According to Figure 2.3, the flue gas must be cooled to a temperature below 70°F in order to condense the desired amount of water. Therefore, the amount of water that can be condensed out of the flue gas is restricted by the ability to cool the flue gas to a certain temperature. This suggests that water recovery from flue gas may only be practical for plants that burn fuel which has a high moisture content.

2.2.2 Acids

Similar to the condensation of water, acids present in flue gas will also be condensed out with respect to their individual dew points. Three acids will be considered in this report: sulfuric acid (H_2SO_4), nitric acid (HNO_3), and hydrochloric acid (HCl).

Flue gas from coal or oil contains small quantities of SO_4^{2-}/H_2SO_4 in the vapor phase (the maximum expected range is 20 to 40 ppmv), depending on the characteristics of combustion and the amount of sulfur in the fuel. The dew point temperature of the SO_4^{2-}/H_2SO_4 depends on this concentration, as well as the amount of water present in the flue gas. For the scope of this project, the expected dew point of the SO_4^{2-}/H_2SO_4 in the flue gas is 280°F to 300°F [3].

The dew points of the other two acids are determined by the same factors. For HNO_3 the expected dew point range is from 65°F to 110°F. For HCl, the expected dew point range is from 100°F to 120°F [3].

The design of the condensing heat exchanger for this project takes advantage of the higher dew point temperature of sulfuric acid. As the flue gas cools, the first vapor to condense out will be the sulfuric acid. Later in the process, water will condense out at roughly the same temperature as both HNO_3 and HCl. For this reason, the heat exchanger will be built into sections so that the sulfuric acid condensate can be collected separately from the water and other acids. This will assist in purification of the flue gas condensate to ensure that it can be used in other processes in the plant. However, it may present some corrosion issues. These topics will be discussed in greater detail later in the report.

2.3 Efficiency

The outlet temperature of the flue gas directly affects the overall thermal efficiency of the plant. If the proposed recovered heat is used somewhere else in the plant, the overall thermal efficiency of the plant will be improved by using heat exchangers to recover water from the stack gas.

2.4 Heat Transfer

The heat exchanger to be modeled will be a shell-and-tube, counter flow system. It will consist of banks of tubes in cross-flow in which, cooling water is flowing on the inside of the tubes, and the flue gas is flowing over the outside of the tubes. It will recover both sensible and latent heat.

2.4.1 Log Mean Temperature Difference Method

The Log Mean Temperature Difference was used to design and develop the stages of the heat exchanger. The main equations are as follows:

$$Q = UA\Delta T_{lm} \tag{2.1) [1}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_2 / \Delta T_1)} \tag{2.2) [1}$$

where, Q is the heat transfer, U is the overall heat transfer coefficient, A is the overall heat transfer surface area, and ΔT_{lm} is the log mean temperature difference.

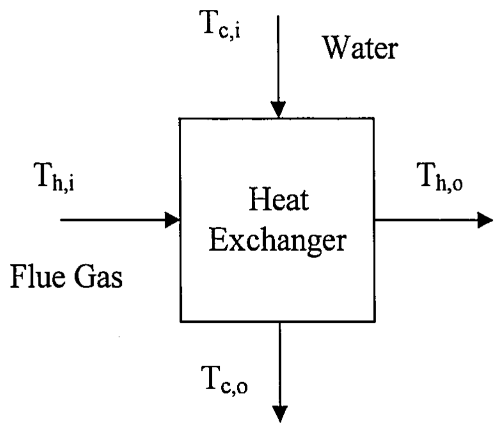

Figure 2.4. Heat Exchanger Flow Diagram

In Figure 2.4, $T_{h,i}$, etc. refer to the temperatures of the hot or cold fluid, at the inlet or the outlet. The hot fluid is the flue gas, and the cold fluid is the cooling water. The temperature differences are then implemented in the Log Mean Temperature difference as follows:

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{2.3) [1}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{2.4) [1}$$

The log mean temperature difference method was developed for either parallel flow or counter flow heat exchangers. However, this study utilizes a cross flow heat exchanger, so a correction factor must be applied [1].

$$\Delta T_{lm,c} = F \cdot \Delta T_{lm} \tag{2.5)[1}$$

The overall heat transfer coefficient for a tube in cross flow is calculated by the following equation, assuming no fouling:

$$\frac{1}{UA} = \frac{1}{U_i A_i} = \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{\ln(D_o / D_i)}{2\pi k L} + \frac{1}{h_o A_o} \tag{2.6) [1}$$

A_o and A_i are the outside and inside surface areas, respectively. D_o and D_i are the outside and inside diameters of the tube in cross flow. "L" is the length of the tube, and "k" is the thermal conductivity of the tube material. "h_i" and "h_o" are the inner and outer convection coefficients.

The design approach involved an iterative process where, the UA needed was calculated and a configuration was chosen with respect to duct dimensions, tube size, number of tubes, etc. The heat transfer properties were then calculated, including the UA supplied by the proposed configuration. The UA needed was compared with the UA supplied, and adjustments were made until the two matched.

2.4.2 Mass and Energy Balance

In order to calculate the UA needed, the total heat transfer must be known. Since there will be condensation in some of the heat exchangers, both sensible and latent heat must be considered. The heat transfer can be derived from a control volume analysis.

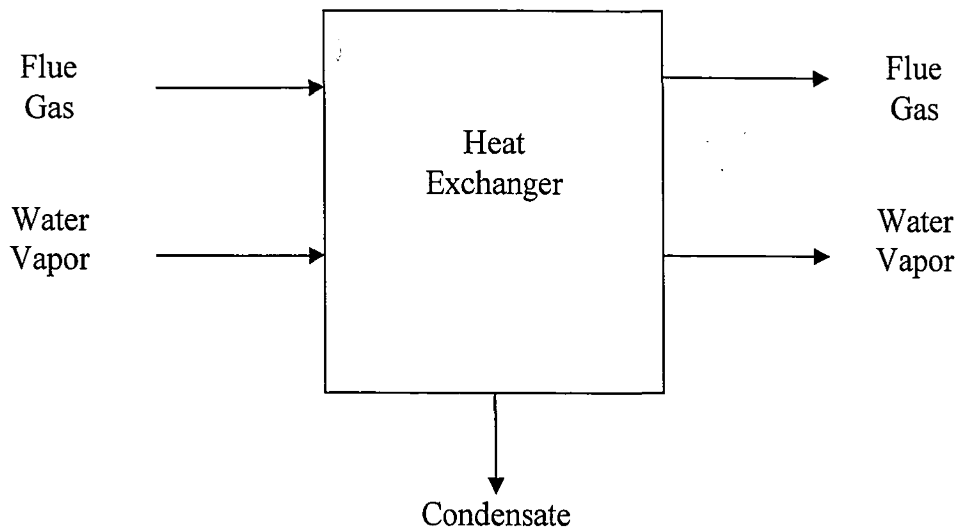

Figure 2.5. Control Volume Analysis

Through conservation of mass, the following equations can be derived:

$$\dot{m}_{dfg,in} = \dot{m}_{dfg,out} \tag{2.7}$$

$$\dot{m}_{v,in} = \dot{m}_{v,out} + \dot{m}_{cond} \tag{2.8}$$

$\dot{m}_{dfg,in}$: mass flow rate of dry gas in

$\dot{m}_{dfg,out}$: mass flow rate of dry gas out

$\dot{m}_{v,in}$: mass flow rate of water vapor in

$\dot{m}_{v,out}$: mass flow rate of water vapor out

$\dot{m}_{cond}$: mass flow rate of condensate out

Through conservation of energy, the heat transferred to the cooling water can be derived:

$$Q_{fg} = \dot{m}_{dfg}(h_{dfg,in} - h_{dfg,out}) + \dot{m}_{v,in}h_{v,in} - \dot{m}_{v,out}h_{v,out} - \dot{m}_{cond}h_{cond} \tag{2.9}$$

Q_{fg} : heat transfer from flue gas

$h_{dfg,in}$: enthalpy of dry gas in

$h_{dfg,out}$: enthalpy of dry gas out

$h_{v,in}$: enthalpy of vapor in
$h_{v,out}$: enthalpy of vapor out
h_{cond} : enthalpy of water condensate out

Equation 2.9 shows the formulation for the heat transferred from the flue gas. In order to do an energy balance, it must be determined to where the heat is transferred. The objective is to transfer all of the heat to the cooling water running through the heat exchanger tubes. However there will be some losses to the surrounding environment.

$$Q_{fg} + Q_{cw} + Q_{loss} = 0 \tag{2.10}$$

Q_{cw} : heat transfer to cooling water
Q_{loss} : heat transfer lost to surroundings

2.4.3 Condensation

In order to design the heat exchanger, Q_{fg} must be calculated. The known parameters for the design are:

1. Flue gas flow rate
2. Flue gas inlet and outlet temperature and pressure
3. Cooling water inlet temperature and pressure
4. Flue gas inlet moisture content

The only unknown left in Equation 2.9 is the condensate mass flow rate. It is assumed that for each heat exchanger that produces condensation, the exiting flue gas will be saturated. This means that the relative humidity of the flue gas exiting the heat exchanger is 100%. The inlet moisture content is specified by volumetric ratio (Φ).

$$\Phi = \frac{\dot{V}_v}{\dot{V}_{wfg}} \tag{2.11}$$

Where $\dot{V}_v$ is the volumetric flow rate of the vapor, and $\dot{V}_{wfg}$ is the volumetric flow rate of the wet flue gas. The total pressure of the flue gas can be broken down into the sum of the partial pressures of the vapor and dry flue gas:

$$P_{fg} = p_v + p_{dfg} \tag{2.12}$$

The moisture content is equivalent to the ratio of the partial pressure of the vapor to the total pressure of the wet flue gas, assuming the ideal gas law applies [4].

$$\Phi = \frac{p_v}{P_{fg}} \tag{2.13}$$

Assuming that the pressure of the wet flue gas is at atmospheric pressure, the partial pressure of the vapor can be calculated, given Φ. The humidity ratio is defined as the mass of water vapor divided by the mass of dry flue gas [4].

$$\omega = \frac{\dot{m}_v}{\dot{m}_{dfg}} \tag{2.14) [4}$$

Using the ideal gas law, this ratio can be expressed in terms of partial pressures and molecular weights.

$$\omega = \frac{\dot{m}_v}{\dot{m}_{dfg}} = \frac{M_v p_v \dot{V} / \overline{R}T}{M_{dfg} p_{dfg} \dot{V} / \overline{R}T} = \frac{M_v p_v}{M_{dfg} p_{dfg}} \tag{2.15) [4}$$

Combining Equations 2.12 and 2.15 results in:

$$\omega = \frac{M_v}{M_{dfg}} \frac{p_v}{(P_{fg} - p_v)} \tag{2.16) [4}$$

Combining Equations 2.13 and 2.16 results in

$$\omega = \frac{M_v}{M_{dfg}} \frac{\Phi}{1 - \Phi} \tag{2.17}$$

Because the flue gas vapor exiting the heat exchanger can be assumed to be saturated, if the exit temperature and pressure are known, the partial pressure of the vapor is equal to the saturation pressure of the vapor. For example, flue gas exiting a condensing heat exchanger at 100°F contains saturated water vapor also at 100°F. Therefore, the partial pressure of the vapor can be found in a table containing properties of saturated water. In this case, the partial pressure is 0.95 psi [4].

At this point, humidity ratios can be calculated for both the inlet and the outlet of the heat exchanger, given the following known properties:

1. Inlet volumetric moisture content
2. Molecular weights for water and flue gas
3. Total pressure of flue gas
4. Outlet temperature of flue gas

By conservation of mass, the condensate flow rate is

$$\dot{m}_{cond} = \dot{m}_{dfg}(\omega_{in} - \omega_{out}) \tag{2.18}$$

Where the mass flow rate of the wet flue gas is comprised of vapor and dry flue gas:

$$\dot{m}_{wfg} = \dot{m}_{dfg} + \dot{m}_{v} \tag{2.19}$$

Combining Equations 2.14, 2.18, and 2.19 leads to

$$\dot{m}_{cond} = \dot{m}_{wfg,in}\frac{\omega_{in} - \omega_{out}}{\omega_{in} + 1} \tag{2.20}$$

Combining Equations 2.14, 2.18, 2.19, and 2.9 leads to the final equation from which the total heat transfer can be calculated:

$$Q_{fg} = \frac{\dot{m}_{wfg,in}}{1+\omega_{in}}\left[(h_{dfg,in} - h_{dfg,out}) + \omega_{in}h_{v,in} - \omega_{out}h_{v,out} - (\omega_{in} - \omega_{out})h_{cond}\right] \tag{2.21}$$

The only unknowns left in equation 2.6 are the convection coefficients and the heat transfer surface areas. The heat transfer required is known from equation 2.21. Now is the time to select a configuration for the heat exchanger, calculate convection coefficients, and calculate the UA and try to match it to the UA required.

2.4.4 Gas Side Convection Coefficient

The heat exchanger will be modeled as banks of cylindrical tubes in cross flow. A relationship must be determined between the characteristics of the flue gas flow and the geometry of the tubes. The Nusselt number for a cylinder in cross flow is:

$$\overline{Nu_D} = \frac{\bar{h}D}{k} \tag{2.22}$$

Where $\bar{h}$ is the gas side convection coefficient, D is the diameter of the cylinder, and k is the thermal conductivity of the gas. There are numerous empirical formulations for calculating the mean Nusselt number. For this study, the Zhukauskas correlation was used for banks of tubes in cross flow:

$$\overline{Nu_D} = C \cdot \mathrm{Re}_D^m \, \mathrm{Pr}^{0.36} \left(\frac{\mathrm{Pr}}{\mathrm{Pr}_s} \right)^{1/4} \qquad (2.23)[1]$$

The Zhukauskas correlation is valid over the following range of conditions:

$$N_L \geq 20$$
$$0.7 < \mathrm{Pr} < 500 \qquad [1]$$
$$1000 < \mathrm{Re}_{D,max} < 2 \times 10^6$$

Where $\mathrm{Re}_{D,max}$ is the maximum Reynold's number based on the diameter of the cylinder:

$$\mathrm{Re}_{D,max} = \frac{v_{max} D}{\upsilon} \tag{2.24}$$

v_{max} is the maximum velocity through the banks of tubes, and υ denotes kinematic viscosity.

In Equation 2.23, Pr is the Prandtl number, and Pr_s is the Prandtl number evaluated at the surface temperature of the cylinder. N_L is the number of banks of tubes in the direction of the flow. A correction factor may be applied for a set of tubes with less than 20 banks of tubes, according to Table 2.1.

$$\overline{Nu_D}\Big|_{(N_L<20)} = C_2 \cdot \overline{Nu_D}\Big|_{(N_L \geq 20)} \qquad (2.25)[1]$$

N_L	1	2	3	4	5	7	10	13	16
Aligned	0.70	0.80	0.86	0.90	0.92	0.95	0.97	0.98	0.99
Staggered	0.64	0.76	0.84	0.89	0.92	0.95	0.97	0.98	0.99

Table 2.1 Correction factor C2 for Equation 2.26 for NL < 20 (ReD > 103) [1]

The constants C and *m* in Equation 2.23 can be found in Table 2.2, depending on the configuration and Reynold's number.

Configuration	$Re_{D,max}$	C	*m*
Aligned, (S_T/S_L > 0.7)	10^3 - 2 x 10^5	0.27	0.63
Staggered, (S_T/S_L < 2)	10^3 - 2 x 10^5	$0.35(S_T/S_L)^{1/5}$	0.60
Staggered, (S_T/S_L > 2)	10^3 - 2 x 10^5	0.40	0.60
Aligned	2 x 10^5 - 2 x 10^6	0.021	0.84
Staggered	2 x 10^5 - 2 x 10^6	0.022	0.84

Table 2.2. Constants of Equation 2.25 for the tube bank in cross flow [1]

"Staggered" and "Aligned" refers to the configuration of subsequent banks of tubes. Figure 2.6 shows the difference, as well as the dimensions S_T, S_L, and S_D.

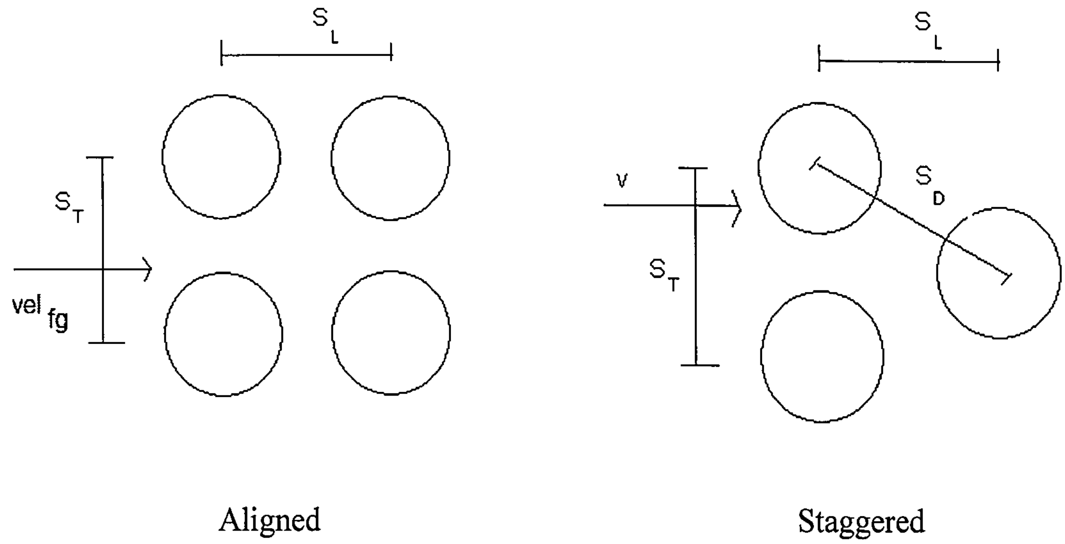

Figure 2.6. Cross Section of Tube bank configurations [1]

For flow where $Re_{D,max} < 1000$, the Nusselt number can be approximated by flow over a single cylinder. For this, the Hilbert correlation can be used:

$$\overline{Nu_D} = C \cdot \mathrm{Re}_D^m \mathrm{Pr}^{1/3} \qquad (2.26)[3]$$

C and m are constants determined by Table 2.3.

$Re_{D,max}$	C	m
0.4 - 4	0.989	0.330
4 - 40	0.911	0.385
40 - 4000	0.683	0.466

Table 2.3. Constants of Equation 2.27 for the circular cylinder in cross flow [1]

Once the Nusselt number has been calculated, the gas side heat transfer convection coefficient can be calculated from Equation 2.22.

2.4.5 Water Side Convection Coefficient

The water side convection coefficient will be determined by modeling the system as flow through a straight cylinder. The Nusselt number for internal flow largely depends on whether the flow is laminar or turbulent. Both cases will be considered for this project.

For turbulent flow, the Gnielinski correlation will be used.

$$Nu_D = \frac{(f/8)(\mathrm{Re}_D - 1000)\mathrm{Pr}}{1 + 12.7(f/8)^{1/2}(\mathrm{Pr}^{2/3} - 1)} \qquad (2.27)[1]$$

For conditions:

$0.5 < \mathrm{Pr} < 2000$

$3000 < \mathrm{Re}_D < 5 \times 10^6$

In Equation 2.27, f is a friction factor determined by the Moody diagram, Re_D is the Reynold's number for the flow of the water inside the tube (see Equation 2.28), and Pr is the Prandtl number evaluated at the bulk temperature of the water.

$$\mathrm{Re}_D = \frac{vD}{\upsilon} \qquad (2.28)$$

The diameter used in Equation 2.28 is the inner diameter of the tube, and v is the bulk fluid velocity. Once the Nusselt number is known, the water side convection coefficient can be calculated from Equation 2.22.

For laminar, fully developed flow, the Nusselt number for internal flow through a circular tube can be approximated as a constant. Assuming uniform surface heat flux,

$$Nu_D = 4.36 \qquad (2.29)[1]$$

Because of the nature of the heat exchanger, the flow may not be fully developed throughout the tube. For low Reynold's numbers,

$$Nu_D = 1.86\left(\frac{\mathrm{Re}_D \mathrm{Pr}}{L/D}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \qquad (2.30)[1]$$

where L is the length of the tube and μ_s is the viscosity evaluated at the surface.

Once the inside and outside convection coefficients for the geometry are known, UA can be calculated by Equation 2.6. Adjustments should be made to the geometry or the outlet gas temperature until the UA supplied equals the UA required.

2.4.6 Condensation effect on heat transfer

In general, water condensing on the surface of a tube in cross-flow will improve the heat transfer. For design purposes, this is neglected in this study to ensure a higher factor of safety.

3.0 Design

3.1 Scope of Design

The scope of this project was to design and build a multi-stage heat exchanger in order to recover water and heat, and to extract acids from the flue gas of a fossil fuel power plant. The heat exchanger was built on the campus of Lehigh University and installed at the on-campus steam generation plant. The unit burns either natural gas or No. 6 fuel oil. A slip stream of flue gas was extracted after the economizer and routed through the heat exchanger assembly.

3.2 Design Specifications

The design criteria for the heat exchanger are listed in the following table.

Flue Gas Inlet Temperature	450	degrees F
Flue Gas Outlet Temperature	90	degrees F
Flue Gas Flow rate	300	lb/hr
Flue Gas Inlet Moisture Content	10	% (vol)
Flue Gas Outlet Moisture Content	5	% (vol)
Cooling Water Inlet Temperature	55	degrees F

Table 3.1. Design Specifications

3.3 Design Approach

The proposed solution [3] consists of three stages of heat exchangers. The purpose of the first stage will be to condense out sulfuric acid. This stage will need to have some type of resistance to corrosion. The second stage will be a "buffer stage", in order to ensure that all of the acid has been removed. The third stage is the condensing stage, where the majority of the water will be recovered.

There are three main components to the design: the duct, the heat exchanger, and the supporting structure. Each will be briefly examined in this section.

3.4 Ductwork

It was necessary to build ductwork in order to transport the flue gas from the exit of the economizer to the part of the plant where the heat exchanger was to be built. First, an adapter to the existing ductwork was needed. For this, a custom built, sheet metal door adapter was built which transitioned into a round duct fitting.

A damper gate was fixed to this section in order to control the flow rate of the flue gas.

From the economizer adapter and damper gate, a high temperature, corrosion resistant, insulated, flexible hose duct was used to carry the flue gas to the heat exchanger.

After the heat exchanger, an eight foot straight length of PVC pipe was used to create a laminar, fully developed flow region (temperatures below 100 degrees Fahrenheit were expected here). In this section a pitot tube was implemented to measure the pressure drop due to the flow velocity.

After the pitot tube, an induced draft fan was used to draw the flue gas through the heat exchanger system.

After the fan, another flexible hose duct was used to exhaust the remaining flue gases to the roof. All ductwork and sheet metal adapters were supported by steel wire-ropes suspended from the framework of the ceiling. High temperature silicone, and glass-fiber gaskets were used to seal all connections. The ductwork and the overall scheme of the system can be seen in Figures 3.1 – 3.3.

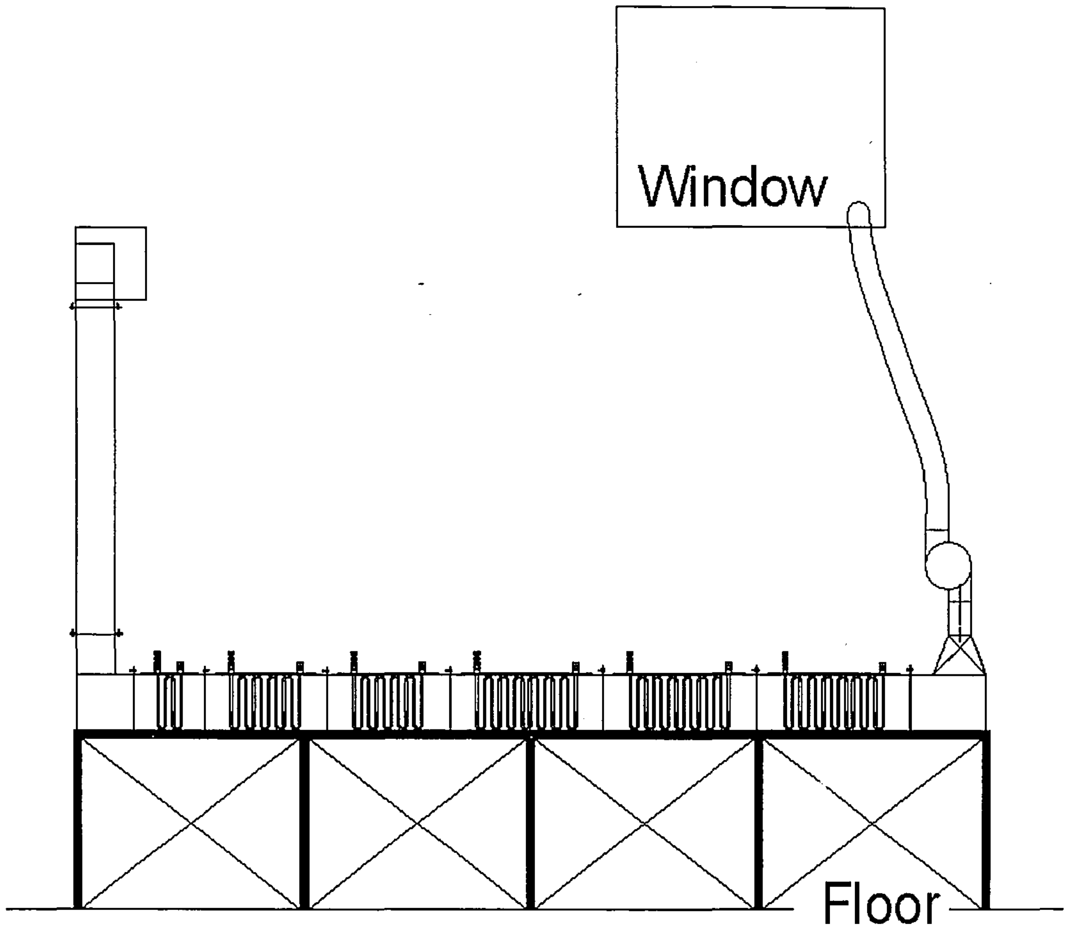

Figure 3.1. Front view of heat exchanger and ductwork

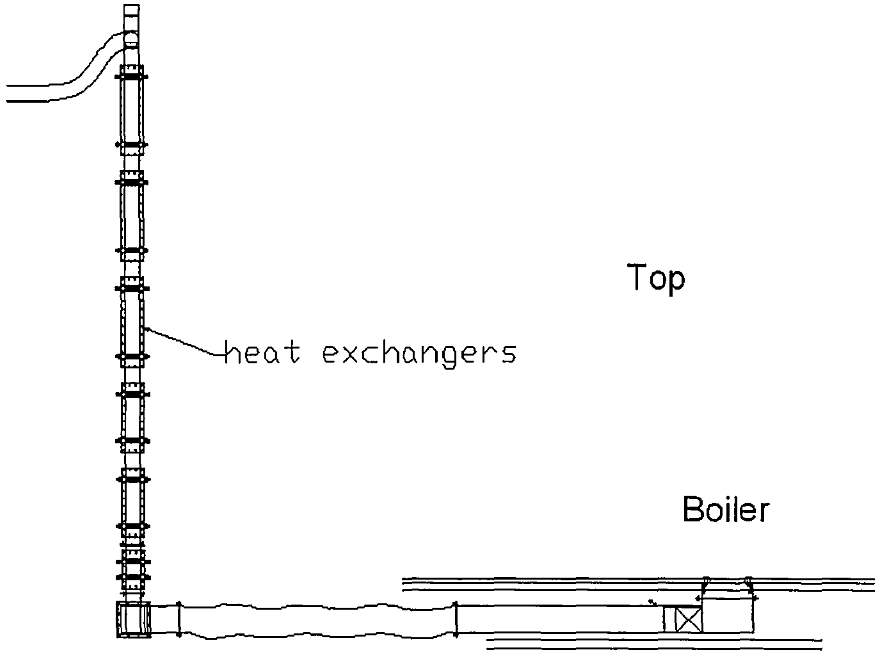

Figure 3.2. Top view of heat exchangers and ductwork

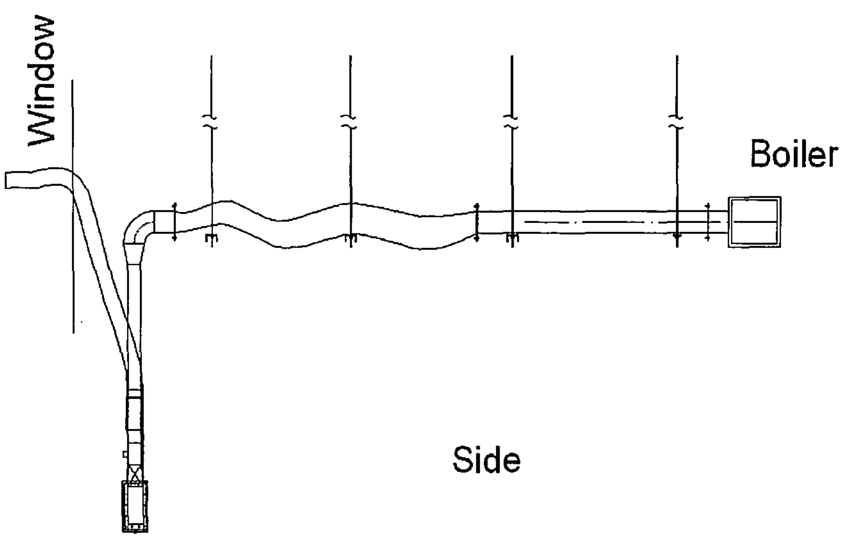

Figure 3.3. Side view of heat exchangers and ductwork

3.5 Heat Exchanger Design

The heat exchanger design utilized banks of aligned, vertical cylinders in cross-flow. Vertical cylinders were chosen in order to control the condensation occurring on the tubes.

It was necessary that the heat exchangers could be easily taken apart to observe and clean any acid buildup on the tubes. For this reason, the heat exchangers were modeled after the standard "shell-and-tube" heat exchanger.

As mentioned in the introduction, a standard shell and tube heat exchanger could not be used because of the high cooling water requirement. A typical shell-and-tube has many straight tubes in a cylindrical shell, and the water flows in a straight path through the heat exchanger. The flue gas makes several u-turns flowing over baffles, in order to enhance heat transfer.

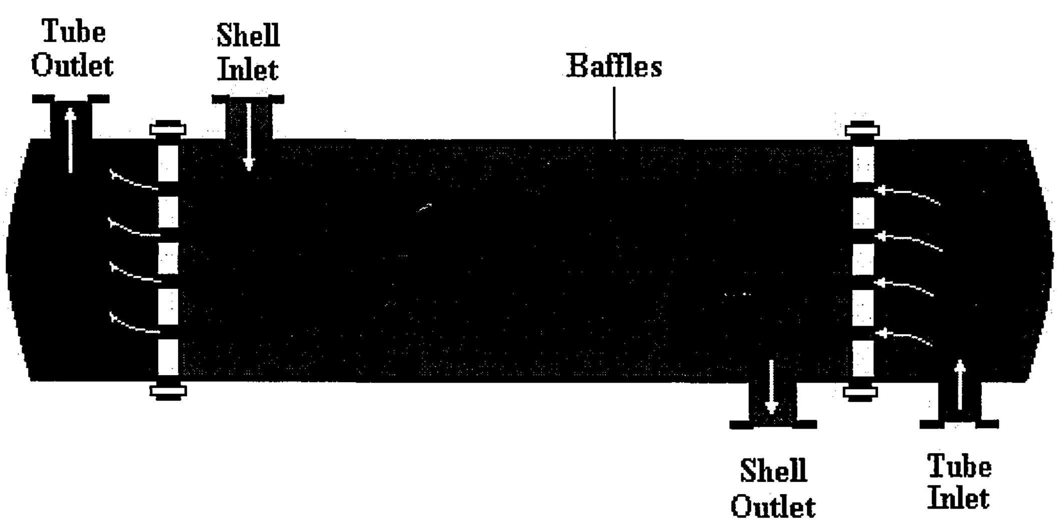

Figure 3.4. Shell-and-tube diagram [Courtesy Washington University – St. Louis]

3.5 Heat Exchanger Design

The heat exchanger design utilized banks of aligned, vertical cylinders in cross-flow. Vertical cylinders were chosen in order to control the condensation occurring on the tubes.

It was necessary that the heat exchangers could be easily taken apart to observe and clean any acid buildup on the tubes. For this reason, the heat exchangers were modeled after the standard "shell-and-tube" heat exchanger.

As mentioned in the introduction, a standard shell and tube heat exchanger could not be used because of the high cooling water requirement. A typical shell-and-tube has many straight tubes in a cylindrical shell, and the water flows in a straight path through the heat exchanger. The flue gas makes several u-turns flowing over baffles, in order to enhance heat transfer.

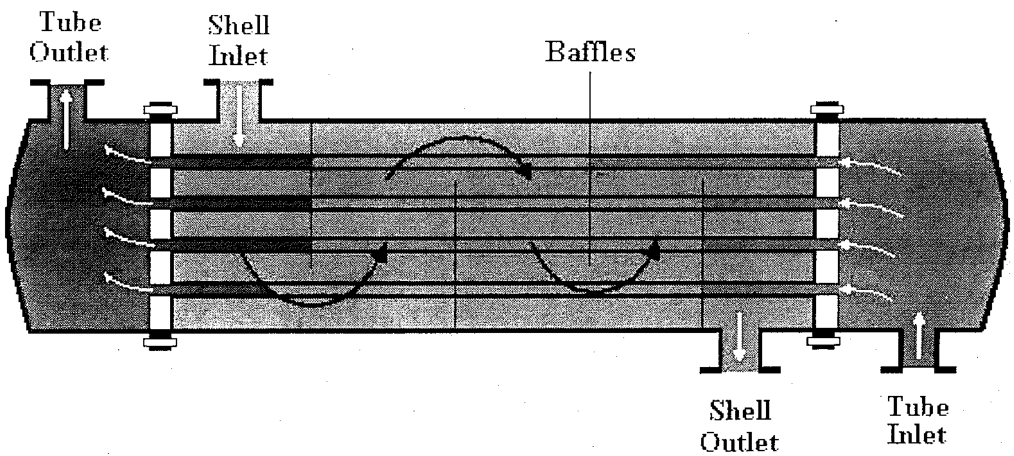

Figure 3.4. Shell-and-tube diagram [Courtesy Washington University – St. Louis]

In order to maintain this effect, the heat exchanger designed in this project simply switches the paths of the two fluids. The new design uses bent tubing in a straight duct, so that the flue gas will flow in a straight path, but the water will make several u-turns.

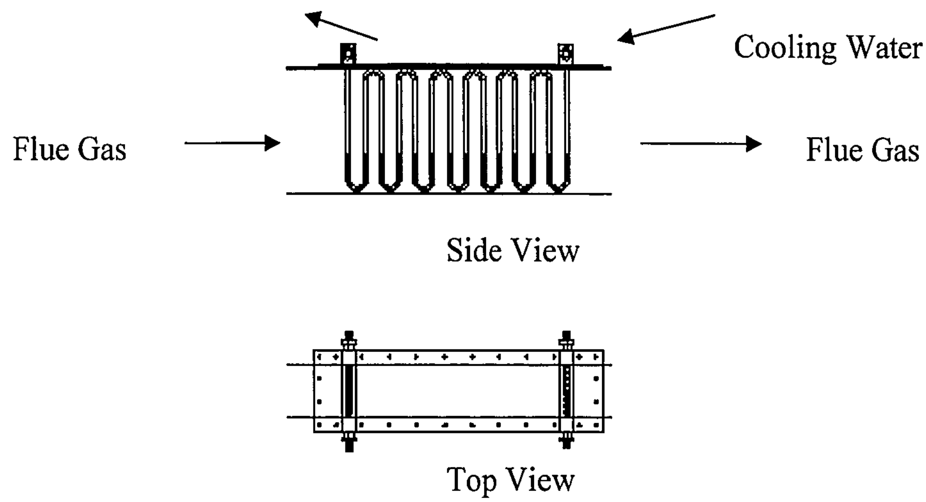

Figure 3.5. New heat exchanger design

Figure 3.5 shows the basic design. This arrangement allows for a similar surface area as compared to the traditional shell-and-tube, but uses fewer tubes.

The tube bundles are welded to the top plate and a cap is welded on the outside of the plate to act as manifolds for the inlet and outlet cooling water.

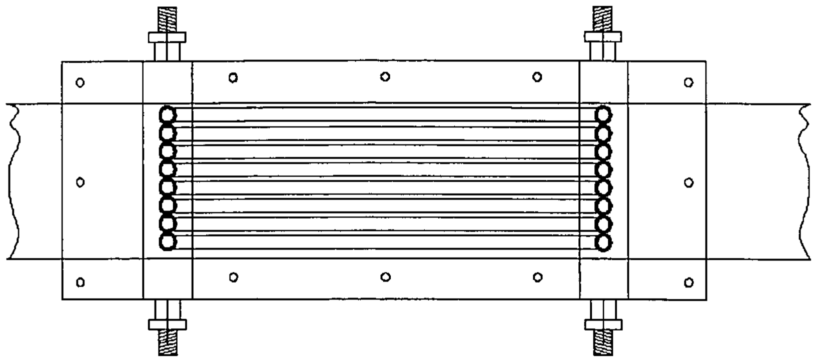

Figure 3.6. Top cover and manifold

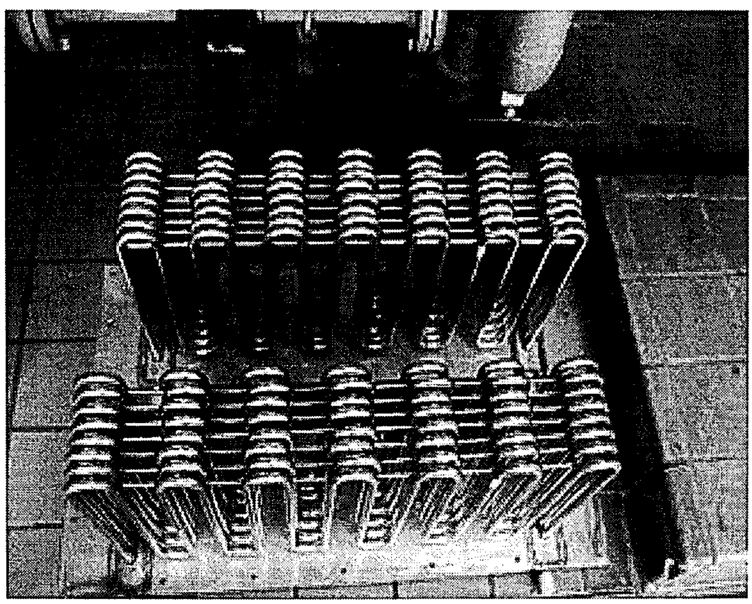

Figure 3.7. Tubing bundles

The tube bundle and top cover units are dropped down into a trough-like, rectangular shell. The lower wall of each shell is slightly tapered to allow the condensate to drain freely. The shells are also equipped with various porthole fittings to allow

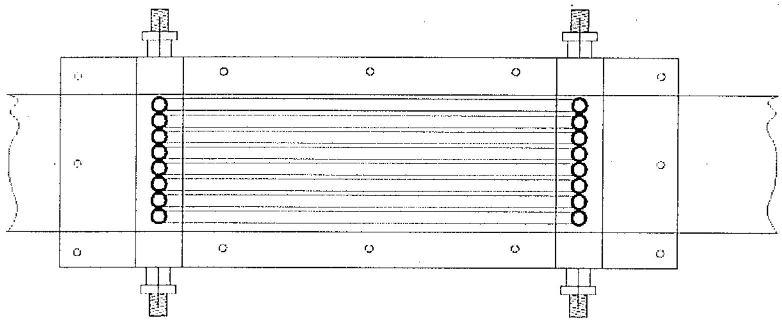

Figure 3.6. Top cover and manifold

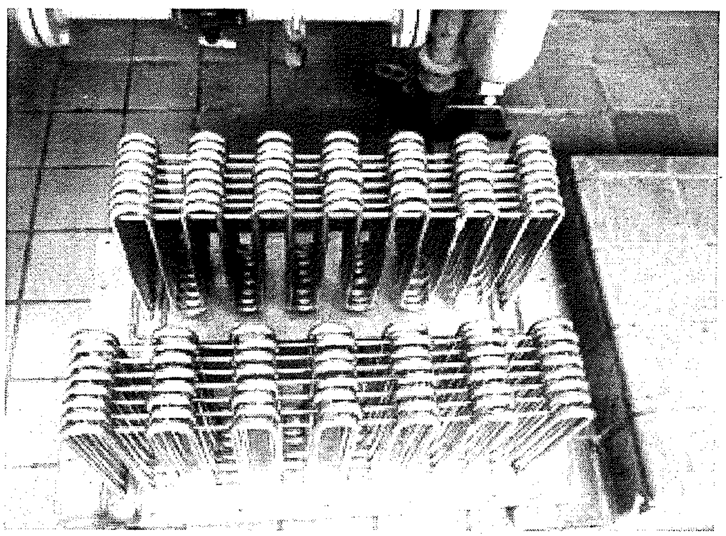

Figure 3.7. Tubing bundles

The tube bundle and top cover units are dropped down into a trough-like, rectangular shell. The lower wall of each shell is slightly tapered to allow the condensate to drain freely. The shells are also equipped with various porthole fittings to allow thermocouples and stack probes to be inserted during operation. Figure 3.8 shows the details of the shell.

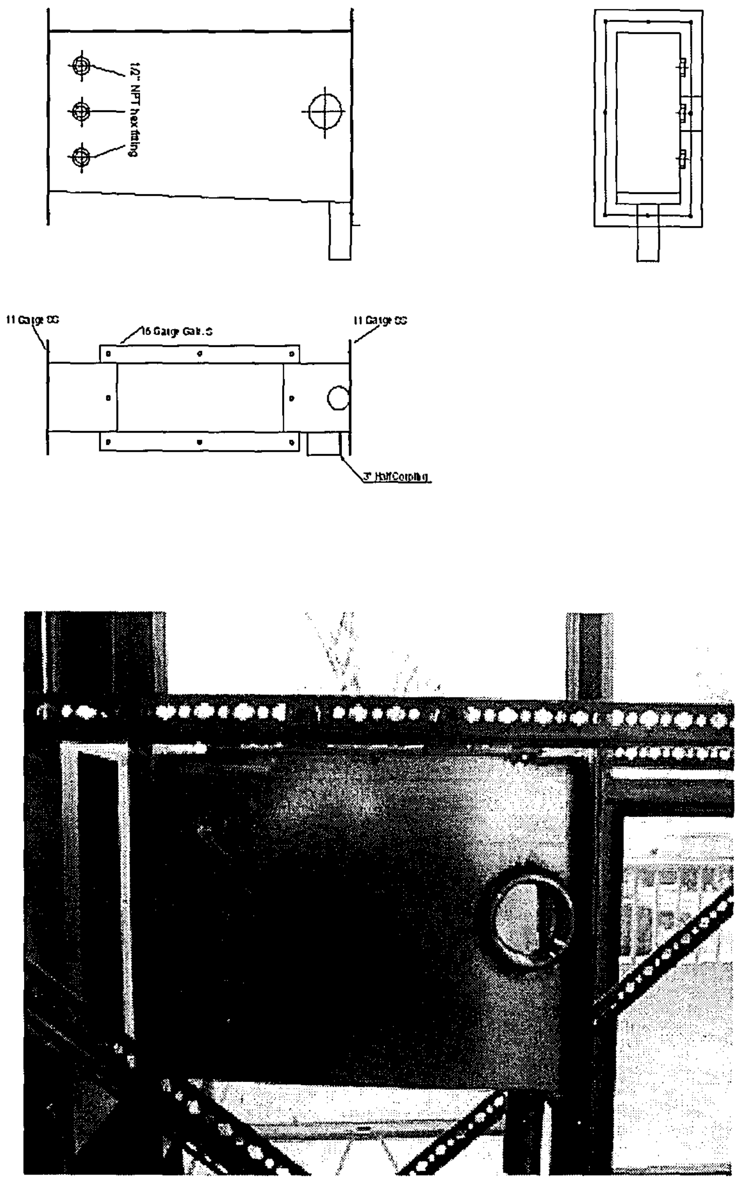

Figure 3.8. Heat exchanger shell

The original design for the heat exchanger called for three stages: the acid stage, the buffer stage, and the condensing stage. After completing the calculations for the thermocouples and stack probes to be inserted during operation. Figure 3.8 shows the details of the shell.

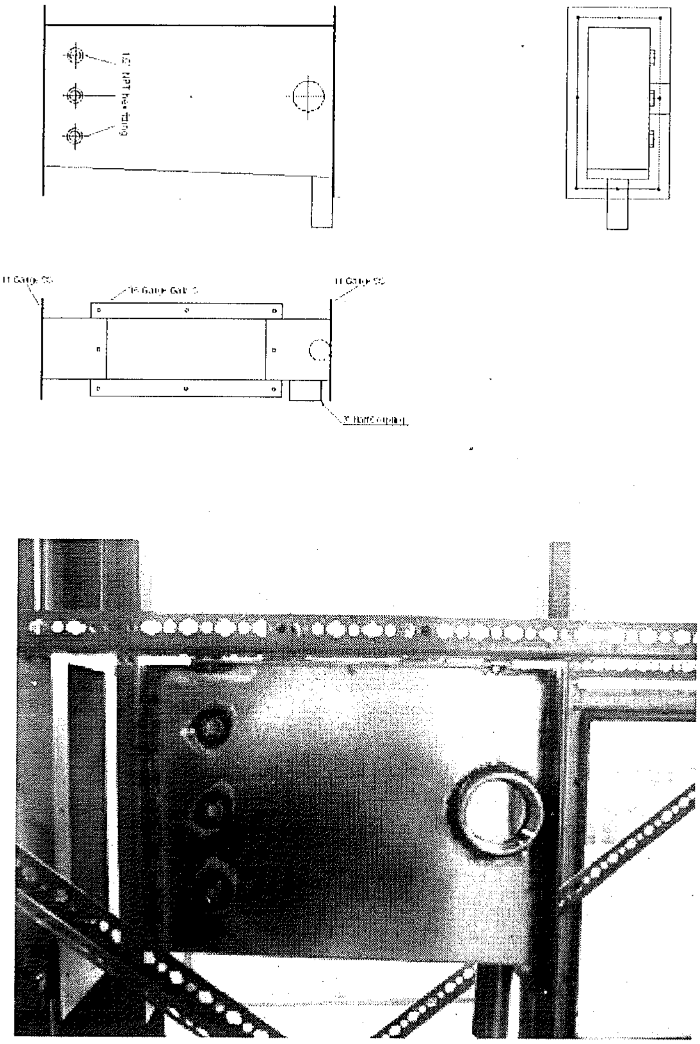

Figure 3.8. Heat exchanger shell

The original design for the heat exchanger called for three stages: the acid stage, the buffer stage, and the condensing stage. After completing the calculations for the design, it was found that it would be best to break the heat exchanger into six parts. The first part is a preliminary section, used only if the inlet flue gas was above the desired temperature. The second part is the "acid stage", third part is the "buffer stage", and parts 4-6 comprise the "condensing stage". The heat transfer performance design specifications can be found in Table 3.2. "Transverse" refers to the direction perpendicular to the flow of flue gas. "Longitudinal" refers to the direction in line with the flue gas.

Heat Exchanger Performance Design Specifications

	Units	HX 1	HX 2	HX 3	HX 4	HX 5	HX 6	total
temperature of flue gas in	deg F	450	310	205	146	103.2	97.1	
temperature of flue gas out	deg F	310	205	146	103.2	97.1	89.2	
mass flow rate of flue gas	lb/hr	300	300	300	300	300	300	
temperature of cooling water in	deg F	125	125	125	65	65	65	
temperature of cooling water out	deg F	136.2	133.4	129.7	74.9	70.2	68.9	
mass flow rate of cooling water	lb/hr	900	900	900	900	900	900	
transverse tube spacing pitch	in	0.722	0.722	0.722	0.722	0.722	0.722	
longitudinal tube spacing pitch	in	2	2	2	2	2	2	
width of flue gas duct	in	14	14	14	14	14	14	
height of flue gas duct	in	6	6	6	6	6	6	
number of rows in the longitudinal direction		4	6	10	14	14	14	
number of rows in the transverse direction		8	8	8	8	8	8	
length of heat exchanger section	ft	0.708	1.042	1.708	2.375	2.375	2.375	10.6
outside diameter of cooling tube	in	0.5	0.5	0.5	0.5	0.5	0.5	
inside diameter of cooling tube	in	0.37	0.37	0.43	0.43	0.43	0.43	
cooling tube material		AL6XN	AL6XN	AISI 316	AISI 316	AISI 316	AISI 316	
heat transfer surface area	ft^2	5.0	7.5	12.5	17.5	17.5	17.5	42.6
total length of tubing (heat transfer)	ft	38.1	57.3	95.7	134.1	134.1	134.1	593
velocity of flue gas	ft/s	3.3	2.8	2.4	2.2	2.0	2.0	
maximum velocity of flue gas	ft/s	9.8	8.3	7.2	6.5	6.0	6.0	
Reynolds # for hot fluid		1126	1252	1333	1379	1381	1398	
Nusselt # for hot fluid		17.7	20.0	21.7	22.4	22.4	22.6	
convection coefficient for hot fluid	BTU/(hr-ft^2-deg F)	8.9	8.9	8.8	8.6	8.3	8.3	
velocity of cooling water	ft/s	0.7	0.7	0.5	0.5	0.5	0.5	
Reynolds # for cold fluid		3784	3736	3161	1692	1637	1622	
Nusselt # for cold fluid		22.8	22.6	18.7	17.6	17.6	17.6	
convection coefficient for cold fluid	BTU/(hr-ft^2-deg F)	277	274	195	171	170	170	
thermal resistance of tube wall	deg F-hr/BTU	1.85E-04	1.23E-04	3.24E-05	2.31E-05	2.31E-05	2.31E-05	
heat transfer	BTU/hr	10080	7560	4248	8891	4670	3522	
overall heat transfer coefficient	BTU/(hr-ft^2-deg F)	8.4	8.4	8.3	8.1	7.8	7.8	
UA supplied by arrangement	BTU/(hr-deg F)	41.8	62.6	102.0	174.9	144.9	136.1	662
average surface temp of metal	deg F	140.9	134.1	129.2	69.7	67.2	66.8	
% moisture in flue gas inlet	% by volume	0.10	0.10	0.10	0.10	0.08	0.06	
% moisture in flue gas outlet	% by volume	0.10	0.10	0.10	0.07	0.06	0.05	
mass flow rate of condensate	lb/hr	0.0	0.0	0.0	5.6	4.1	2.8	12.5

Table 3.2. Heat Exchanger Performance Design Specifications

3.6 Heat Exchanger Materials

Since the heat exchanger was expected to be in a corrosive environment, material selection was a critical part of the design. The environment was expected to consist of high concentrations of sulfuric acids in Heat Exchangers 1-3, and fairly high concentrations of nitric and hydrochloric acids in Heat Exchangers 4-6 (this is because the dew point of sulfuric acid is much higher than that of water and nitric / hydrochloric acids). Cost of materials was also a consideration.

Stainless steel (SS 316) is known for having a high resistance to corrosion from nitric and hydrochloric acids [6]. For this reason, SS 316 was chosen for the shell-duct material, and also the tube material for HX 4-6. SS 316 is also a moderately priced material and widely available.

For Heat Exchangers 1-3, a more resistant alloy would be needed to protect against high concentrations of sulfuric acid, because in these stages, water will not be condensing out to rinse the tubes off. A moderately priced alloy was chosen called AL-6XN. It is a high molybdenum, high chromium, stainless steel. There were other alloys available that may protect better against sulfuric acid, but they would not be economical for this project.

4.0 Experimental Setup

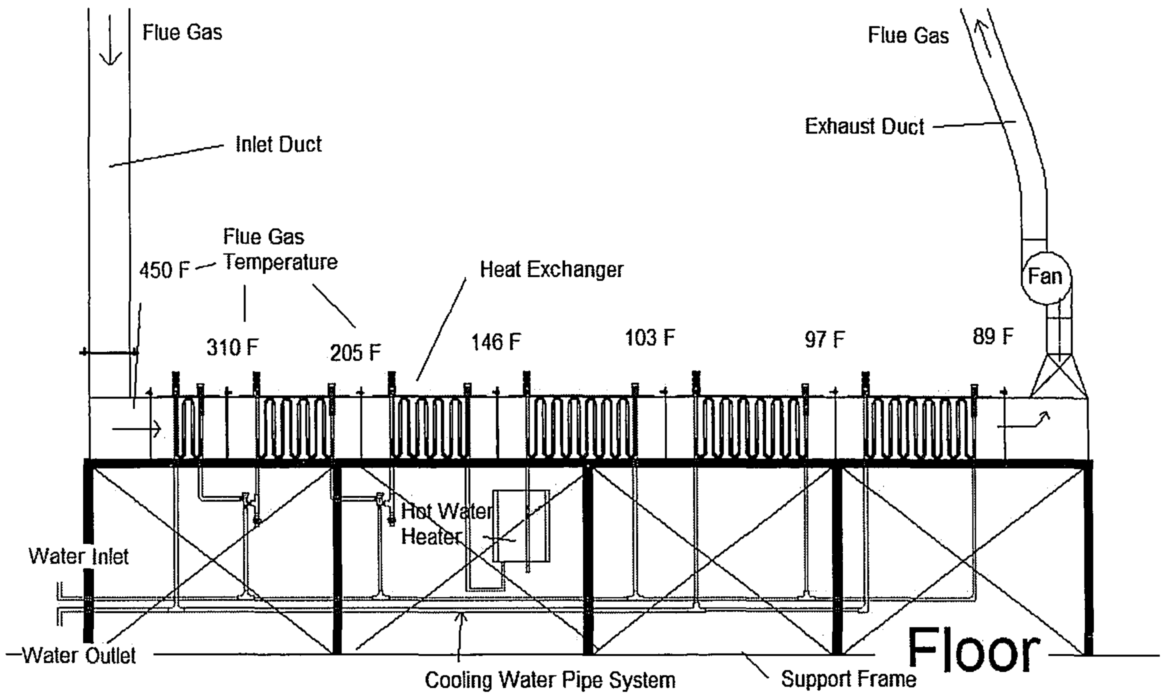

Figure 4.1. Overview of heat exchangers

As seen in Figure 4.1, the experimental setup consists of six heat exchangers, an inlet and outlet duct, an induced draft fan, cooling water lines, and a hot water heater. The reason for the hot water heater is to keep the cooling water in the first three heat exchangers above the dew point temperature for the water vapor in the flue gas. Since one of the objectives is to separate sulfuric acid from the water condensate, it is desired to have water vapor condense out only in the last three heat exchangers. If the cooling water in the tubes causes the surface temperature of any tube to drop below the dew point temperature, there will be condensation. The hot water heater is used to control the condensation. Figure 4.1 also shows the design flue gas temperatures as they progress through the heat exchanger.

Mixing valves were used in the cooling water line between Heat Exchangers 1 & 2, and between 2 & 3. These valves attempted to take the hot water from the previous heat exchanger and mixed with the cold tap water to lower the cooling water temperature to roughly 125 °F. These valves did not work very well, and only succeeded in lowering the water temperature by a few degrees. However, these valves were not essential to the overall performance of the heat exchanger.

4.1 Measurement Equipment

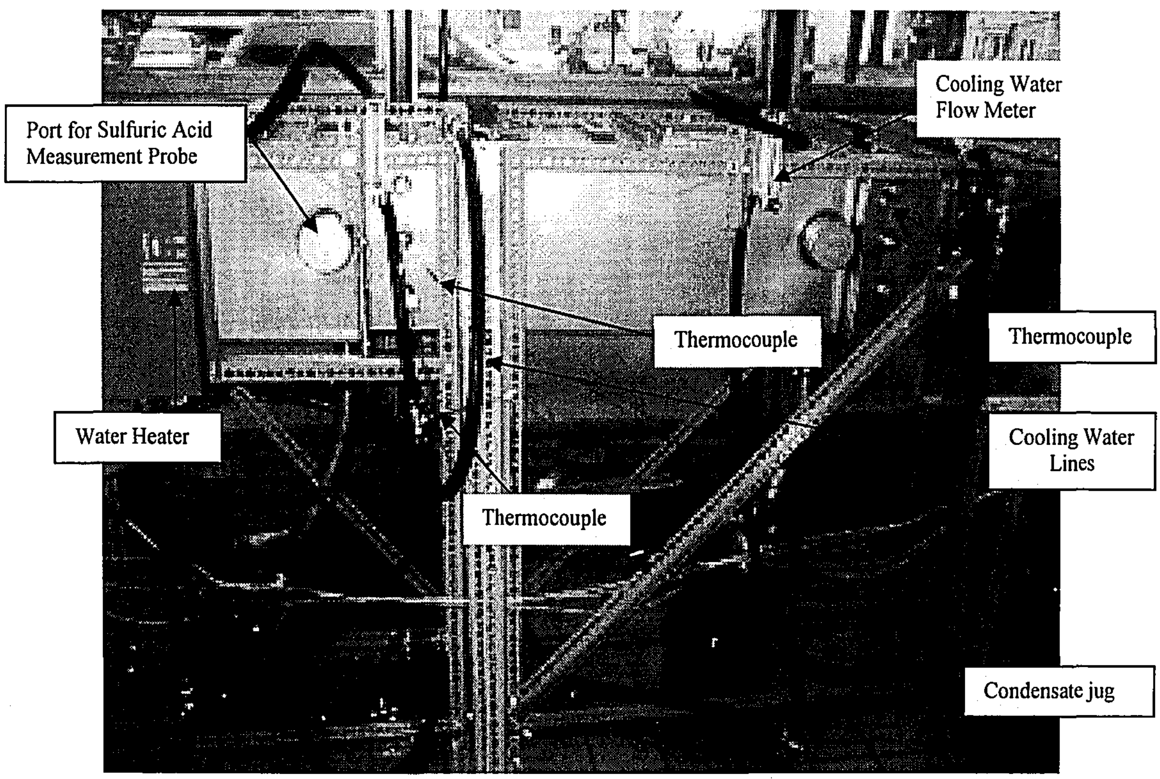

Figure 4.2. Test Equipment

K and T type thermocouples were used to measure the flue gas temperature before and after each heat exchanger. The cooling water temperature was also measured before and after each heat exchanger. Thermocouples were also attached to heat exchanger tubes to monitor the metal surface temperature (see Figure 4.3).

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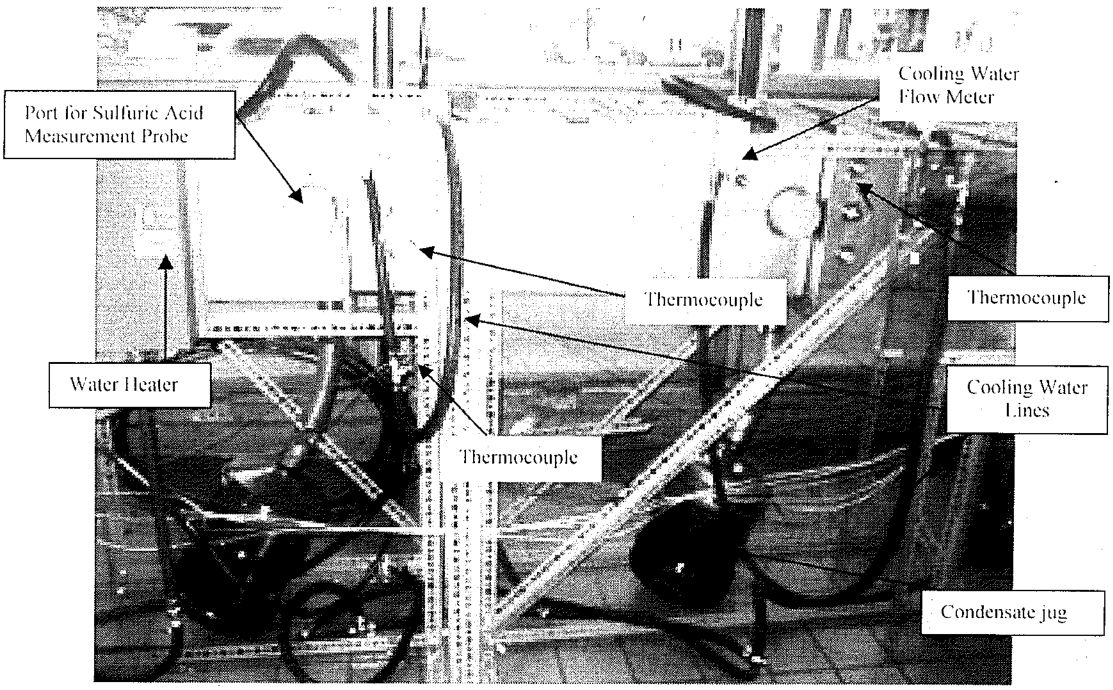

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K and T type thermocouples were used to measure the flue gas temperature before and after each heat exchanger. The cooling water temperature was also measured before and after each heat exchanger. Thermocouples were also attached to heat exchanger tubes to monitor the metal surface temperature (see Figure 4.3).

Rotameter type flow meters were used between each heat exchanger to measure the cooling water flow rate. Plastic jugs were used to collect the condensate at each heat exchanger.

A pitot tube and manometer were used in a straight section of the exhaust duct in order to measure the flue gas velocity. The S-type pitot tube was placed in the center of the circular duct. The following velocity profile was assumed:

$$\frac{V}{V_{max}} = \left(\frac{R-r}{R}\right)^{\frac{1}{6.6}} \qquad (4.1)[1]$$

where V is the velocity of the flue gas as a function of r (radius). V_{max} is the velocity at the center of the duct, and *R* is the actual radius of the duct. The flue gas velocity equals 0 at r = R. The flue gas flow rate can be determined from Equation 4.2.

$$\dot{m} = \int_0^R \rho V A dr \qquad (4.2)$$

Combining Equations 4.1 and 4.2 and evaluating the integral, the flue gas flow rate reduces to

$$\dot{m} = \frac{66}{71}\rho V_{max} A$$

Insulation was not initially installed, but after a few initial tests, it was found that insulation was necessary in order to control the condensation. The ducts and the heat exchangers were insulated as seen in Figure 4.4.

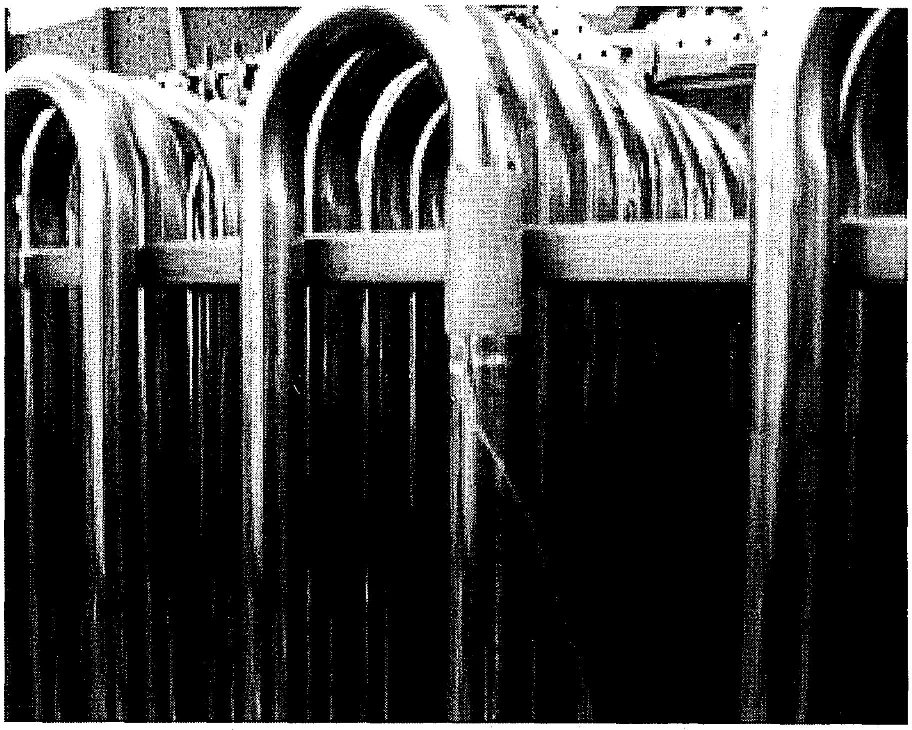

Figure 4.3. Surface thermocouple on heat exchanger tube

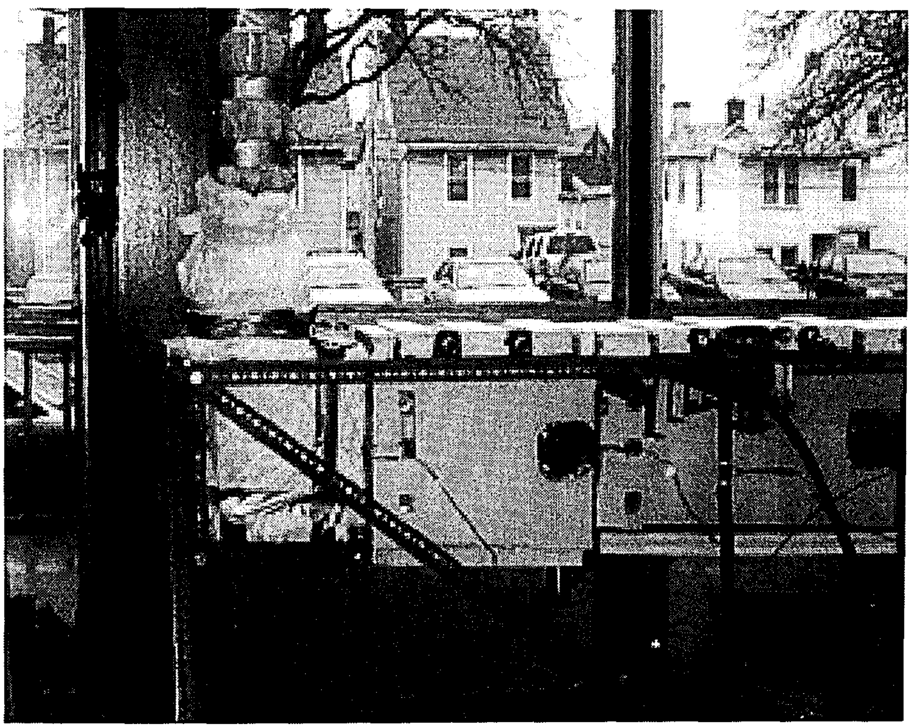

Figure 4.4. Heat exchanger insulation

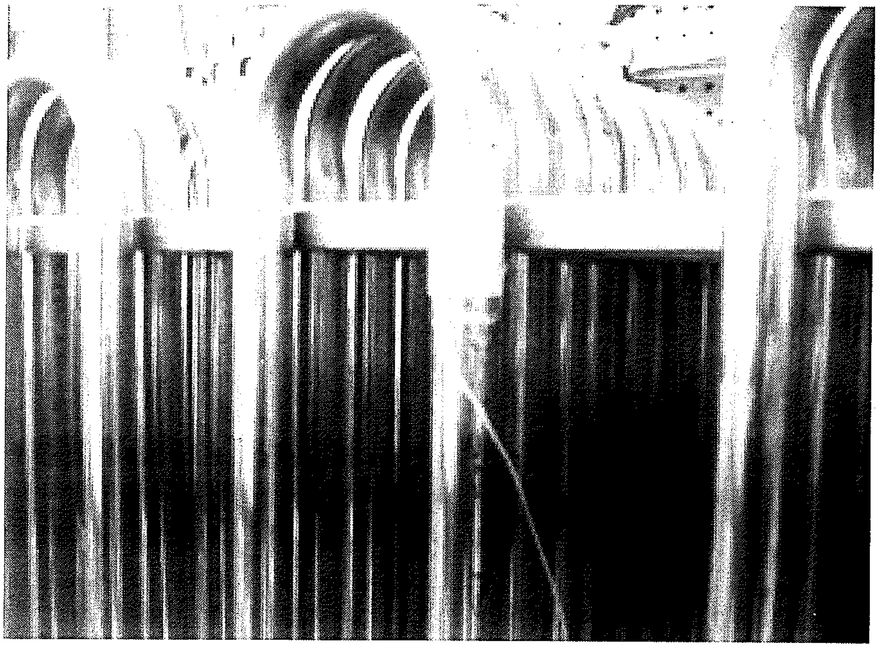

Figure 4.3. Surface thermocouple on heat exchanger tube

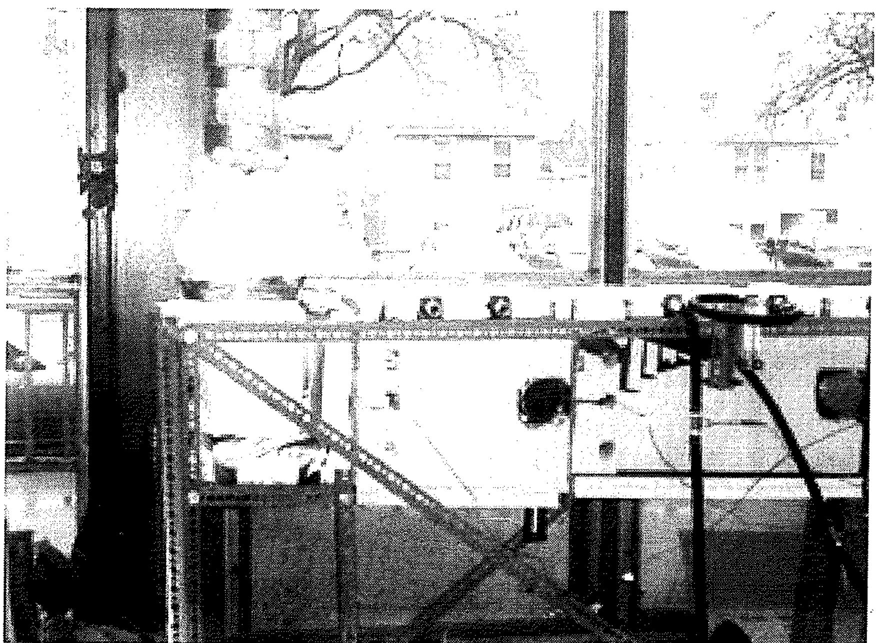

Figure 4.4. Heat exchanger insulation

5.0 Test Procedures

5.1 Data Acquisition

The heat exchanger process was evaluated on a total run, time average basis. This means that the performance of the heat exchangers was determined using the average temperatures over the run, once the system reached steady state.

Thermocouples were continuously monitored using a multiple channel thermocouple data acquisition system (DAS) which was connected to a PC. Certain thermocouples were not connected to the PC DAS.

Cooling water flow meters and the pitot-tube manometer were read manually every ten minutes.

The condensate jugs were emptied after the system reached steady state. After the run, the total volume of condensate was measured manually. The time was kept so that the condensate flow rates could be determined.

The following chart and diagram show which measurements were made and the frequency.

Data Acquisition

ID	Device	Location	Measurement Frequency
TC-1	Thermocouple, Type K	Flue Gas, Economizer outlet	10 min.
TC-2	Thermocouple, Type K	Flue Gas, before HX1	5 sec.
TC-3	Thermocouple, Type K	Flue Gas, between HX 1 & HX 2	5 sec.
TC-4	Thermocouple, Type K	Flue Gas, between HX 2 & HX 3	5 sec.
TC-5	Thermocouple, Type K	Flue Gas, between HX 3 & HX 4	5 sec.
TC-6	Thermocouple, Type K	Flue Gas, between HX 4 & HX 5	5 sec.
TC-7	Thermocouple, Type K	Flue Gas, between HX 5 & HX 6	5 sec.
TC-8	Thermocouple, Type K	Flue Gas, after HX 6	5 sec.
TC-9	Thermocouple, Type K	Flue Gas, Wet Bulb, after HX 6	10 min.
TC-10	Thermocouple, Type K	Flue Gas, before Pitot Tube	10 min.
TC-11	Thermocouple, Type T	Cooling Water, Inlet	10 min.
TC-12	Thermocouple, Type T	Cooling Water, between HX 5 & HX 6	5 sec.
TC-13	Thermocouple, Type T	Cooling Water, between HX 4 & HX 5	5 sec.
TC-14	Thermocouple, Type T	Cooling Water, between Hot Water Heater & HX 4	5 sec.
TC-15	Thermocouple, Type T	Cooling Water, between HX 3 & Hot Water heater	5 sec.
TC-16	Thermocouple, Type T	Cooling Water, between Water Mixing Valve & HX 3	5 sec.
TC-17	Thermocouple, Type T	Cooling Water, between HX 2 & Water Mixing Valve	5 sec.
TC-18	Thermocouple, Type T	Cooling Water, Outlet	5 sec.
TC-19	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 2 front	5 sec.
TC-20	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 2 rear	5 sec.
TC-21	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 3 front	5 sec.
TC-22	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 3 rear	5 sec.
TC-23	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 4 front	5 sec.
TC-24	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 4 rear	5 sec.
TC-25	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 5 front	5 sec.
TC-26	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 5 rear	5 sec.
TC-27	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 6 front	5 sec.
TC-28	Thermocouple, Type K	Heat Exchanger Tube Surface, HX 6 rear	5 sec.
RM-2	Rotameter	Cooling Water, HX 2 inlet	10 min.
RM-3	Rotameter	Cooling Water, HX 3 inlet	10 min.
RM-4	Rotameter	Cooling Water, HX 4 inlet	10 min.
RM-5	Rotameter	Cooling Water, HX 5 inlet	10 min.
RM-6	Rotameter	Cooling Water, HX 6 inlet	10 min.
PT	Pitot Tube, Manometer	After HX 6	10 min.
CJ-3	Condensate Jug	HX 3	when full, or once per run
CJ-4	Condensate Jug	HX 4	when full, or once per run
CJ-5	Condensate Jug	HX 5	when full, or once per run
CJ-6	Condensate Jug	HX 6	when full, or once per run

Table 5.1. Data Acquisition

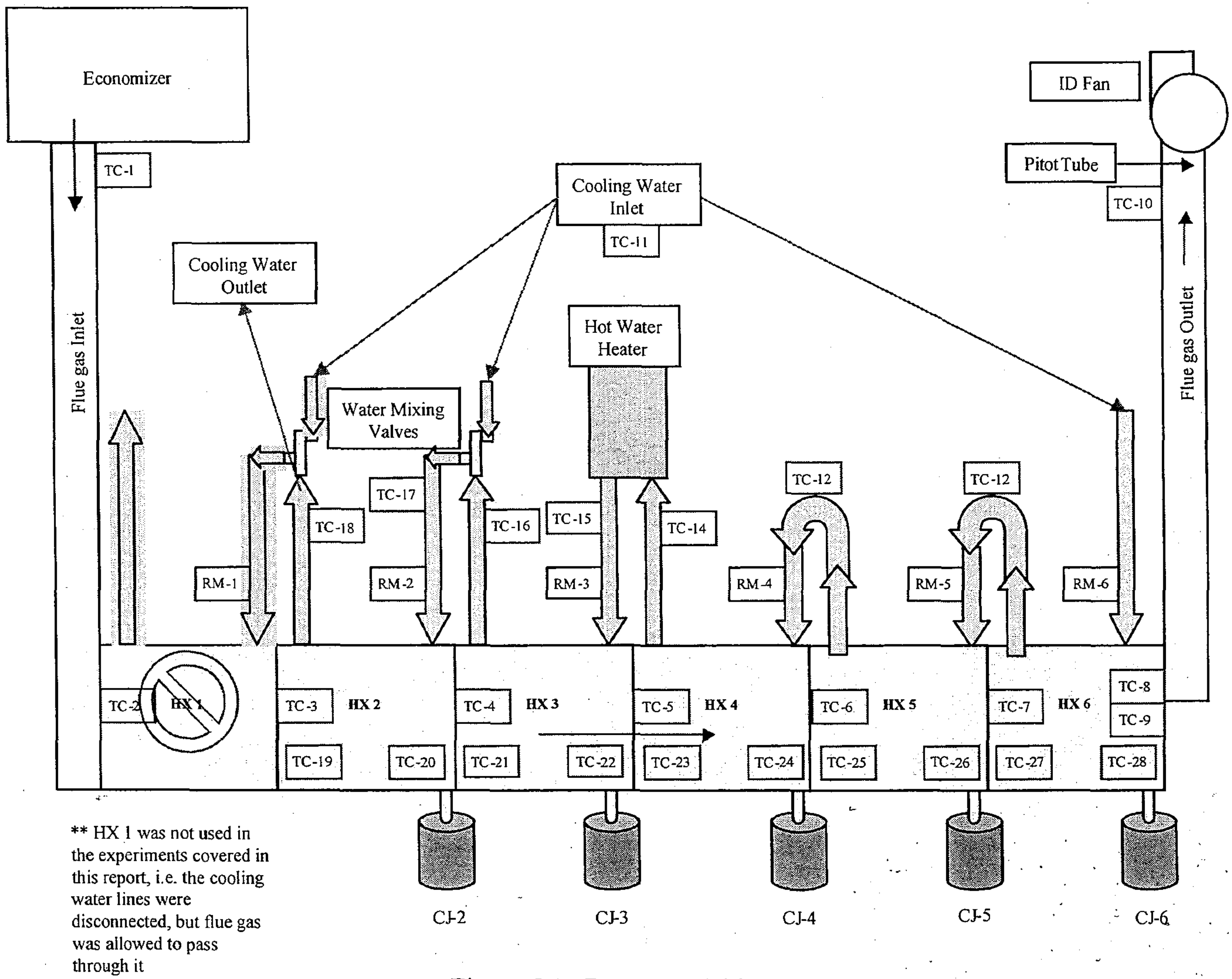

Figure 5.1. Data Acquisition

5.2 Sulfuric Acid Measurement

In order to measure the sulfuric acid content of the flue gas, equipment rented from CleanAir Engineering was used. It is a modified ASTM Method D3226-73T (Controlled Condensation, also referred to as EPA Method 8B) for sulfuric acid mist (see Figure 5.2). This method was developed especially for the determination of sulfuric acid emissions from combustion sources. The technique utilizes a glass lined probe heated to 600 °F which pulls the flue gas sample through at a constant flow rate. The flue gas then passes through a glass fiber filter maintained at a temperature of 550 °F, then through an impaction type condenser for the collection of sulfuric acid vapor and/or mist. The condenser is then rinsed with water and the rinse is tested for sulfuric acid concentration. The samples were sent to a lab for ion chromatography analysis. The samples were tested for the SO_4^{-2} ion [2].

Only one location could be sampled at a time with the equipment available for testing, however it was desired that the flue gas sulfuric acid concentration be known at each point between each heat exchanger. The heat exchangers were designed with a 3 inch diameter port located between each section. Because only one probe was available, the probe was moved from port to port, down the line, from one heat exchanger to the next. Sampling time took about 30 minutes for each run. Six port locations were sampled, encompassing five heat exchanger sections (the first heat exchanger section was not used because flue gas temperature was below 350 °F).

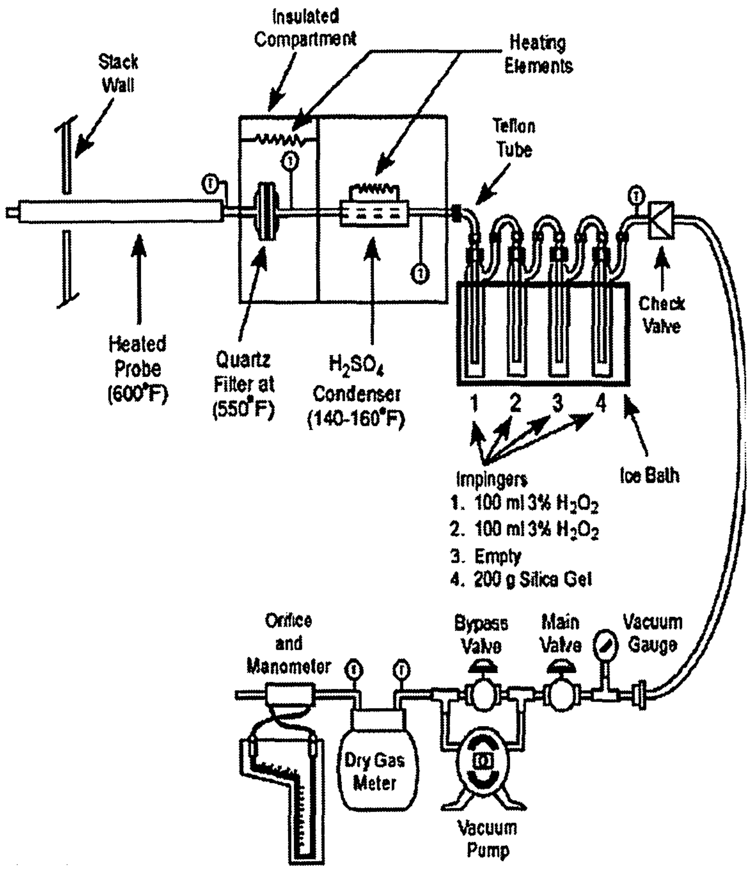

Figure 5.2. Sulfuric acid sampling train

This method also allowed the measurement of water vapor during the test using the condensing impingers. This value was compared with moisture calculated by other means. Wet and dry bulb temperatures were measured after the last heat exchanger.

5.3 Short-Term Test Plan

The short-term test plan was carried out in six experiments. Among the six experiments, the variable parameter was the flue gas flow rate. This was set by adjusting the damper gate before the heat exchangers. The pressure drop over the pitot tube was monitored to set the flue gas flow rate. Other parameters naturally changed between tests. The inlet flue gas temperature from test to test fluctuated as much as 100 °F. The inlet moisture content varied as temperature varied. The ambient temperature, as well as inlet cooling water temperature varied. Because of these factors, actual flue gas flow rate was very difficult to control. In the end flue gas flow rate varied from 300 lb/hr to 450 lb/hr.

Each test took about eight hours to run. Because the tests spanned such a long period of time, there was some variation in the loading of the boiler. The flue gas inlet temperature seemed to be a fairly good indicator of this variation.

5.4 Long-Term Test Plan

The long-term test plan includes the use of finned heat exchangers and installation at a coal-fired power plant. The long-term testing is not within the scope of this report.

6.0 Results

Table 6.1 shows the tests carried out and the specific parameters observed in each test.

Test Date	Flue Gas Flow Rate (lb/hr, wet)	EPA Method 8B	*Condensate Lab Analysis*		
			SO_4^{--}	NO_3^-	Cl^-
01/08/07	300	Port 1 only	none	none	none
01/10/07	300	All 6 ports	none	none	none
01/12/07	370	All 6 ports	none	none	none
01/16/07	420	All 6 ports	HX 1-6	HX 3-6	HX 3-6
01/18/07	380	All 6 ports	none	none	none
01/19/07	350	All 6 ports	none	none	none

Table 6.1. Completed Tests

The Condensate Lab Analysis refers to ion chromatography tests done on the condensate collected from the heat exchangers. This analysis was performed by Benchmark Analytics, Center Valley, PA. EPA Method 300.0 was used.

6.1 Temperatures

The average temperatures of the system for one run can be seen in Figure 6.1 below. These are the average temperatures over the duration of the test on 01-10-2007.

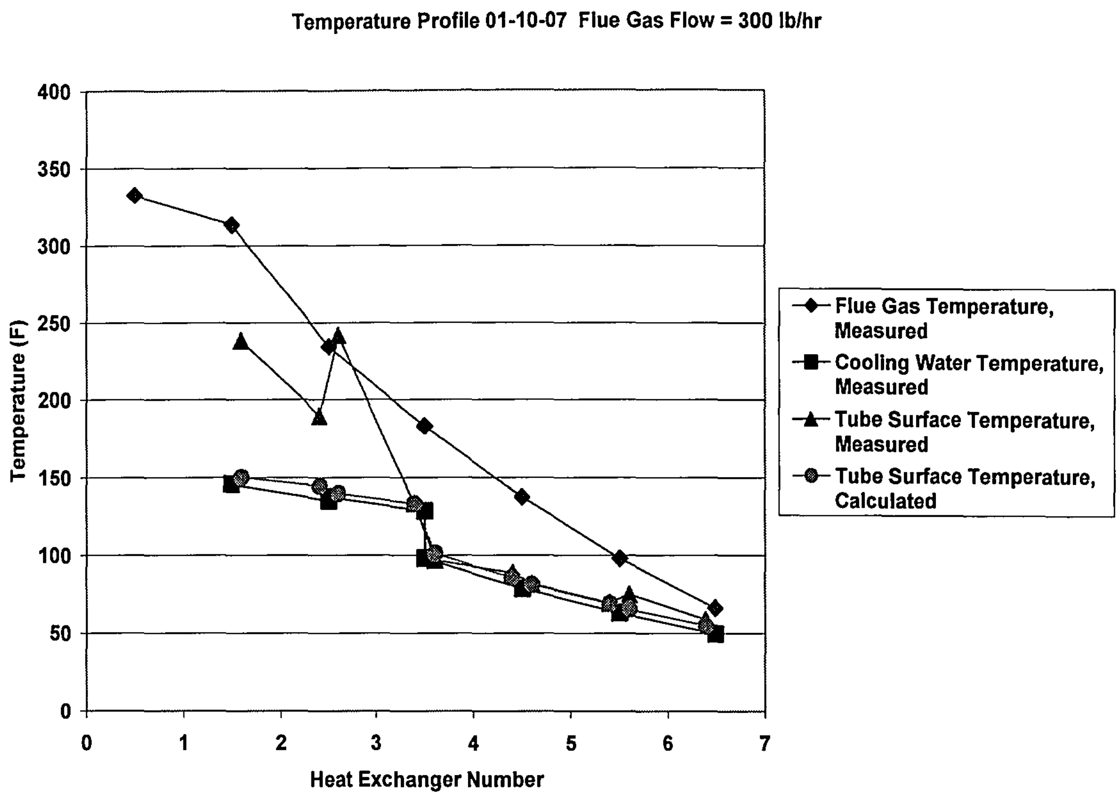

Figure 6.1. Temperature progression through heat exchanger, 01/10/2007

Figure 6.1 shows the flue gas temperature, the cooling water temperature, and the heat exchanger tube wall surface temperature. The reason there are two values for tube wall surface temperature is because these measurements were very difficult to obtain using thermocouples due to the high temperatures and corrosive environment. Paste-on thermocouples were used for tube wall surface temperature, and a layer of high temperature epoxy was applied over top of the thermocouple patch. Nonetheless, the thermocouples seemed to detach from the tube, especially in the high temperature zones. This is why Figure 6.1 shows measured tube wall surface temperatures that are erratic. The indicated temperatures are actually closer to the flue gas temperature than the expected tube wall surface temperature. This is simply because the thermocouple wire was severed at the point of connection to the patch, due to corrosion. The "calculated tube wall temperature" in Figure 6.1 is back-calculated by using the measured cooling water temperature at that point, utilizing the theoretical tube wall conduction resistance.

Also notice the increase in cooling water temperature (from right to left) due to the hot water heater between heat exchangers 3 and 4. Only a very small temperature drop was observed over the mixing water valves. Even though these valves did not work as planned, they did not negatively affect the heat exchanger performance. Temperature profiles for all the runs can be found in Appendix A.

6.2 Flue Gas Moisture Content

Three different methods were used to measure flue gas moisture content. First, during the controlled condensation method for sulfuric acid measurement, four impingers were used to collect the moisture in the sample, as a standard procedure inherent to the method. This provided a manual measurement of the moisture content.

Second, a wet bulb temperature was measured at the exit of the heat exchanger system. This measurement, coupled with the dry bulb measurement at the same location, allows for the determination of the flue gas moisture content. This measurement was only made at one point.

The third method used to measure flue gas moisture content assumed the water in the flue gas exiting Heat Exchanger 6 was saturated (see section 2.2.1). The flue gas moisture content was calculated using the flue gas temperature after Heat Exchanger 6. The rates of condensation from each individual heat exchanger were measured. In order to calculate the flue gas moisture content in Heat Exchangers 1 through 5, the condensation rates were added back into the total vapor content of the flue gas. For example,

$$\dot{m}_{vapor,in,6} = \dot{m}_{vapor,out,6} + \dot{m}_{condensate,6} \tag{6.1}$$

where, the mass flow rate of vapor in the flue gas coming into Heat Exchanger 6 equals the mass flow rate of vapor going out of Heat Exchanger 6 plus the mass flow rate of condensate produced by Heat Exchanger 6. The mass flow rate of vapor entering Heat Exchanger 6 equals the mass flow rate of vapor exiting Heat Exchanger 5, and so forth.

Figure 6.2 shows results from the three measurement methods plotted as the gas progresses through the heat exchanger.

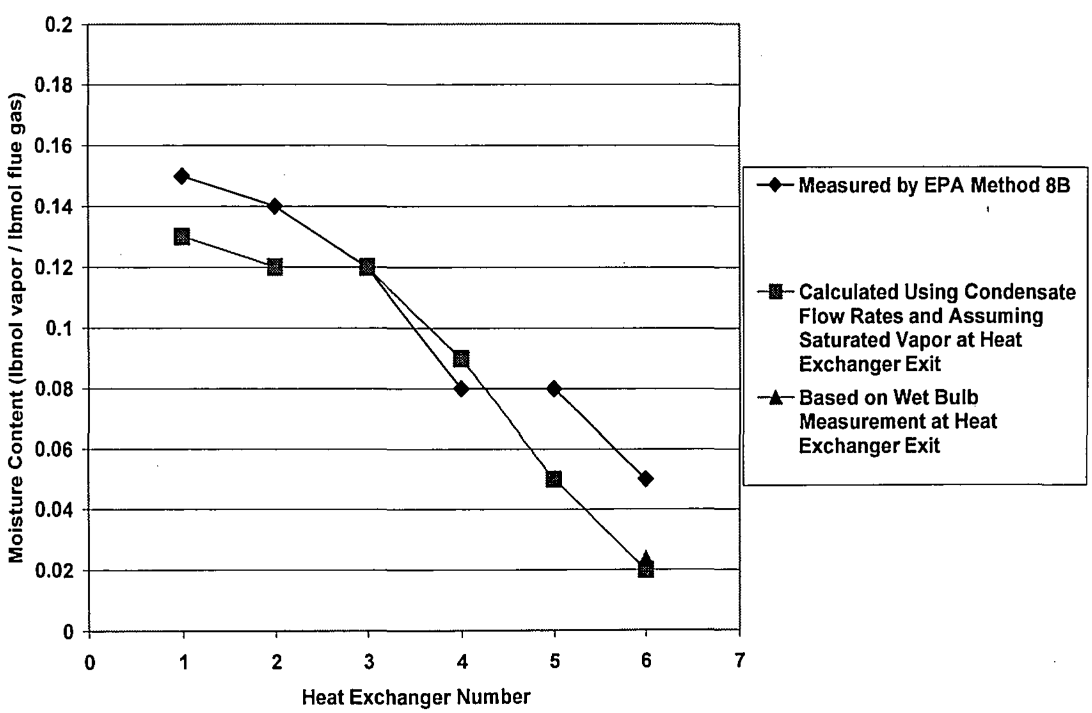

Figure 6.2. Moisture Content for Test on 01-18-2007, Flue Gas Flow = 380 lb/hr

For the most part, the three methods are in agreement as far as the general amount of moisture. Looking at the same charts for other tests (in Appendix B), it seems that the measured values are somewhat inconsistent. The wet bulb measurement is very consistent with the calculated method using the saturated vapor assumption. Measuring the weight of the water in the impingers proved to be difficult; therefore it should be considered the least accurate form of moisture content measurement. It does, however serve as a good approximation.

Figure 6.2 indicates that the inlet flue gas moisture content was roughly 14%. After the flue gas passed through the heat exchangers, the moisture was reduced to about 4%. Over all six tests, these numbers were fairly consistent, with flue gas inlet moisture contents ranging from 11% to 14%, and outlet moisture contents ranging from 2% to 5%.

6.3 Flow Rates

6.3.1 Water Flow Rates

There was some discrepancy in the measurement of the water flow rates through the heat exchanger tubes. Heat Exchangers 3 – 6 were in series, such that the water from Heat Exchanger 6 flowed directly into Heat Exchanger 5, and so forth. Each heat exchanger had its own water flow meter. Even though it was known that the same amount of water was flowing through each flow meter, they read slightly different flow rates. This represents the error in the flow meters. Also, the water temperature changed as it flowed through the system, causing density changes which affects the reading on the flow meters (scaled in gpm). The flow rates were corrected for temperature, but this still causes some variation.

Each water flow meter was manually calibrated before testing at different flow rates to ensure minimal error. Data taken from flow meters that were in series were corrected for temperature, corrected for any calibration errors, and then averaged because it was known that the mass of the water did not change through heat exchangers that were in series.

6.3.2 Flue Gas Flow Rate

Flue gas flow rate proved to be one of the most difficult measurements to make. The S-type pitot tube used to measure pressure drop became clogged up over time, causing inaccuracies in measurement. Efforts were made to clean the pitot tube before every test.

6.4 Heat Transfer Calculations

An overall heat and mass balance analysis was performed on each test run, using the equations derived in Section 2.4. The analysis accounts for energy transfer from the wet flue gas, energy leaving the system in the form of flue gas moisture condensation, and the energy transfer to the cooling water. Even though the system was insulated, there inevitably was heat lost to the surrounding environment. Efforts were made to estimate the extent of the heat loss.

Figure 6.3 shows a heat balance for one day's testing. A negative (-) value of heat transfer represents heat loss from a control volume encompassed by one whole section of the heat exchanger, and a positive (+) value of heat transfer represents heat gain to the control volume. Each column represents one heat exchanger in Figure 6.3. Each column has an equal amount of energy above and below the horizontal axis. The "Heat Loss / Gain / Error" portion of the heat balance is simply the amount of heat unaccounted for in the heat balance. If this portion appears as a positive value it means that amount of heat is attributed to heat loss. If the portion appears as a negative value it means that amount of heat is attributed to heat gain. More discussion on this will take place later in the report, but it is not rational that there is heat gain to the system because the ambient temperature was always less than the flue gas outlet temperature.

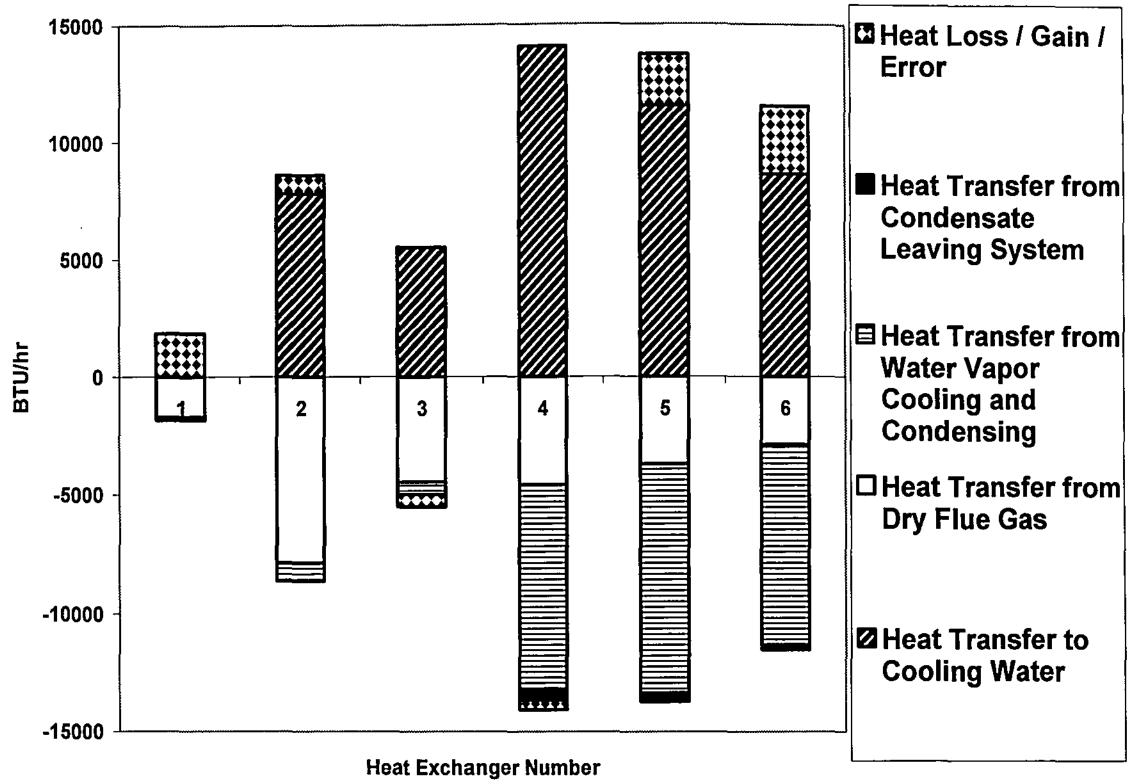

Figure 6.3. Heat Balance Chart for Test on 01-18-2007, Flue Gas Flow = 380 lb/hr

Figure 6.3 shows that the majority of the heat transfer takes place in the last three heat exchangers. Almost all of the condensation occurs here. The remaining heat balance charts can be found in Appendix C.

Table 6.2 shows the data used to make Figure 6.3. "Q water vapor" denotes the heat transfer from the flue gas vapor cooling and condensing (from Equation 2.9, the terms $\dot{m}_{v,in}h_{v,in} - \dot{m}_{v,out}h_{v,out}$). "Q condensate" represents the heat leaving the control volume in the form of flue gas condensate (from Equation 2.9, the term $\dot{m}_{cond}h_{cond}$).

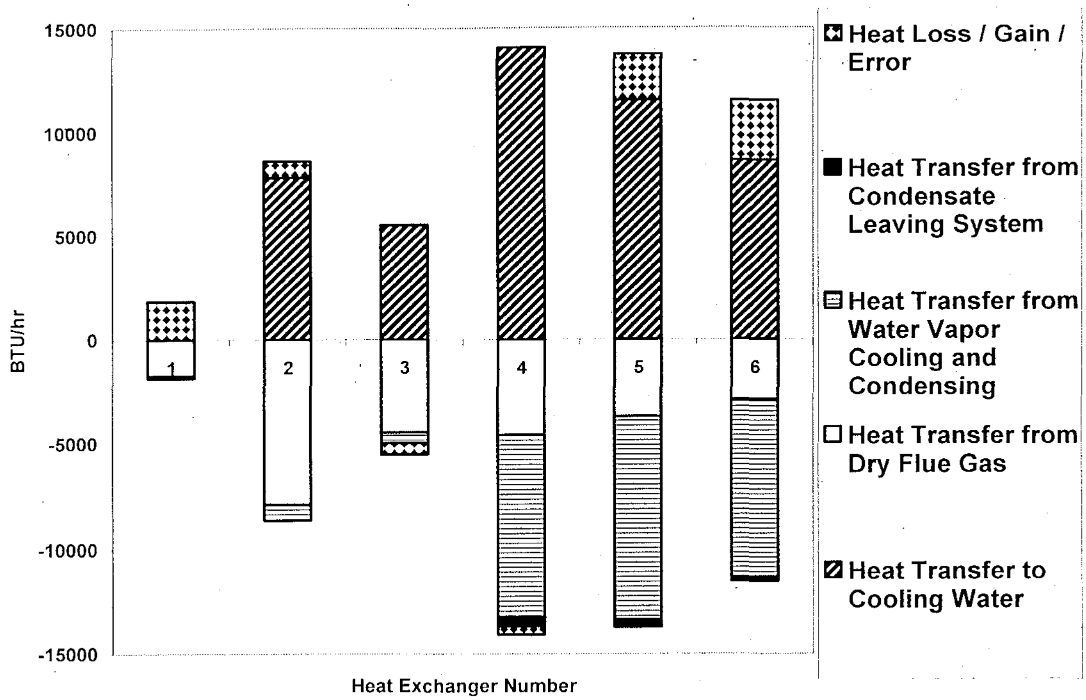

Figure 6.3. Heat Balance Chart for Test on 01-18-2007, Flue Gas Flow = 380 lb/hr

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HX		1	2	3	4	5	6
Q cooling water	BTU/hr	0	7800	5500	14100	11500	8600
Q dry flue gas	BTU/hr	-1700	-7900	-4500	-4600	-3700	-2900
Q water vapor	BTU/hr	-100	-800	-500	-8600	-9600	-8400
Q condensate	BTU/hr	0	0	0	-500	-400	-200
Q loss	BTU/hr	1800	800	-500	-400	2200	2900

Table 6.2. Heat Transfer, Test Date: 01/18/2007

Overall heat transfer coefficients were calculated for each heat exchanger and for each test. These coefficients were calculated using Equation 2.1 where Q is "Q Cooling Water". Figure 6.4 shows the results.

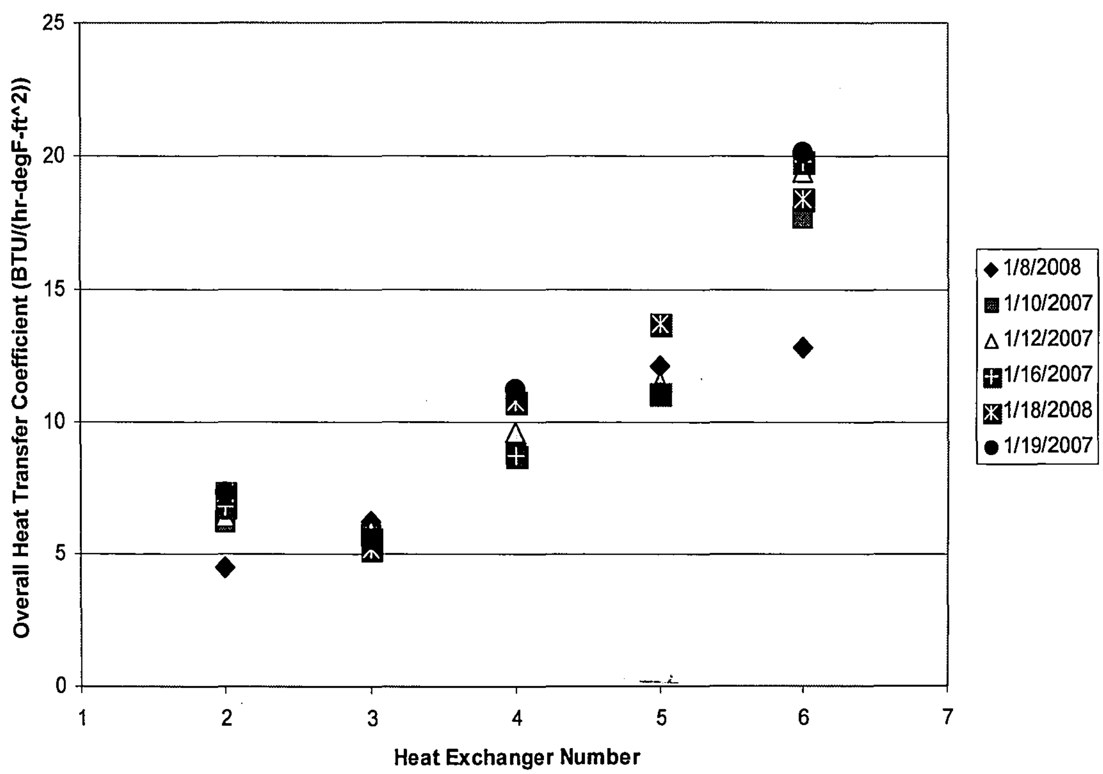

Figure 6.4. Overall Heat Transfer Coefficient Results

For the most part the results are relatively consistent. It is consistent that the condensing heat exchangers (HX 4 – 6) have a higher overall heat transfer coefficient. This is because condensation improves the convection heat transfer coefficient.

The amount of condensation by each heat exchanger varied slightly by test. Figure 6.5 shows how each condensing heat exchanger performed with respect to flue gas flow rate, and with respect to the other condensing heat exchangers.

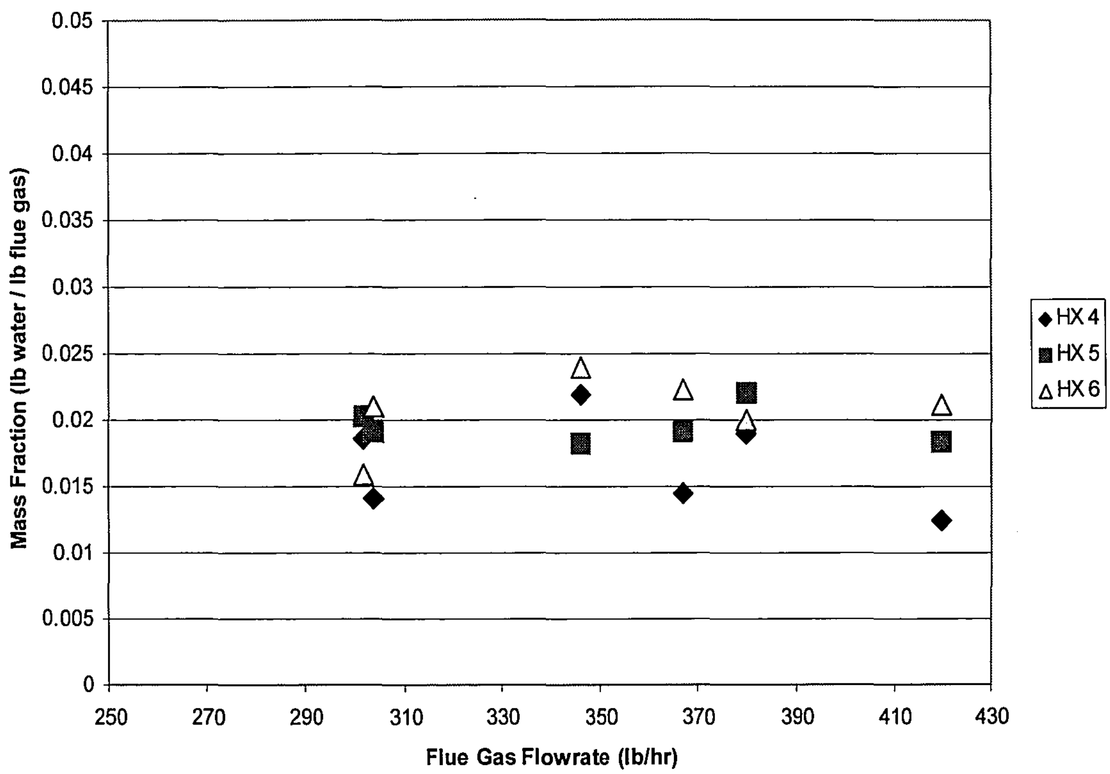

Figure 6.5. Individual Heat Exchanger Condensation Performance

One conclusion that can be taken from this plot is that the amount of water recovered from the flue gas does not change as flue gas flow rate increases. The individual heat exchanger sections did perform differently from test to test. Heat

Exchanger 6 condensed the most water, among the three condensing heat exchangers, four out of six times. The other two tests, Heat Exchanger 5 condensed the most water.

Perhaps the most important outcome of the test is the amount of total water condensed by the entire system. Figure 6.6 shows the total amount of water recovered in each test, as a fraction of the total water present in the flue gas.

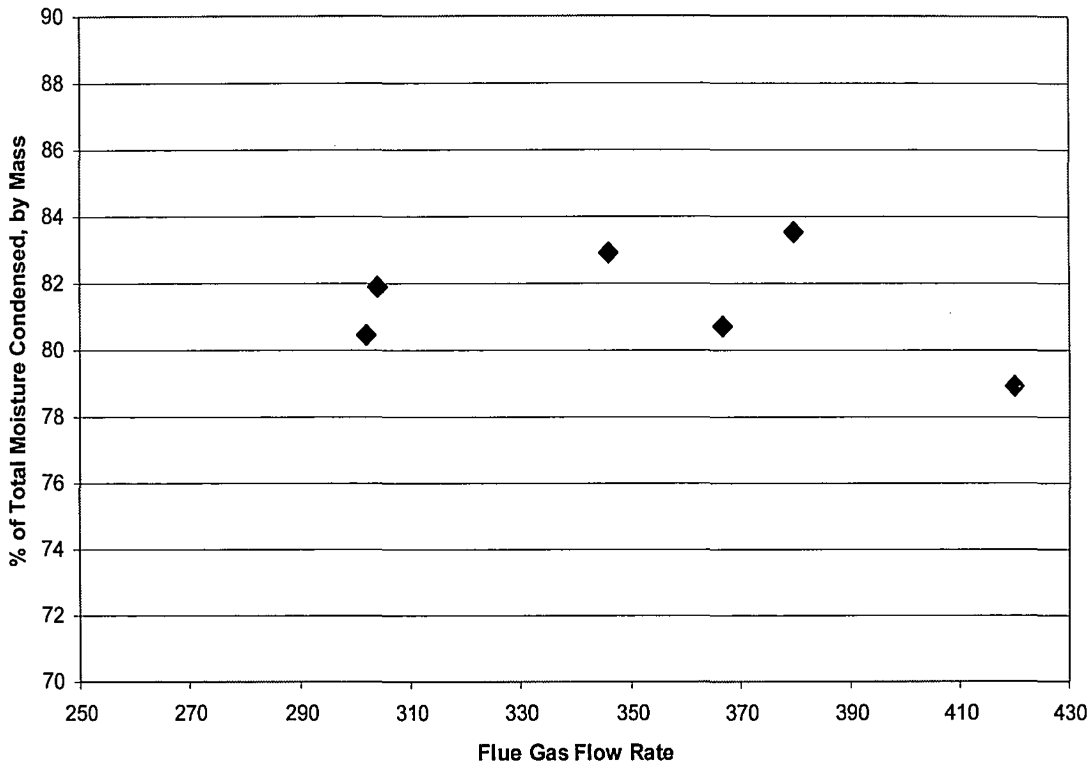

Figure 6.6. Total Water Recovered from Flue Gas

It can also be seen in Figure 6.6 that the total amount of water that can be recovered is not dependent on flue gas flow rate. In each test, roughly 80% (by mass) of the total water existing in the flue gas was condensed out and collected.

6.5 Acid Measurements

6.5.1 Flue Gas Measurements

The controlled condensation method was used at each port to measure the concentration of sulfuric acid mist in the flue gas after each heat exchanger coil. Figure 6.7 shows the results as a progression through the heat exchanger sections, for each test.

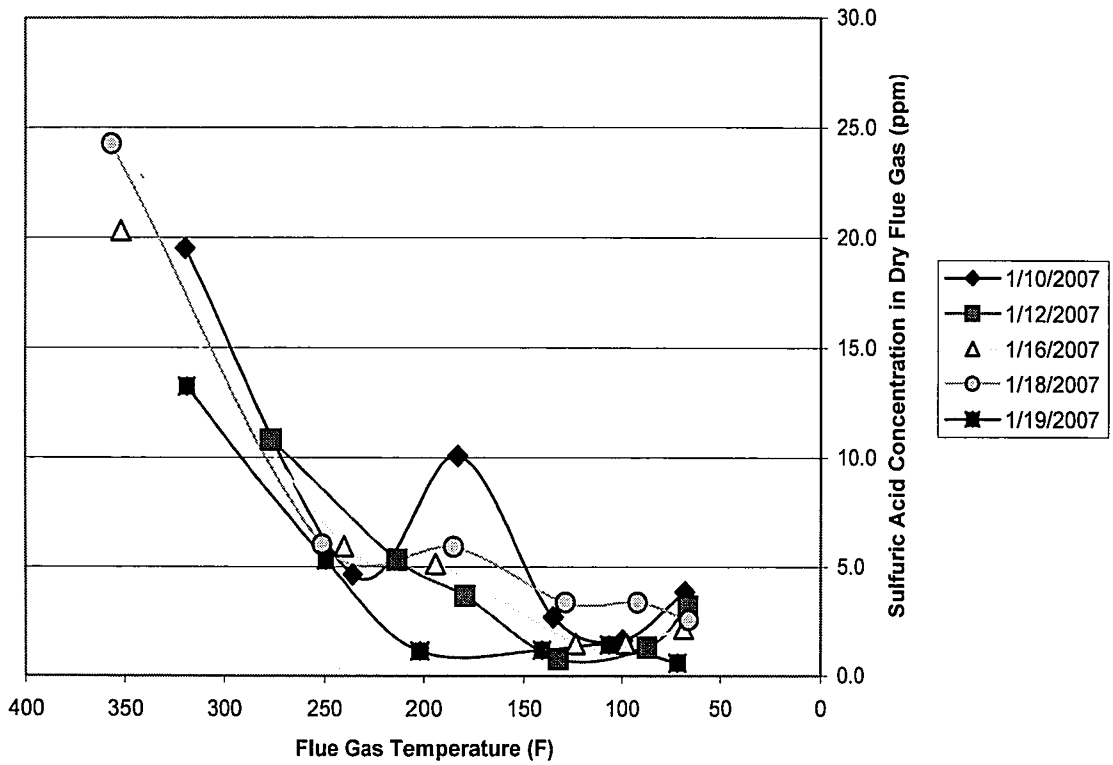

Figure 6.7. Sulfuric Acid Flue Gas Concentration as a function of Flue Gas Temperature

It can clearly be seen that there is a reduction in sulfuric acid flue gas concentration as the flue gas cools in each test. The concentrations of the flue gas in the condensing sections of the heat exchanger are all below 5 ppm. Figure 6.8 shows the overall sulfuric acid reduction in the flue gas for each test.

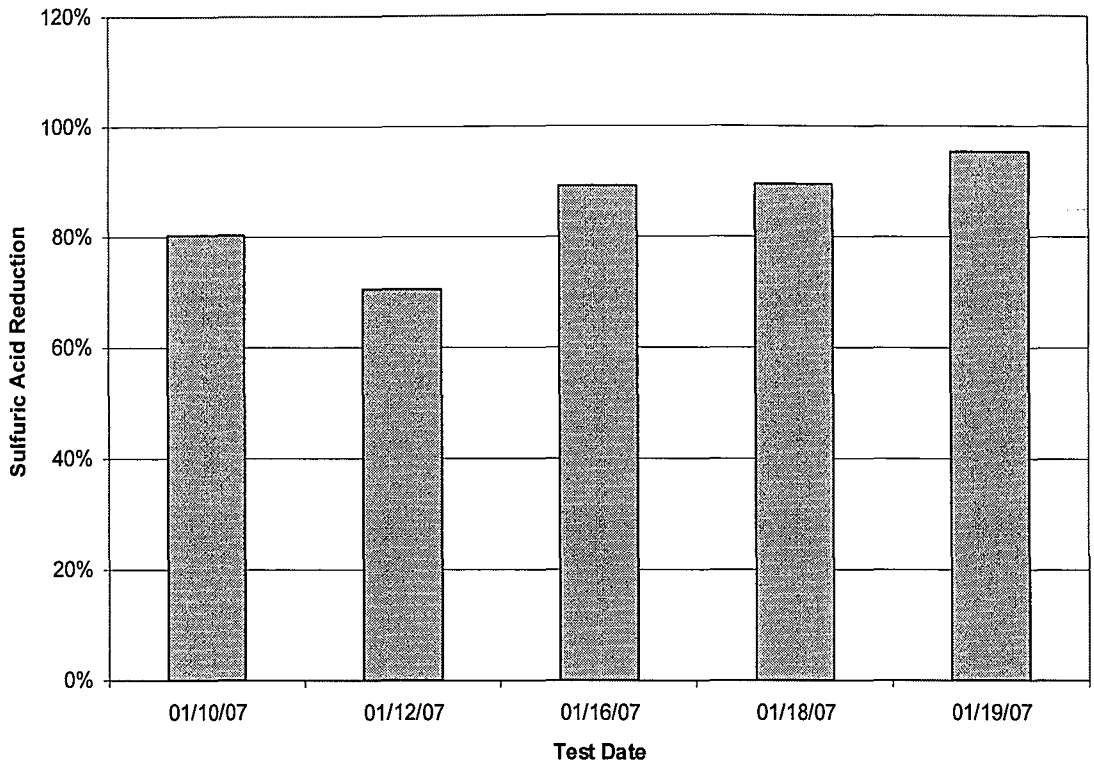

Figure 6.8. Total Sulfuric Acid Reduction in Flue Gas by Test Date

Among the five tests, it is consistent that about 80% of sulfuric acid was removed from the flue gas.

6.5.2 Heat Exchanger Condensate Acid Measurements

The condensate of only one test was analyzed for acids. The samples were sent to a laboratory and measured using ion chromatography. Sulfate (SO_4^{2-}) was tested for in all six heat exchangers. Nitrate (NO_3^-) and Chloride (Cl^-) were tested for in Heat Exchangers 4 – 6. Figures 6.9 – 6.11 show the concentrations in each heat exchanger condensate sample.

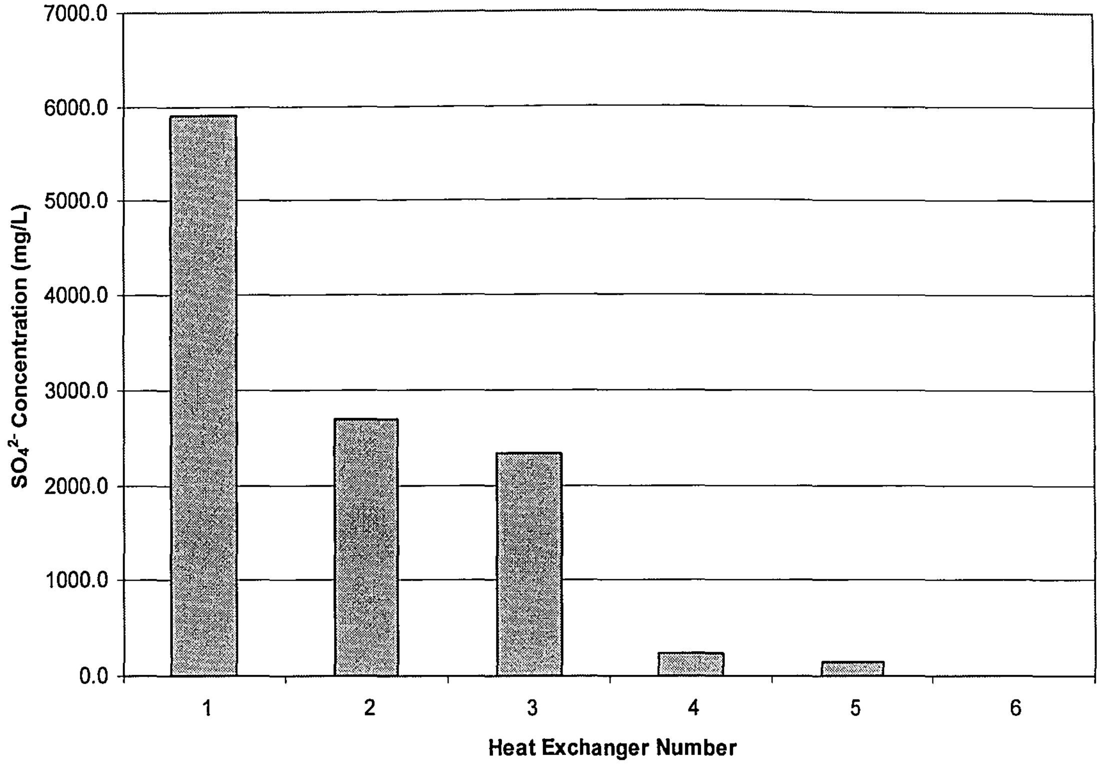

Figure 6.9. Heat Exchanger Condensate Analysis, Sulfuric Acid

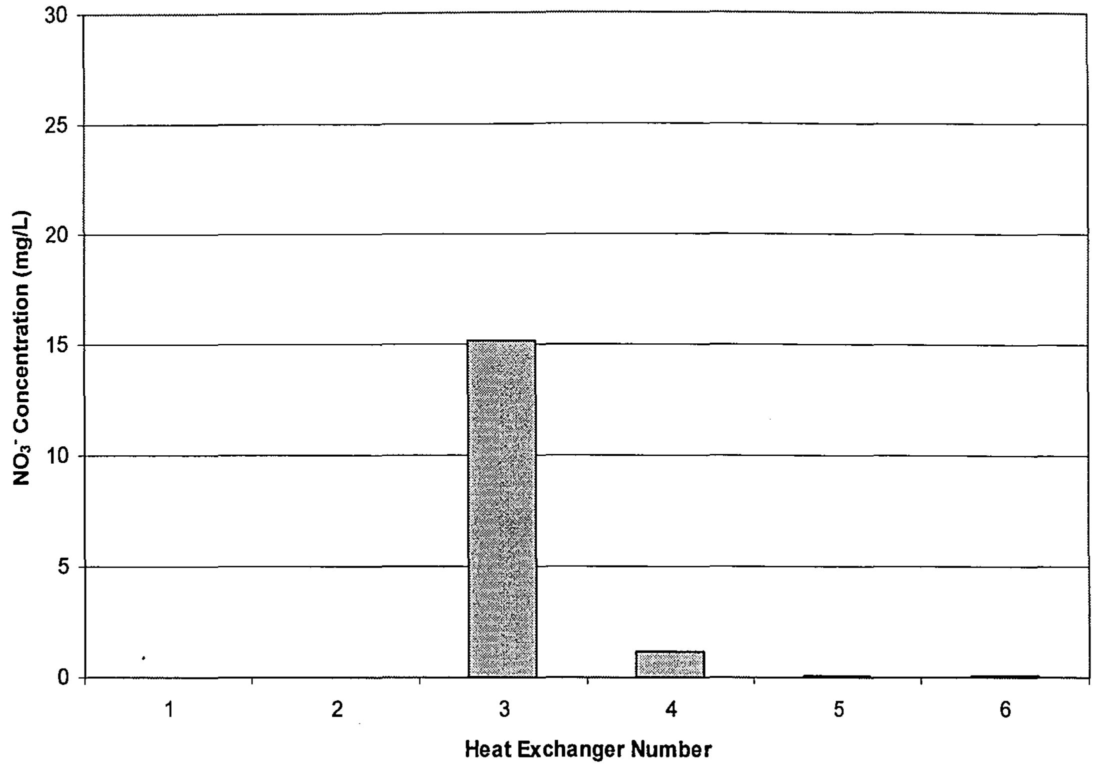

Figure 6.10. Heat Exchanger Condensate Analysis, Nitric Acid

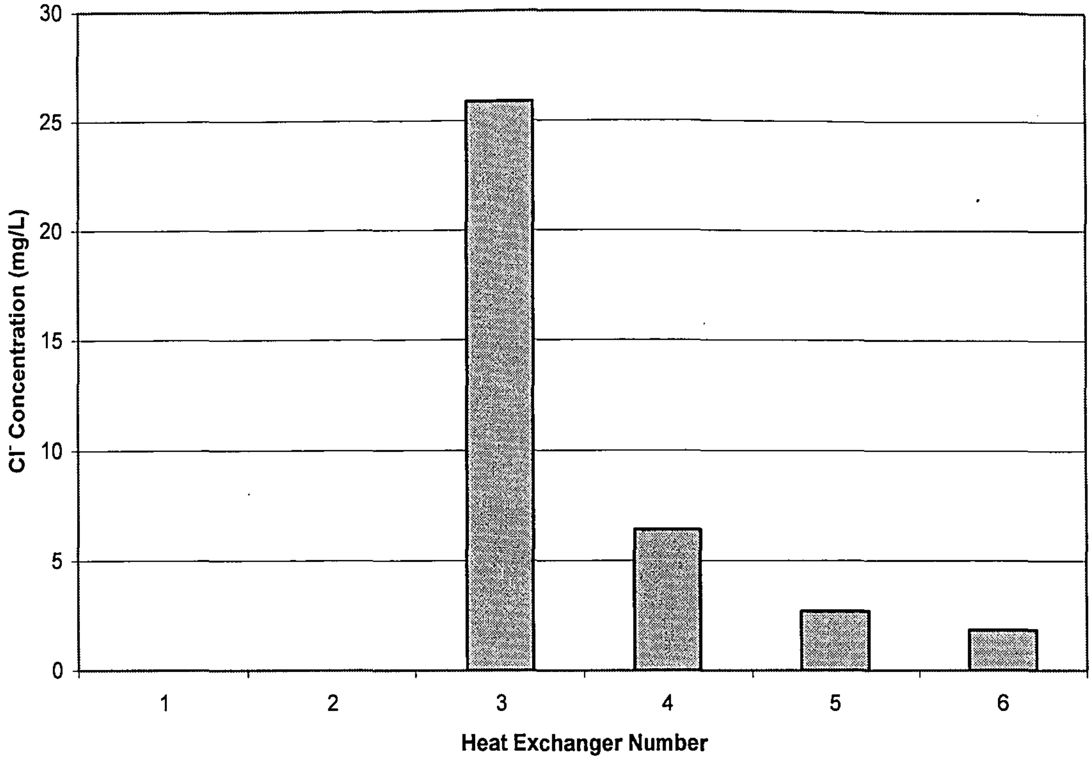

Figure 6.11. Heat Exchanger Condensate Analysis, Hydrochloric Acid

Figure 6.9 shows sulfuric acid concentrations of nearly 6000 mg/L (equivalent to ppm by mass). While this is a high concentration (0.6 % by mass), it is not nearly as high as expected. This figure supports the hypothesis that the majority of the sulfuric acid would condense out in the first three heat exchangers. Sulfuric acid is negligible in the condensing heat exchangers (HX 4 – 6).

Figures 6.10 and 6.11 show very low concentrations of nitric and hydrochloric acids. Heat Exchanger 3 seemed to have the most significant amount of acids. This is contrary to what was expected. It was expected that the condensing heat exchangers would have the most significant amounts of nitric and hydrochloric acid because of the expected flue gas temperature. On a positive note, the condensate collected in HX 4 – 6 is considerably clean.

7.0 Discussion

The primary goal of the project was to recover water from flue gas. Eighty percent of the water in the flue gas (by weight) was recovered in five out of six tests. This is a substantial portion of the existing water, and from this standpoint, the project was successful.

The secondary goal of the project was to recover heat from the flue gas. This was accomplished in conjunction with the first goal, simply by lowering the stack temperature of the flue gas. A low stack temperature is essential to achieving a high thermal efficiency. The heat was recovered from the flue gas and transferred to the heat exchanger cooling water. The condensate does have an acidic nature and depending on the use, water treatment may be necessary to neutralize the acid.

The final goal of the project was to reduce emissions. Figure 6.8 showed that the sulfuric acid mist emissions were reduced by about 80% in every test. Chlorides and nitrates were also shown to be present in the condensate, highlighting two more pollutants that were captured instead of being emitted into the atmosphere.

7.1 Complications

One of the major discrepancies in the analysis of the results was the heat balance. Appendix C shows the heat balance for each of the six tests. Heat Exchangers 3 & 4 both showed a heat gain to the system in every test. This is an obvious error because there is no reason why there would be any heat gain to the system. There should be calculated heat loss from the system because the insulation cannot be perfect. The tests were run in wintertime when the ambient temperature was low enough to suspect significant heat loss. The hot water heater was a source of heat gain to the system, but this was taken into account with thermocouples in the water lines before and after the water heater.

One likely cause of this error is in the measurement of the amount of water condensed from each individual heat exchanger section. As moisture is condensed in one heat exchanger, the water collects on the tube surface and should drip down into the drain for that particular heat exchanger. Because of the high velocity of the flue gas through the heat exchanger tube banks, it is possible that some of the moisture from one heat exchanger is entrained in the flue gas and carried into the next heat exchanger. This would attribute condensation of one heat exchanger to the subsequent heat exchanger. This would in effect create a situation where a lower amount of condensate would be measured than expected from the temperature rise in the cooling water. Since the heat exchangers are so close to each other, this theory is likely the reason why a heat gain is being observed in Heat Exchangers 3 & 4, and not in 5 & 6.

A mass balance for sulfuric acid was also attempted. More acid was measured in the flue gas than was measured in the condensate. Also, in Figure 6.7 it can be seen that some flue gas sulfuric acid measurements were higher as the flue gas progressed through the heat exchanger. This suggests that the noise range of the controlled condensation manual method is too large to consider an acid mass balance. The actual uncertainty range could only be obtained with repeat testing.

Significant fouling was observed on the three heat exchangers that were not condensing water (HX 1 – 3). Black soot had built up on the tube wall surface. Upon inspection, the heat exchanger tubes were removed and rinsed off with a neutralizing solution. There was no observed corrosion over the period of testing, on any of the heat exchangers. Also, there was no soot buildup on the condensing heat exchangers (HX 4 – 6).

Some of the tube wall surface thermocouples were destroyed during testing. This is apparent by looking at Appendix A. Some of the measured tube wall surface temperatures are far off from the calculated values and are closer to the actual flue gas temperature in many cases. This is because after examination it was found that the thermocouple insulation had deteriorated (most likely due to the acid condensation). Because the thermocouple wires were exposed, they ended up short-circuiting in the flue gas environment, which means the thermocouple is measuring the temperature at that point: the flue gas. This is why during some tests the tube wall surface thermocouple would read a reasonable value for a while, and then jump 50 – 100 degrees F for awhile, and then go back to a reasonable temperature. In later tests, these thermocouples measured consistently higher than expected. This is because the thermocouple wire had become completely detached from the thermocouple in some instances. The calculated tube wall surface temperatures in Appendix A are considered to be more accurate than the measured values.

7.2 Conclusions

The acid in the condensate collected was much less than was anticipated. Since the dew point of sulfuric acid is higher than that of water, it was expected that very high concentrations of sulfuric acid would be observed in Heat Exchangers 1-3. The highest sulfuric acid concentration measured was only 0.6% (6000 mg/L). In the condensing heat exchangers, the concentrations of sulfuric, nitric, and hydrochloric acids were even lower. Sulfuric acid was under 250 mg/L, hydrochloric acid was less than 10 mg/L, and nitric acid was less than 5 mg/L.

These results mean that super alloys or other technologies may not be needed to avoid heat exchanger corrosion. More testing would need to be done to observe any variance in acid measurement and heat exchanger corrosion. There also should be more work done to observe nitric and hydrochloric acids in all heat exchanger sections.

This thesis shows that a significant portion of water can be recovered from boiler flue gas, and in doing so, sulfuric acid emissions are reduced.

7.3 Next Steps

Future work for this project includes the use of finned heat exchangers to reduce the overall size of the heat exchangers. Also, this system will be transported to a full-scale coal fired power plant in order to analyze the effects of different flue gas.

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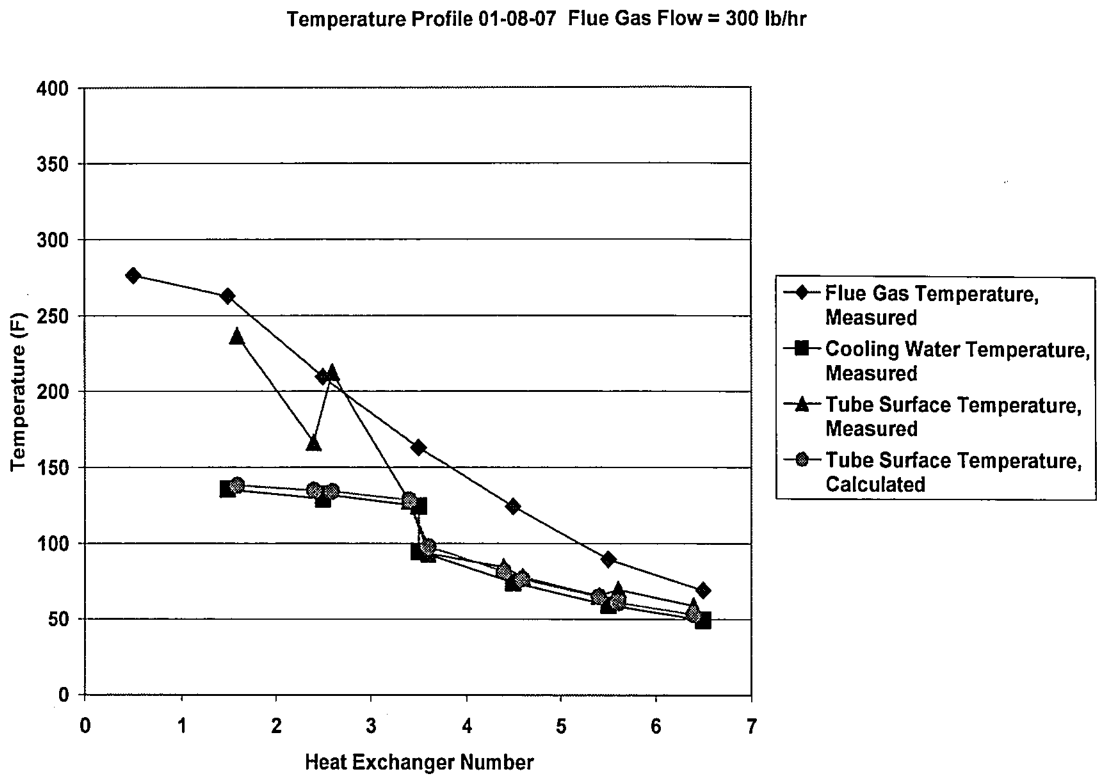
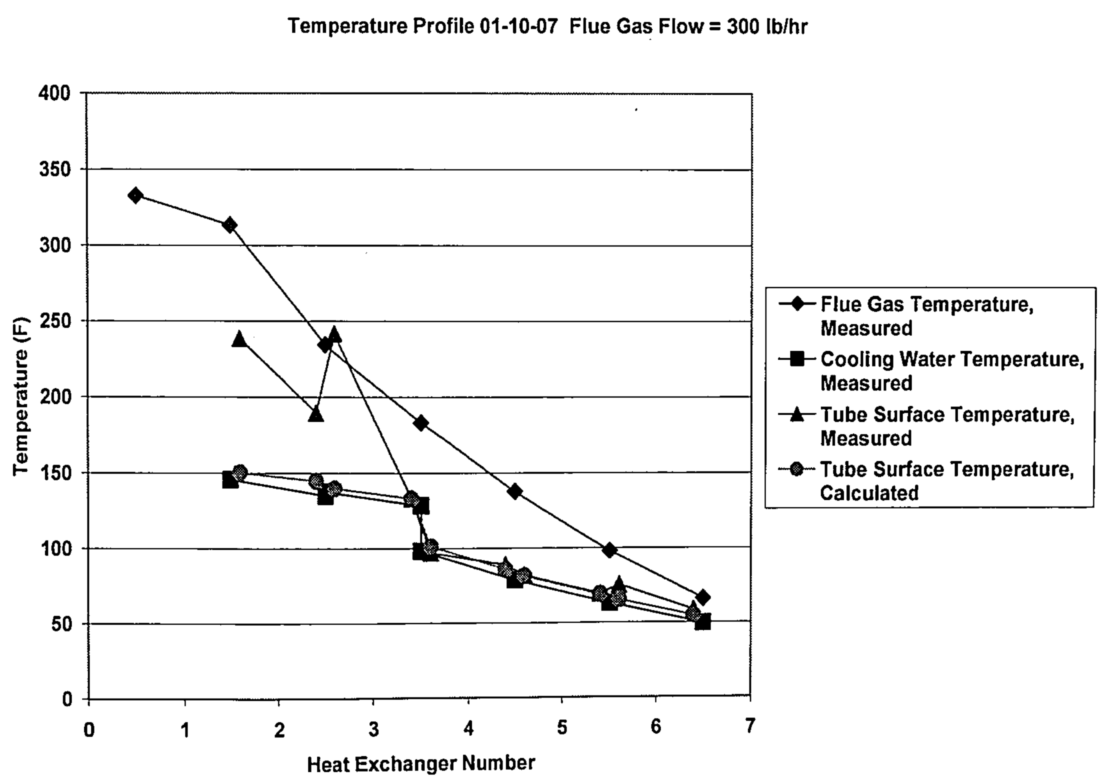
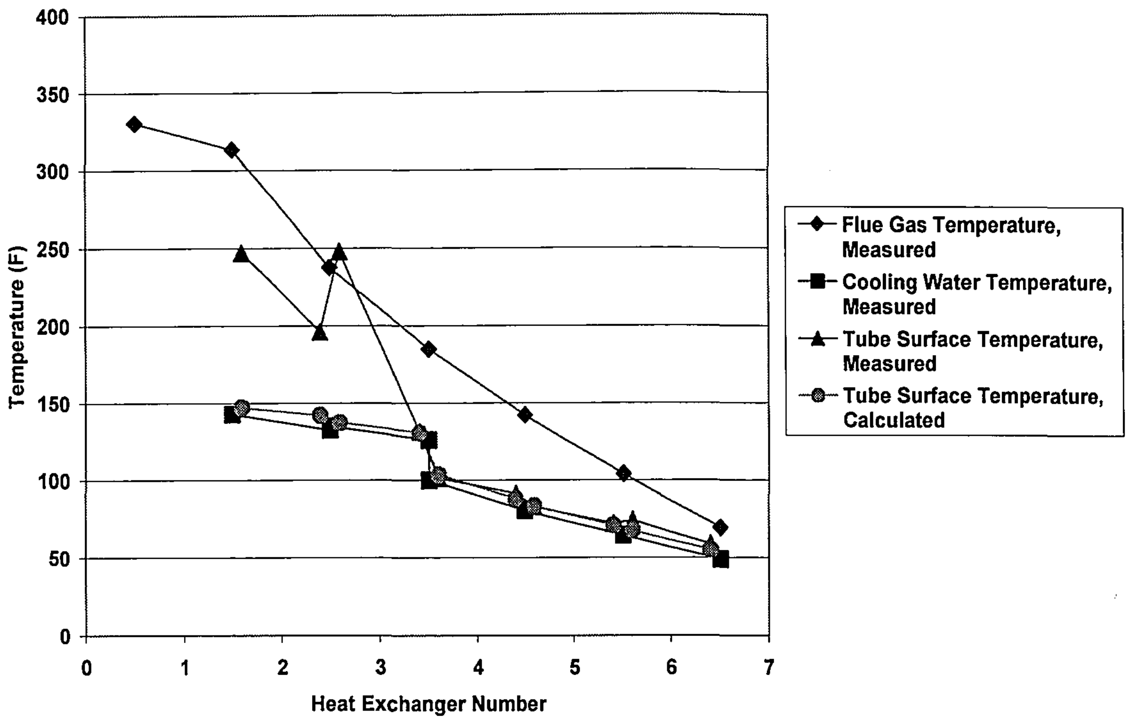
Appendix A: Temperature Profiles

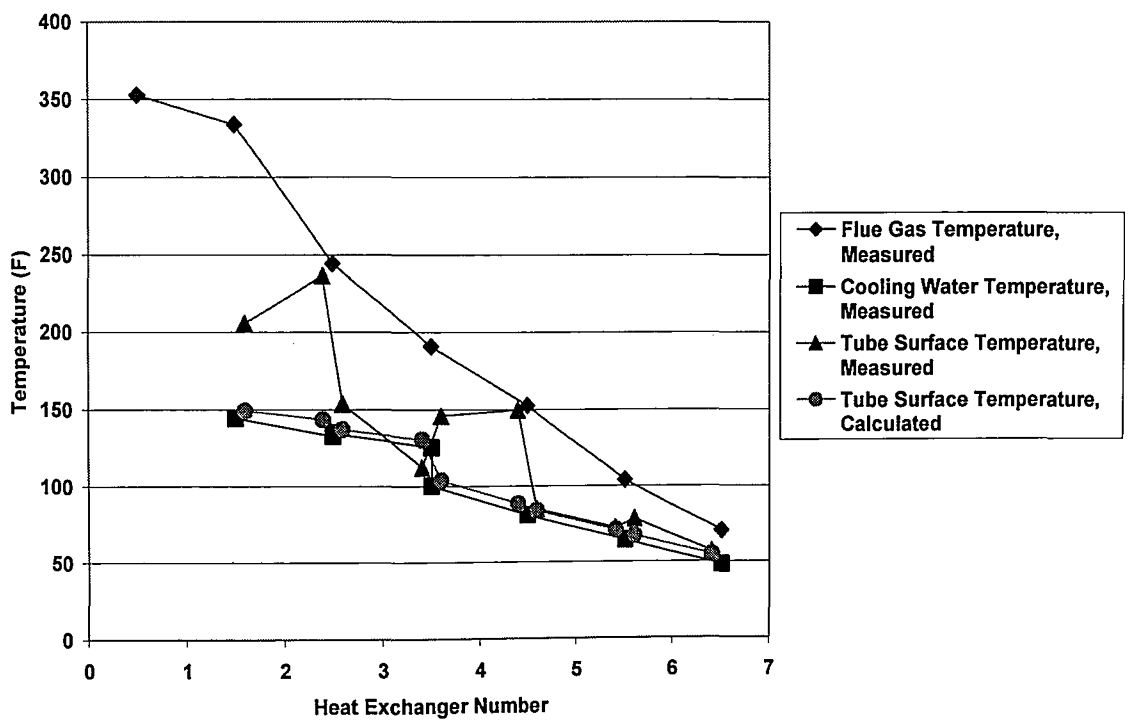




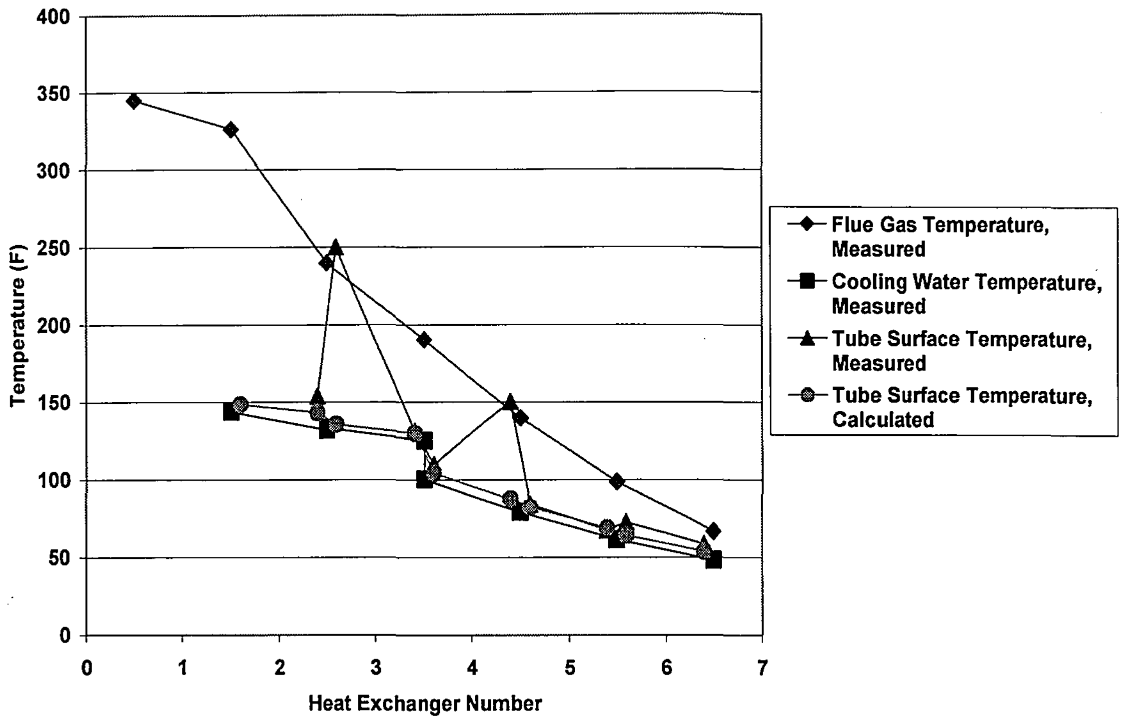
Temperature Profile 01-12-07 Flue Gas Flow = 370 lb/hr



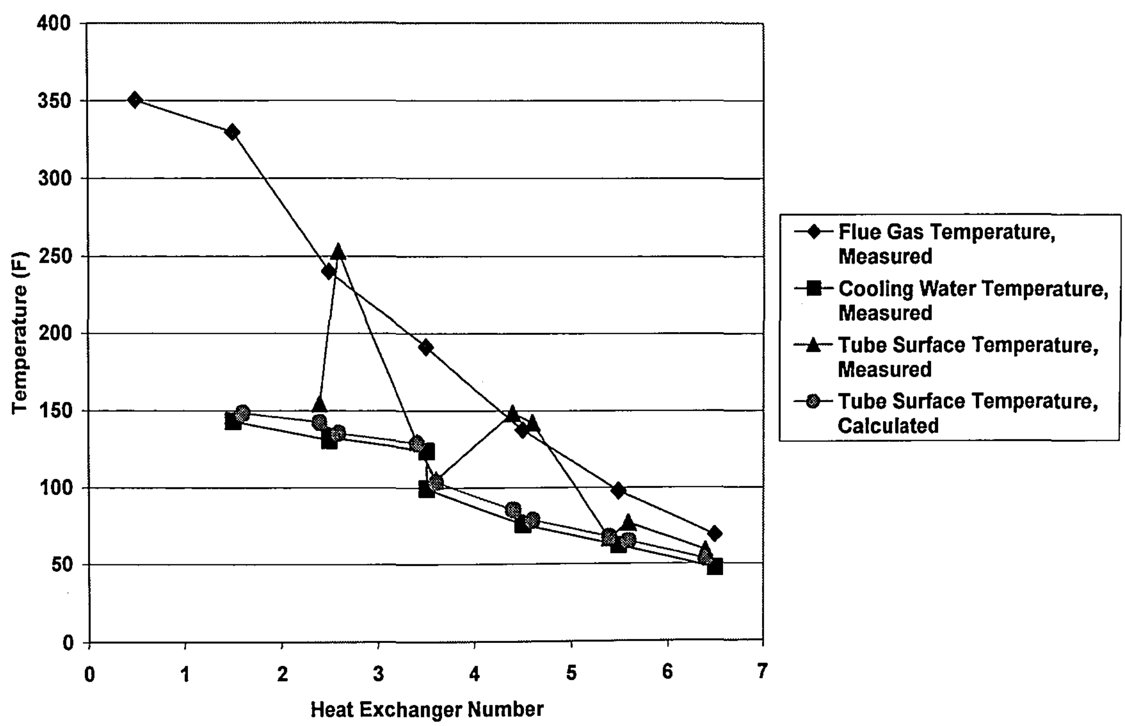
Temperature Profile 01-16-07 Flue Gas Flow = 420 lb/hr



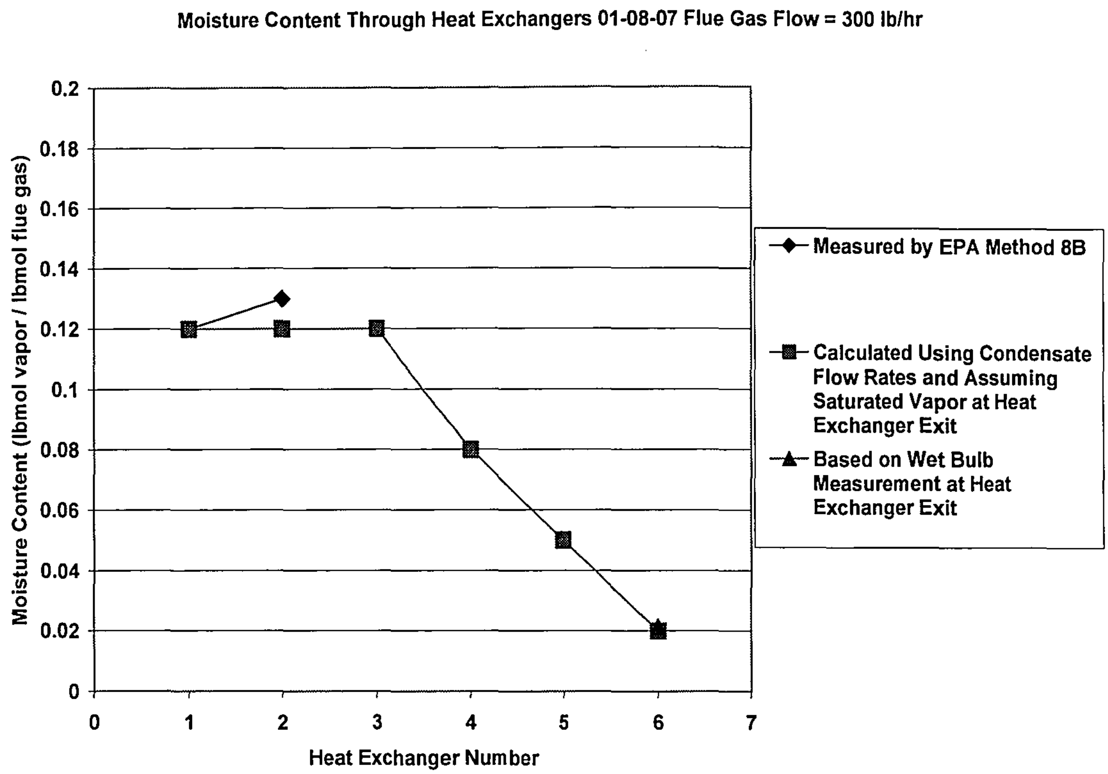
Temperature Profile 01-18-07 Flue Gas Flow = 380 lb/hr

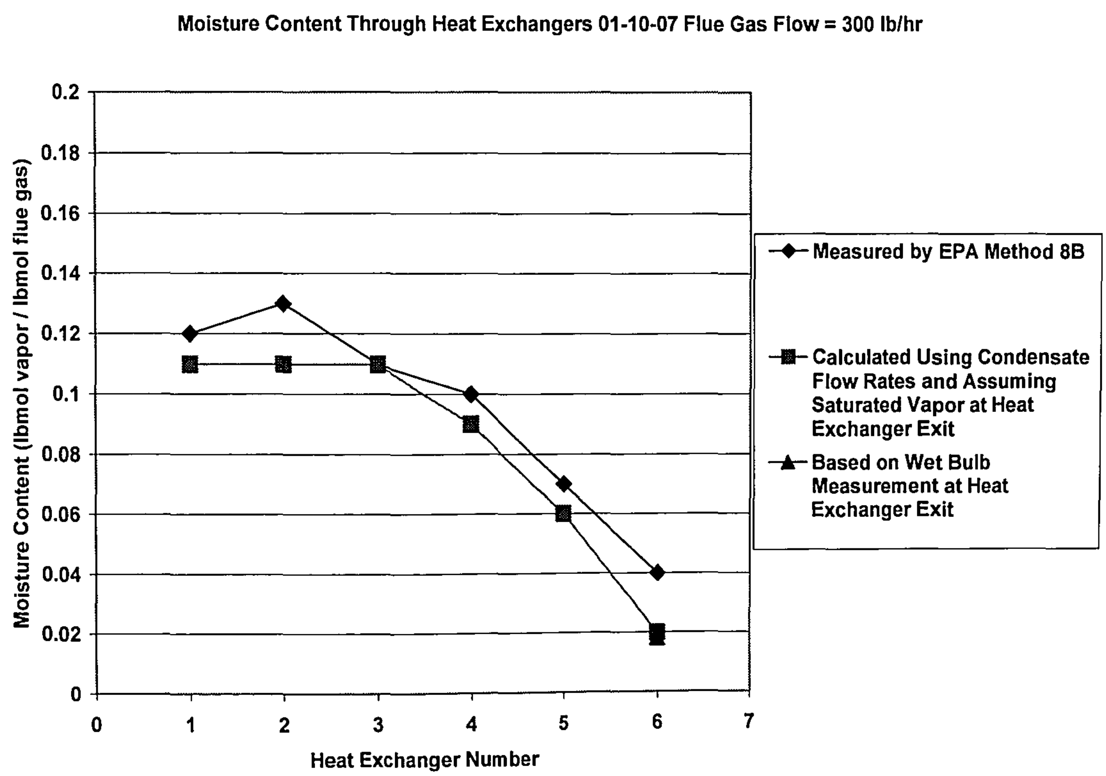


Temperature Profile 01-19-07 Flue Gas Flow = 350 lb/hr

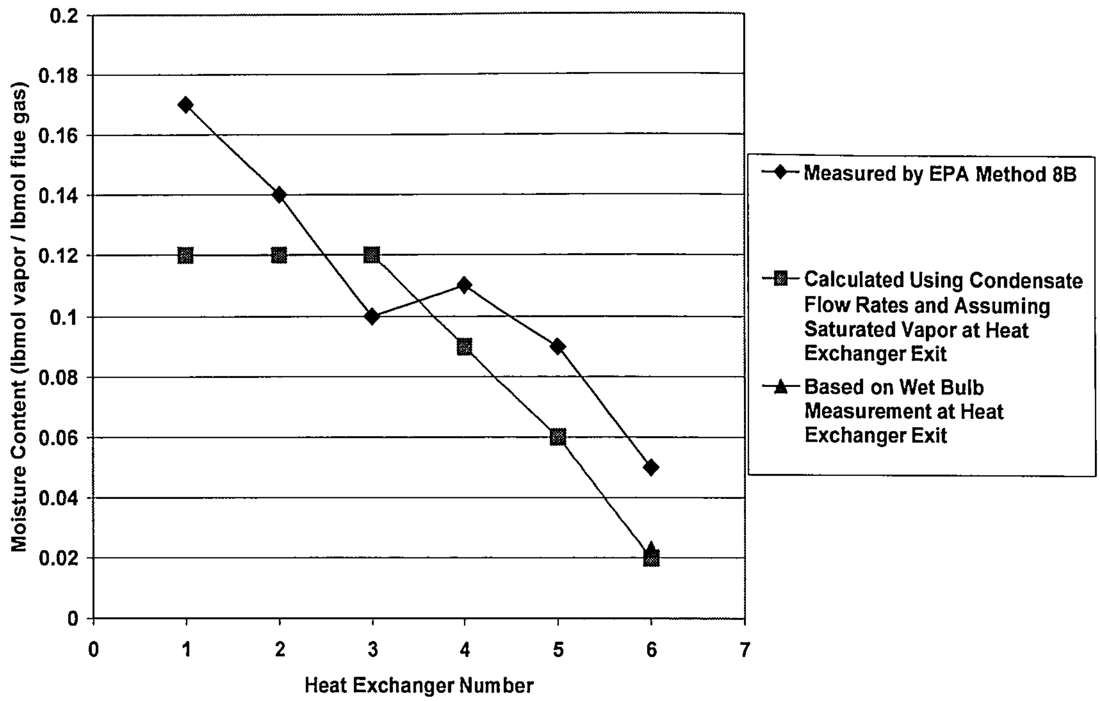


Appendix B: Moisture Content

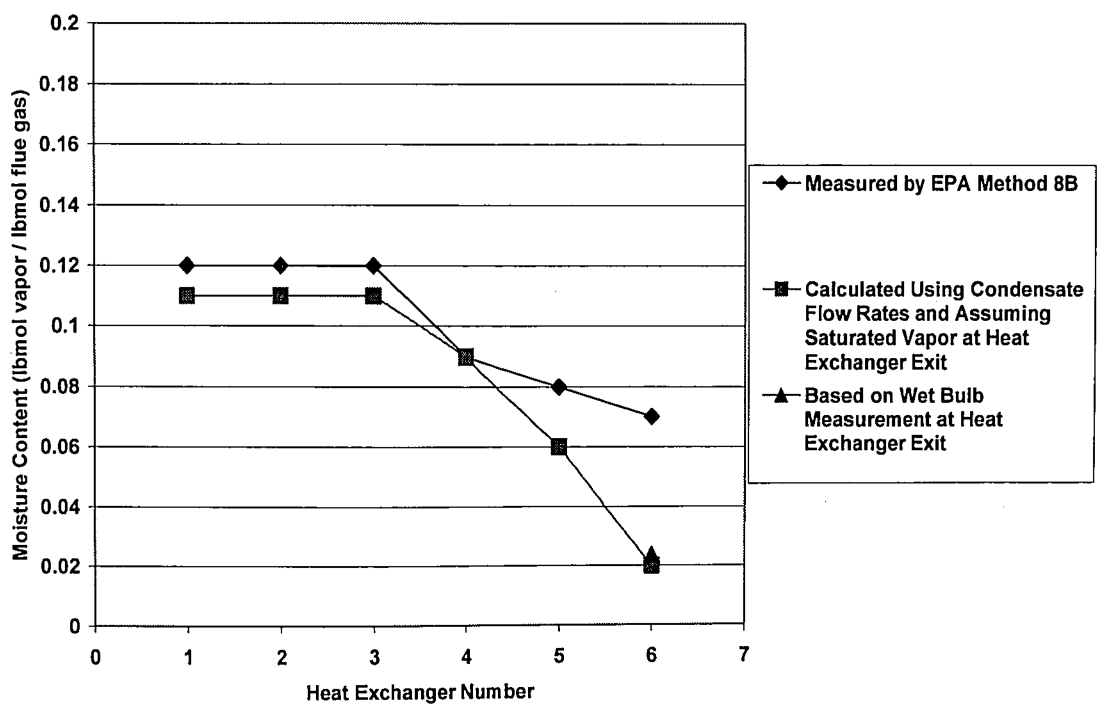




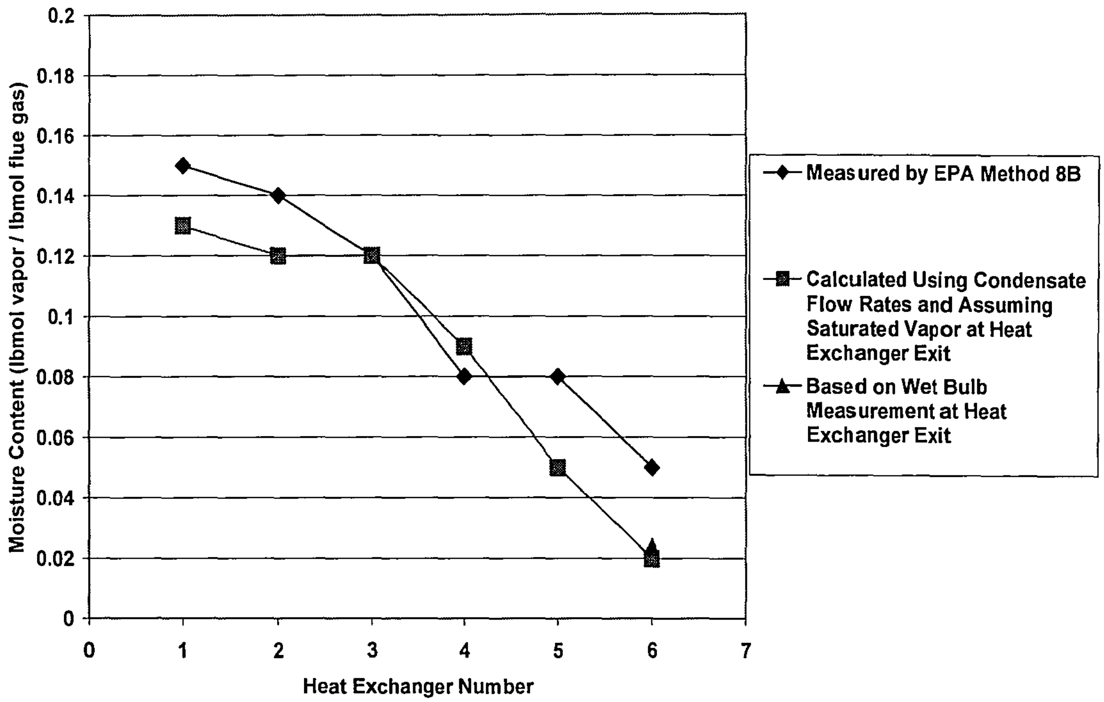
Moisture Content Through Heat Exchangers 01-12-07 Flue Gas Flow = 370 lb/hr



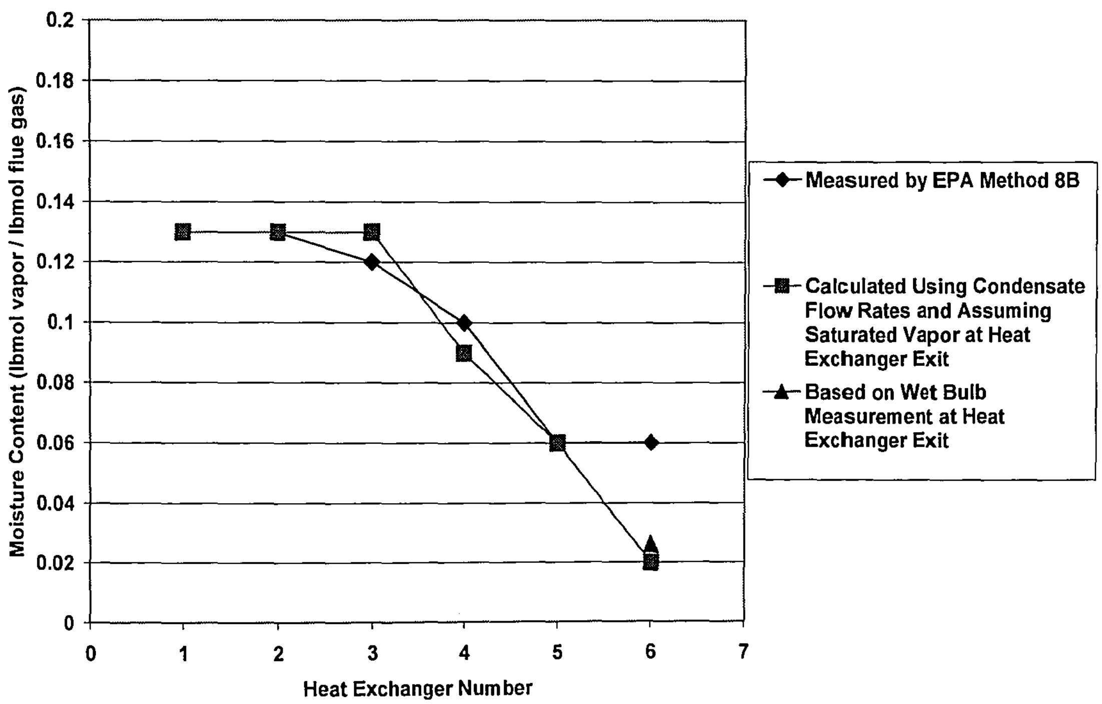
Moisture Content Through Heat Exchangers 01-16-07 Flue Gas Flow = 420 lb/hr



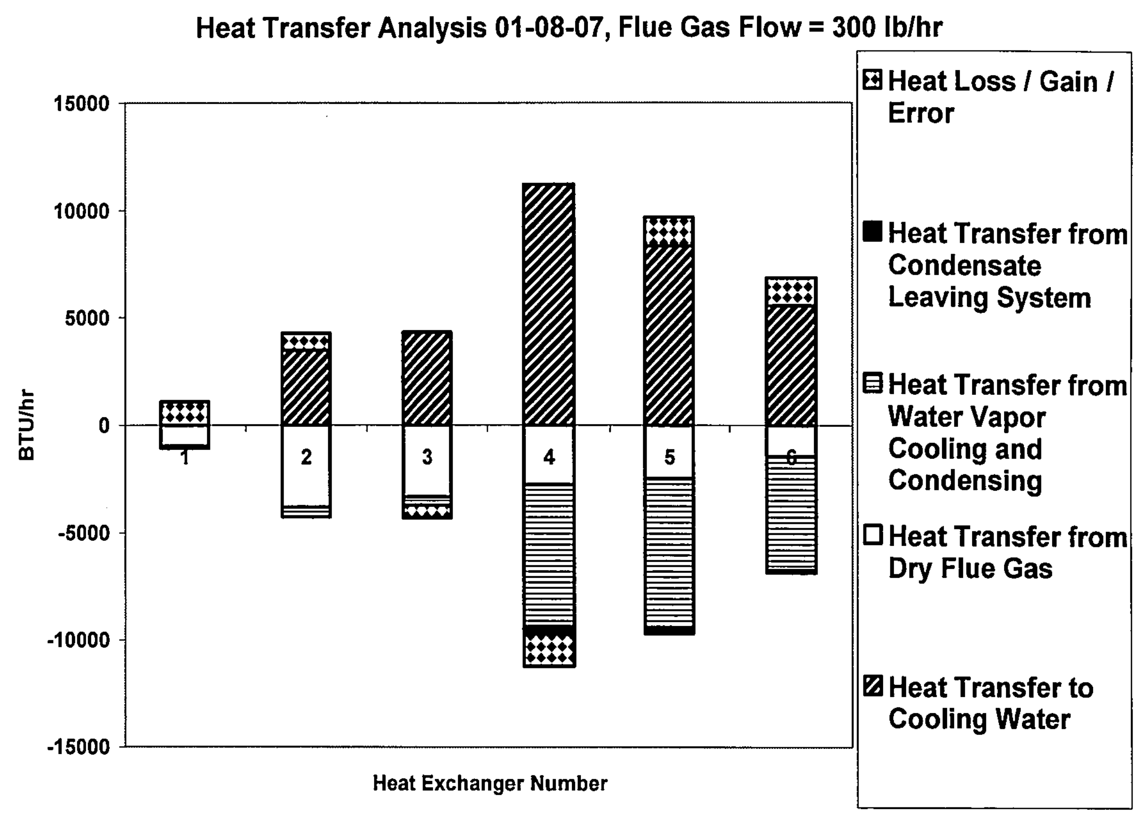
Moisture Content Through Heat Exchangers 01-18-07 Flue Gas Flow = 380 lb/hr

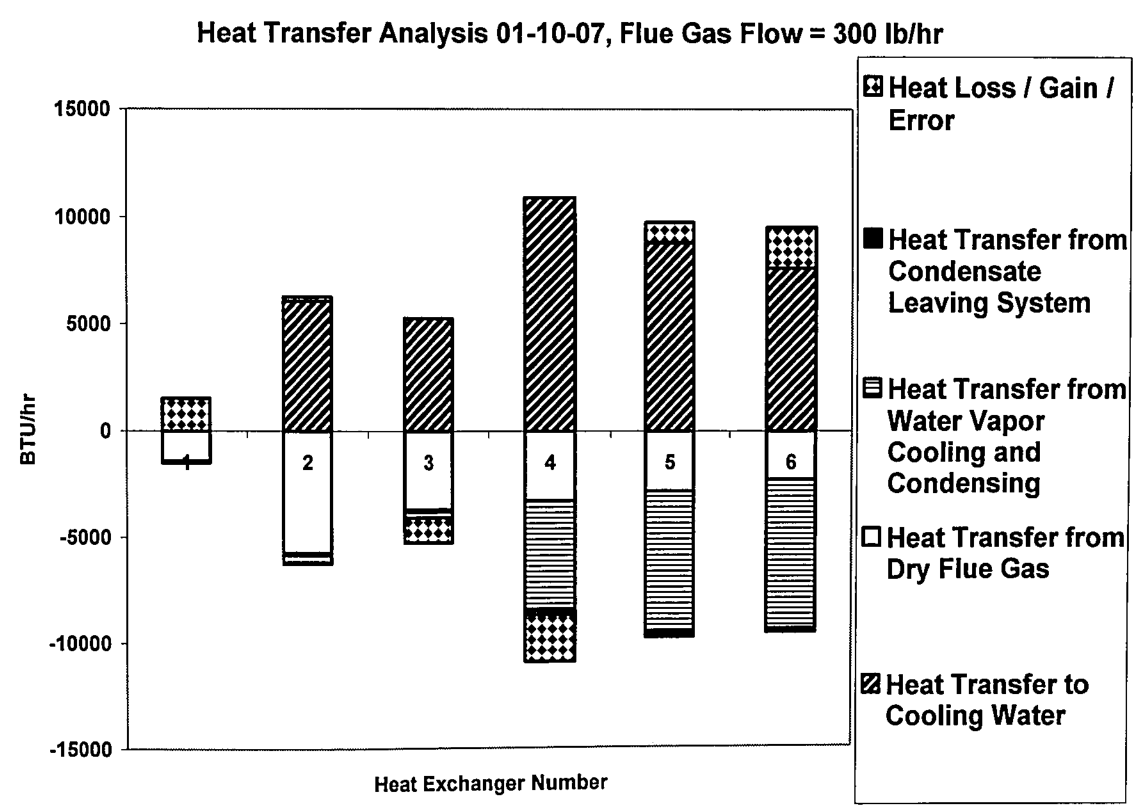


Moisture Content Through Heat Exchangers 01-19-07 Flue Gas Flow = 350 lb/hr



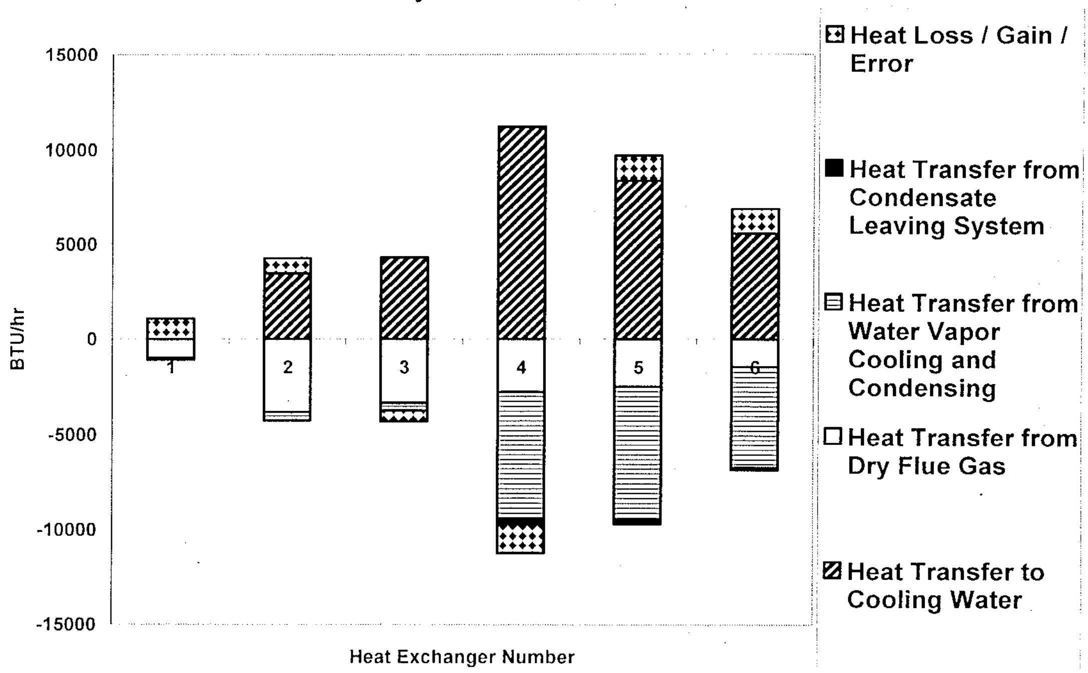
Appendix C: Heat Balance



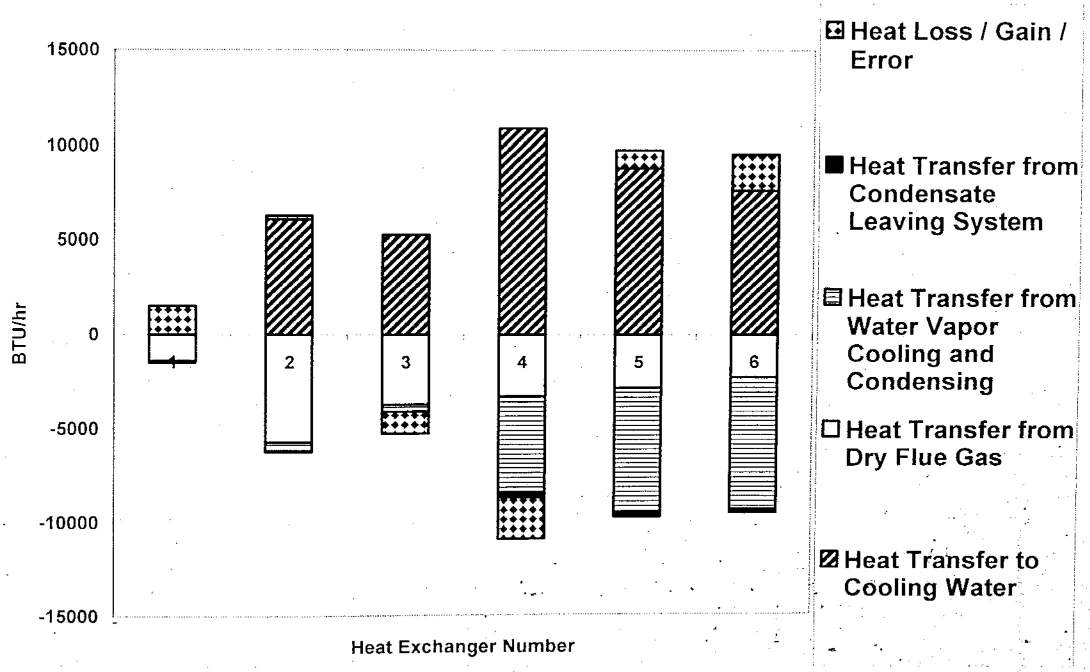


Appendix C: Heat Balance

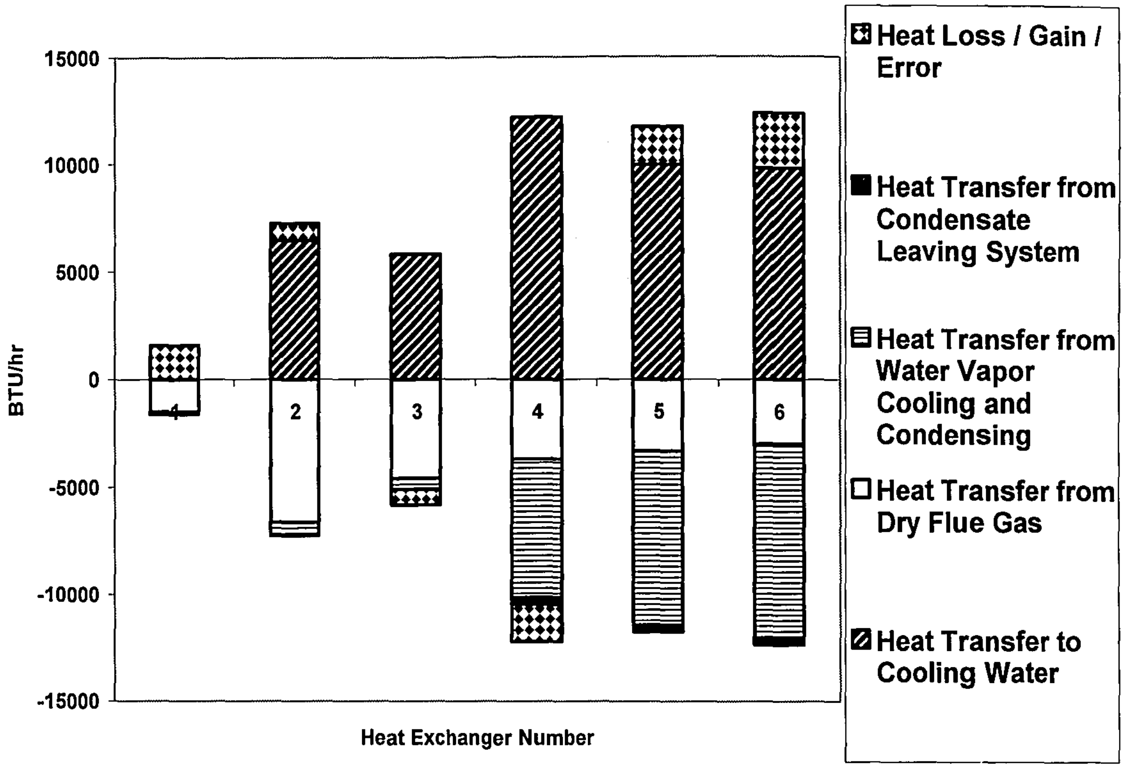
Heat Transfer Analysis 01-08-07, Flue Gas Flow = 300 lb/hr



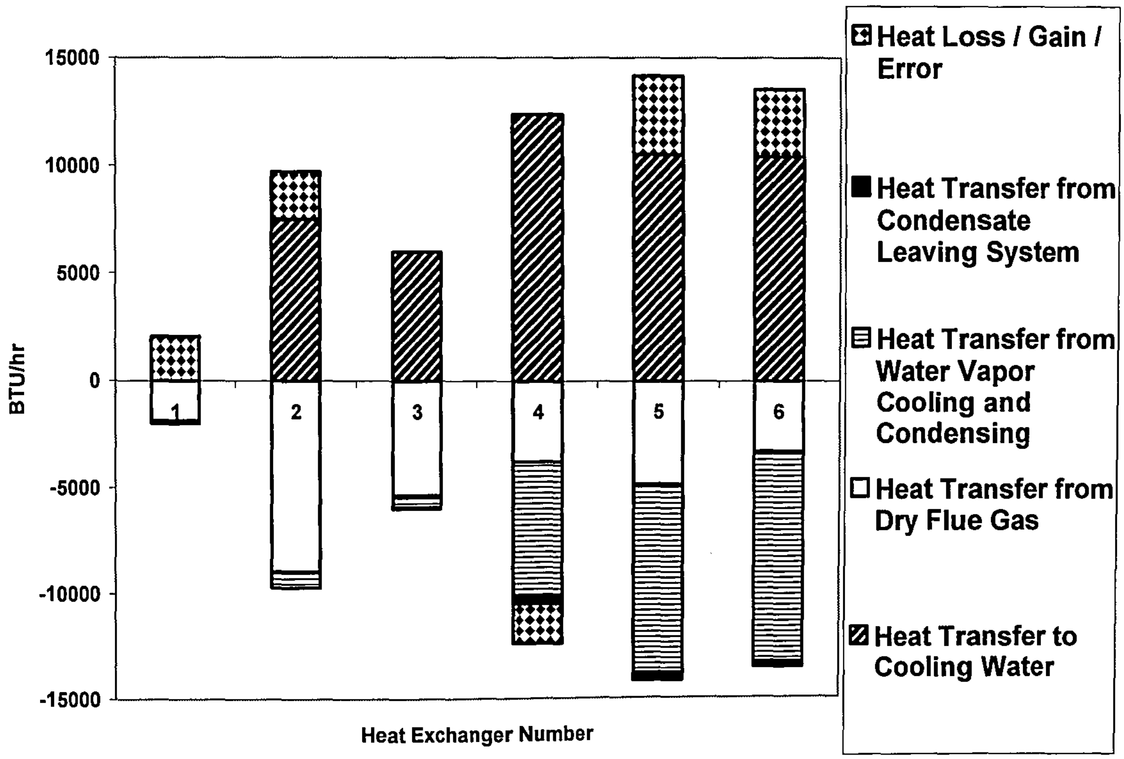
Heat Transfer Analysis 01-10-07, Flue Gas Flow = 300 lb/hr



Heat Transfer Analysis 01-12-07, Flue Gas Flow = 370 lb/hr

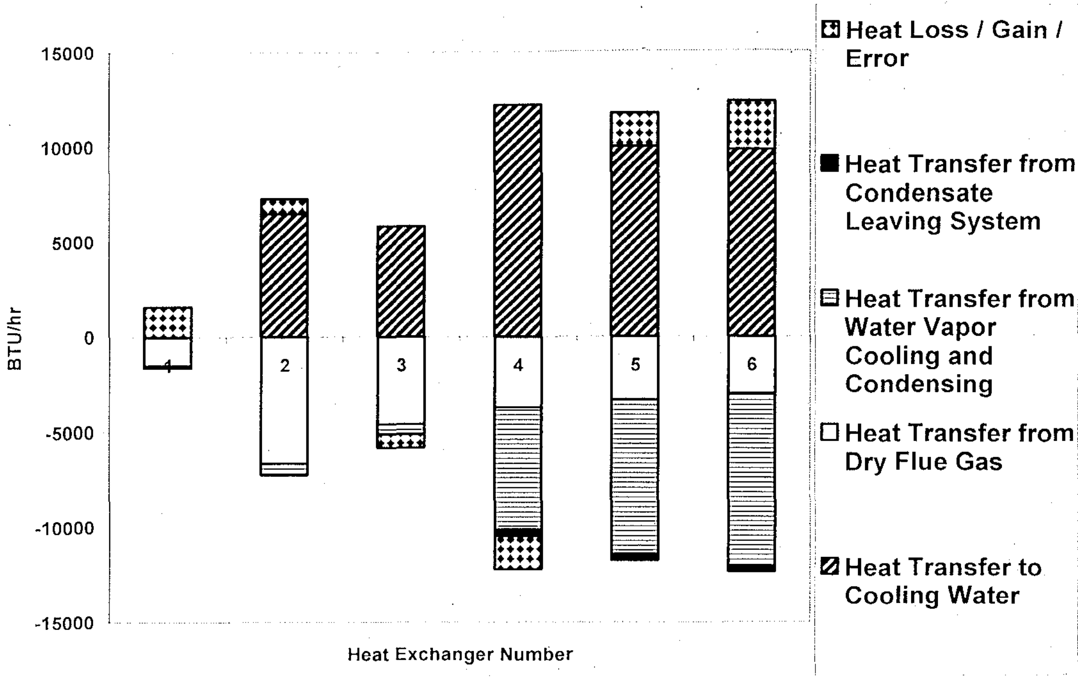


Heat Transfer Analysis 01-16-07, Flue Gas Flow = 420 lb/hr

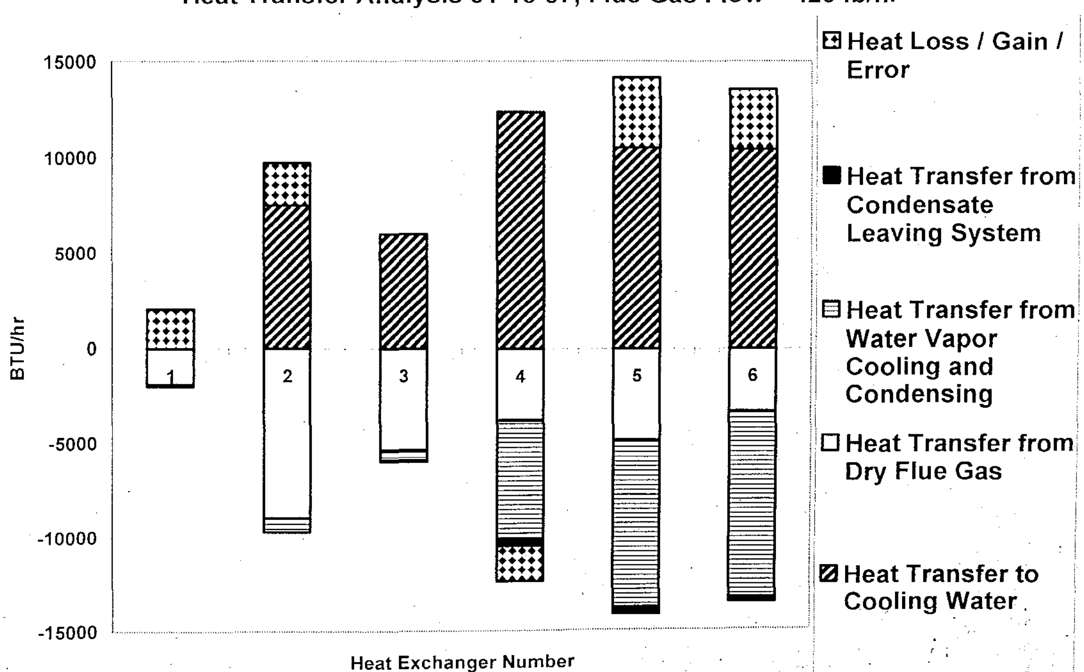


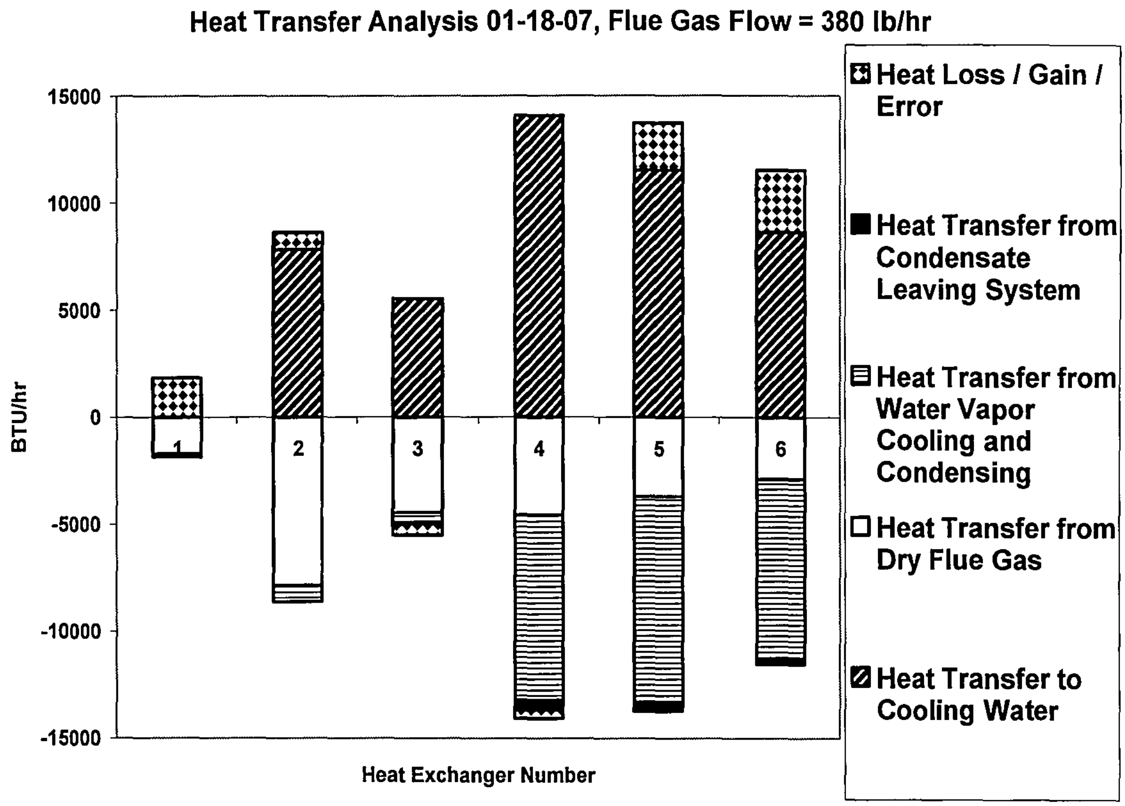
INTENTIONAL SECOND EXPOSURE [redacted]

Heat Transfer Analysis 01-12-07, Flue Gas Flow = 370 lb/hr

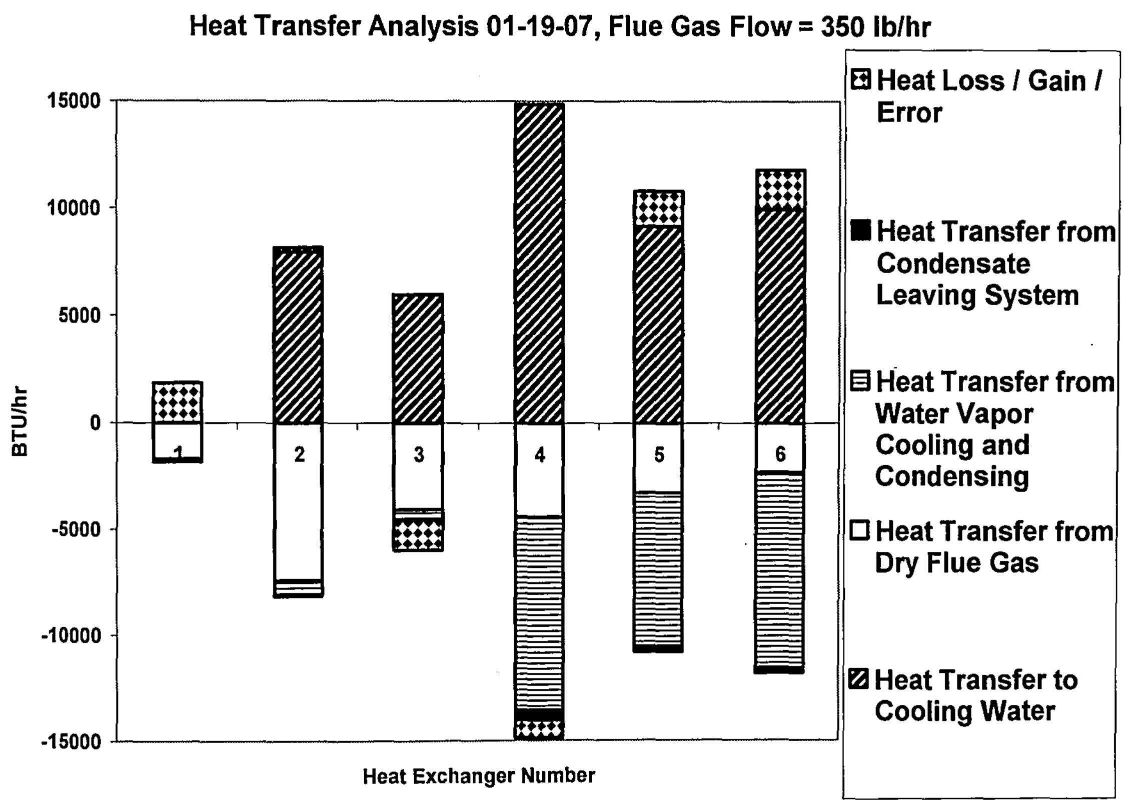


Heat Transfer Analysis 01-16-07, Flue Gas Flow = 420 lb/hr





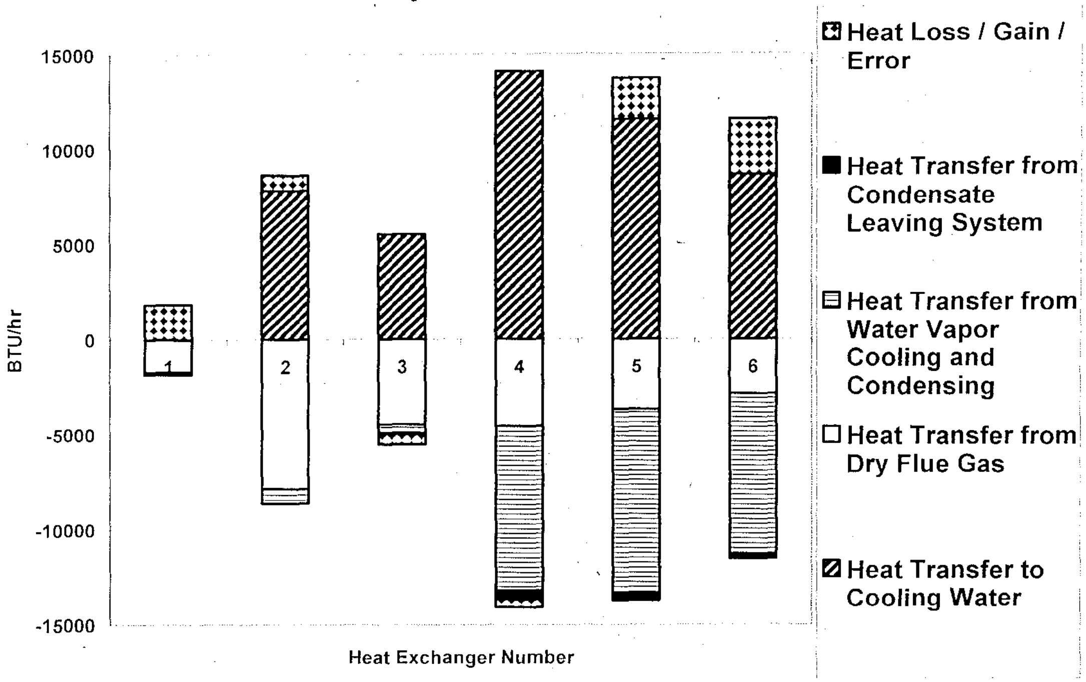
Heat Transfer Analysis 01-18-07, Flue Gas Flow = 380 lb/hr
15000
10000
5000
0
-5000
-10000
-15000
BTU/hr
1
2
3
4
5
6
Heat Exchanger Number
Heat Loss / Gain / Error
Heat Transfer from Condensate Leaving System
Heat Transfer from Water Vapor Cooling and Condensing
Heat Transfer from Dry Flue Gas
Heat Transfer to Cooling Water



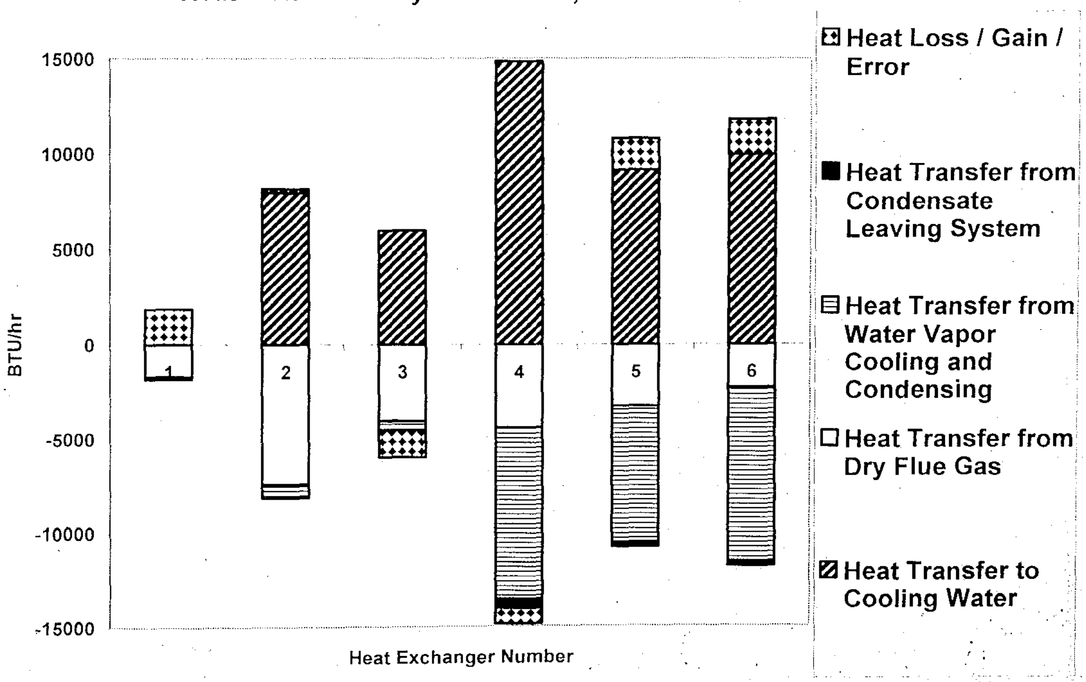
Heat Transfer Analysis 01-19-07, Flue Gas Flow = 350 lb/hr
15000
10000
5000
0
-5000
-10000
-15000
BTU/hr
1
2
3
4
5
6
Heat Exchanger Number
Heat Loss / Gain / Error
Heat Transfer from Condensate Leaving System
Heat Transfer from Water Vapor Cooling and Condensing
Heat Transfer from Dry Flue Gas
Heat Transfer to Cooling Water

INTENTIONAL SECOND EXPOSURE

Heat Transfer Analysis 01-18-07, Flue Gas Flow = 380 lb/hr



Heat Transfer Analysis 01-19-07, Flue Gas Flow = 350 lb/hr



Vita

Christopher A. Samuelson was born in Carlisle, Pennsylvania on November 14, 1981. His parents are Carl L. Samuelson, Jr. and Robin E. Samuelson. Christopher attended Boiling Springs High School and continued on to graduate from Villanova University with a Bachelor of Science degree in Mechanical Engineering. He then worked for the Energy Research Center while pursuing a master's degree at Lehigh University. Christopher currently works for General Electric Environmental Services in Santa Ana, California.

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