Fluidized bed combustion shows potential for burning high sulfur coal and other low grade fuels efficiently and clearly. In order to achieve higher combustion efficiency in the current fluidized bed combustor at OARDC, refuse was reintroduced into the burner with new fuel under step recycling operation. Results of these tests showed that combustion efficiency on the carbon was around 85% in the fuel and only 30-70% in the recycle refuse. These results predict that the recycle method can improve the combustion efficiency under continuous recycling operation, but would require a recycle ratio of 1.7 to achieve 99% combustion efficiency. In general, the recycle mode can reduce the pollutants emission, but the pollutants emission, S0₂ and NOx, are still higher than the limits of the EPA.
APPLYING A RECYCLE MODE TO A
FLUIDIZED BED COMBUSTOR

A Thesis
Presented in Partial Fulfillment of the Requirements for
the Degree Master of Science in the
Graduate School of the Ohio State University

by
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* * * * *
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1988

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Energy Conversion
Thermal Analysis
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# LIST OF SYMBOLS

- **A**: area, $m^2$
- **$A_c$**: cross-sectional area, $m^2$
- **$C_p$**: specific heat at constant pressure, $J/kg*K$
- **D**: diameter, m
- **$D_H$**: hydraulic diameter, m
- **E**: emissivity
- **g**: gravitational acceleration, $m/s^2$
- **h**: convection heat transfer coefficient, $W/m^2*K$
- **K**: thermal conductivity, $W/m*K$
- **L**: characteristic length, m
- **M**: mass flow rate, $kg/hr$
- **m**: mass flow rate, $kg/s$
- **Nu**: Nusselt number
- **P**: static pressure, $kg/m^2$
- **Pr**: Prandtl number
- **Q**: mass flow rate, $kg/s$
- **q**: heat transfer rate, $W$
- **R**: thermal resistance, $K/W$
- **Ra**: Rayleigh number
- **Re**: Reynolds number
- **T**: temperature, K
- **V**: cylinder radius, m
- **W**: side length of square, $m$
- **$\alpha$**: thermal diffusivity, $m^2/s$
- **$\beta$**: volumetric thermal expansion coefficient, $K^{-1}$
- **$\mu$**: viscosity, $kg/s*m$
- **$\nu$**: kinematic viscosity, $m^2/s$
- **$\rho$**: fluid mass density, $kg/m^3$
- **$\sigma$**: Stefan-Boltzmann constant

## Subscripts
- **cond**: conduction
- **conv**: convection
- **rad**: radiation
- **s**: surface condition
- **$\infty$**: freestream condition
CHAPTER I
INTRODUCTION

Since the energy crisis of the 1970's, developing methods to convert non-petroleum sources to useful energy has become one of the most important world-wide tasks. Since there are significantly large reserves of coal and its cost is low compared with oils and natural gas, many researchers have been studying how to convert its energy in an environmentally safe manner. The basic concept about the processes and techniques of coal conversion is the transmutation of coal into acceptable energy. There are a lot of methods to accomplish this energy conversion. Among the methods, fluidized bed combustion is a good way for getting energy from coal. The reasons are as follows:

1. It enhances fuel flexibility by allowing a variety of low grade fuels with high moisture or ash content to be burned successfully.

2. It can reduce SO$_2$ and NOx emission by controlling the combustion process, and eliminating the need for flue gas treatment.

3. It can prevent ash clinging by controlling the bed temperature below the ash softening temperature.
These advantages do not mean that any given fluidized bed combustor (FBC) can burn any fuel without problems. The FBC must be properly designed to adjust for the different fuels.

Researchers at OARDC-OSU have started to develop a commercial burner system (Patent No. 4671251) using fluidized bed technology. Tests in 1986-1987 by Keener and Henry showed a significant amount (7-19%) of fuel carbon in the refuse. Since burning fuels completely should improve the burner performance, a series of tests have been conducted with recycling the refuse to study the effect of recycling on the performance of burner system at OARDC.
CHAPTER II
LITERATURE REVIEW

The term "fluidization" came into wide use in the petroleum industry in the 1940's. The technique of fluidization enlarges the particle surface area exposed to the fluid which is advantageous since the combustion rate of solid fuel is proportional to solid surface area. Although the combustion of coal and other solid fuels is more difficult than the combustion of liquid and gaseous fuels, solid fuels can be handled in the fluidized state and are easy to burn in a FBC. Therefore, the FBC is an ingenious and low cost method for obtaining energy in an environmentally acceptable manner.

There are two general classifications of fluidized bed combustors: atmospheric and pressurized. The atmospheric FBC operates at or very near atmospheric pressure, while the pressurized FBC operates at a pressure of 3 to 10 atmospheres. The performance of the pressurized FBC is generally higher than that of the atmospheric FBC (more efficient sulphur capture and a higher permissible operating temperature), but the pressurized FBC has more technical difficulties. Therefore, the development of atmospheric FBCs
is much more advanced.

2.1 THE PERFORMANCE OF FLUIDIZED BED COMBUSTOR

In FBCs, the fluidized media serves as a large heat sink and/or source preventing rapid fluctuations in bed temperature. This allows fuels to be combusted at a steady temperature. Therefore, combustion can be controlled easier in a FBC than in a conventional furnace. Because of this characteristic, FBCs have been found to work well with biomass material.

Keener, Henry, and Anderson [2] conducted experiments on the combustion of corncobs in the OARDC-FBC in 1982. They indicated a trend that burner efficiency declined as excess air increased. This phenomenon results from the fact that excess combustion air carries more heat away with it.

Ash remains solid and is carried out of the burner with the exhaust air after combustion. This kind of ash is a fine particulate matter which is collected by a cyclone. Rickman [6], in order to burn the fuels completely, reintroduced the ash into the burner with new fuels. He applied this recycle technology to a FBC and obtained a relationship between recycle rate and combustion efficiency. According to his findings, a recycle rate three times the coal feed rate had 99% overall combustion efficiency.

Combustible loss is estimated from the carbon loss in the CO in the exhaust gas and the elutriated unburned
carbon. Studies of combustion engineering [20] on combustible loss showed no significant relationship between bed temperature and combustible loss. Thus, combustion efficiency has weak and possibly insignificant temperature dependence. Generally, recycling the refuse can increase the combustion efficiency 3 to 4%. The value of combustion efficiency without recycling is about 92 percent whereas that for recycling is about 95 percent (Joseph G. Singer [20]). However, the value of recycle ratio used to obtain the above results was not specified.

2.2 THE EFFECT OF FUEL PARTICLE SIZE

Schonauer et al. [3] studied the effect of particle size and moisture content on FBC and found that to keep the bottom of the OARDC-FBC burning was very difficult with very coarse and/or fine corn. Their conclusions were as follows: the best particle size distribution for this burner was #9 rollermilled corn (Table 1), and dried corn could be burned more stably than wet corn. However, their results looked only at burner operation stability and efficiency, and did not quantify combustion efficiency.

The findings of a study on the effect of particle size on FBC by D'Amore, Donsi, and Massimilla [4] using coal showed that the effect of particle size decreased as the reactivity of fuel decreased. Reactivity of fuel is defined as the rate that fuel combines with oxygen at the
temperature above the ignition point.

Tatebayashi, Okada, and Ikeda [11] used four groups of coal particles (smaller than 0.25 mm, 0.25 to 0.5 mm, 0.5 to 1.0 mm, and 1.0 to 2.0 mm) to investigate the effect of particle size. They found that the combustion efficiency increased as particle size was increased and NOx emission was greatest when coal particles were between 0.25 to 0.5 mm. Their NOx emission level was around 250 PPM when combustion occurred at 850°C and an excess air level of 20%.

Table 1. Particle Size of #9 Rollermilled Corn

<table>
<thead>
<tr>
<th>Sieve Mesh per Inch</th>
<th>Weight Fraction (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>on top of 4</td>
<td>7.5</td>
</tr>
<tr>
<td>4-5</td>
<td>19.9</td>
</tr>
<tr>
<td>6-10</td>
<td>39.7</td>
</tr>
<tr>
<td>10-14</td>
<td>13.3</td>
</tr>
<tr>
<td>14-20</td>
<td>8.5</td>
</tr>
<tr>
<td>20-30</td>
<td>4.2</td>
</tr>
<tr>
<td>through 30</td>
<td>6.9</td>
</tr>
<tr>
<td>Bulk Density (kg/m³)</td>
<td>527</td>
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</table>

2.3 POLLUTION CONTROL

Oxides of nitrogen and sulphur dioxide are the most offensive pollutants produced by the combustion processes. Current standards have been set by EPA. The standards are as follows: NOx emission from boilers larger than 100 MW can not exceed 370 PPM, and units smaller than 100 MW can not exceed 430 PPM; the limit of SO₂ and NOx emissions are 1.2
and 0.6 lb/10^6 Btu respectively.

NOx begins chiefly as nitric oxide (NO). This later gradually combines with hydrocarbons and ozone in sunlight to produce smog and compounds that may irritate the eyes, aggravate certain respiratory diseases, and injure plants. NOx is derived from two sources: (1) NOx from the nitrogen of fuel, which is called fuel NOx. (2) NOx from nitrogen of combustion air, which is called thermal NOx.

Three technologies [17] for reducing NOx emission are:

1. Firing with low excess air. The practical level of excess air is typically down to 15%. The reason for this value is that decreasing the combustion air can increase CO concentration. There is thus a tradeoff between NOx reduction and an increase in other emissions.

2. Flue gas recirculation. This method reduces thermal NOx formation by lowering the bulk flame temperature. NOx-reduction levels of 20-50% have been achieved with 20-40% recirculation. Since this method just inhibits thermal NOx, it is most useful for natural gas fired burner. When firing coal, it is necessary to filter particulate matter from the recirculation gas in order to avoid the burner plugging and eroding.

3. Selective catalytic reduction. This method is based on the preference of NOx to react with ammonia (NH3), rather than with other flue gas constituents. NH3 typically is injected into the flue gas and reacts with NOx to form
nitrogen gas and water vapor. The chemical reactions are as follows:

\[
\begin{align*}
4\text{NH}_3 + 4\text{NO} + \text{O}_2 & \rightarrow 4\text{N}_2 + 6\text{H}_2\text{O} \\
4\text{NH}_3 + 2\text{NO}_2 + \text{O}_2 & \rightarrow 3\text{N}_2 + 6\text{H}_2\text{O}
\end{align*}
\]

The variables that affect the degree of NOx removal by this method are the reaction temperature, the flue gas flow rate, and the mole ratio of NH3 to NOx.

Sulphur removal can be improved by three methods [ref 18. p. IV-29]:

1. Increase excess air.
2. Increase average residence time of particles.
3. Recycle the refuse from the cyclone.

Another technique which can remove most of the SO2 during/after combustion is calcium-based sulphur removal. In this process limestone is reacted with sulphur dioxide to form solid CaSO4 as the coal is burned and thus prevent its release to the environment as gaseous sulphur dioxide. The chemical reaction is as follows:

\[
2\text{CaO} + 2\text{SO}_2 + \text{O}_2 \rightarrow 2\text{CaSO}_4
\]

Coal power generation plants usually produce the pollutants of NOx and SO2 emission. In order to reduce pollution, it is necessary to apply some advanced technologies. Therefore, the cost of de-NOx and de-SO2 should be considered in the total power generation cost. According to the findings of Tatebayashi, Okada, and Ikeda [11], the facility costs for purifying combustion were about 20% of the total power generation cost. The effects of various parameters on NOx and SO2 emission in FBCs were as
follows: as the excess air became smaller, NOx emission simply dropped; NOx emission increased as the bed temperature rose; and de-SO₂ efficiency reached maximum level at a bed temperature of 800–830°C; then de-SO₂ efficiency suddenly dropped as the temperature exceeded 850°C.
CHAPTER III
OBJECTIVES

The review of literature indicates that the performance of a FBC can be improved if the effects of various variables are thoroughly understood. The main purpose of this study was to investigate the effect of recycling refuse on the combustion and burner efficiency of the fluidized bed combustor at CARDC in order to evaluate the advantages of installing recycle equipment. In addition, since environmental regulations are becoming more stringent, it is necessary to determine the pollutant emission of this system. Therefore, the objectives of this research were to determine by experimentation:

1. The effect of a recycle mode on combustion and burner efficiency.

2. The effect of fuel particle size on combustion efficiency.

3. The pollution associated with combustion in the FBC.

Although heat losses from the burner body do not lower combustion efficiency, they do lower burner efficiency. Thus a fourth objective of this research was to develop a mathematical model to predict steady state heat losses from
the burner body and thermocouples and compare those predictions with the experimental results.
CHAPTER IV
EXPERIMENTAL EQUIPMENT AND PROCEDURE

The burner performance, with and without recycling the refuse, and fuel particle size effect were determined by experimentation. The experimental equipment and procedure are described in this chapter.

4.1 EXPERIMENTAL EQUIPMENT

Experiments were carried out in the Agricultural Engineering Department, OARDC, Wooster, Ohio. The equipment used in this study is listed as follows:

1. Atmospheric fluidized bed combustor. The burner system is the second generation OARDC-FBC designed by Robert J. Anderson, Harold M. Keener, and James E. Henry.

2. Fuel feed system. A 0.142 m³ tapered hopper (patent pending), equipped with a metering device, is used for feeding powdered coal. Fuel feed rate is controlled by an SCR motor speed controller. A stirrer and a vibrator are installed in the hopper in order to ensure continuous operation. The fuel is fed pneumatically into the bottom of burner.
3. Ash collection system. The ash collection system is made up of a cyclone and a bag filter. The cyclone achieves particulate removal by centrifugal, inertial, and gravitational forces developed in a vortex separator. These particles fall into a storage container that can be manually emptied when the burner is off. The bag filter, down stream from the cyclone, is designed to be easily removed in order to calculate the ash collected in the bag filter.

4. Gas analyzers. Gas analyzers were used to measure the mass flow of several constituents in the exhaust gas. Since only the volume ratio can be measured by using a gas analyzer, it was necessary to do some calculations to convert volume ratio to mass flow. A model 300 gas analyzer of Mine Safety Application Company in Pittsburgh, Pennsylvania, was used to measure CO, SO₂, and CO₂, and a model 400 was used to measure O₂; the accuracy of model 300 is ± 1% of full scale, and ± 2% of full scale for model 400. A model 10 gas analyzer of Thermo Electron Instrument in Hopkinton, Mass, was used to measure NO and NOₓ; the accuracy of this NOₓ gas analyzer is ± 1% of full scale. In addition, there were several calibration gases, N₂ (100%), O₂ (10%), CO₂ (15%), CO (0.99%), SO₂ (0.2%), and NO (805 PPM), which were used in setting the zero and span for each gas analyzer. During the experiments, calibration of the gas analyzers were performed every week to minimize errors in measurement due to instrument drift.
5. Instruments for measuring air mass flow. Thin-plate orifices with flange taps were used to measure both the combustion and cooling air flow. Total pressure, pressure difference, and air temperature were measured and used to calculate mass flow of combustion air and cooling air.

6. Control system. A electronic control system was used to automatically control the FBC. For these tests, the fuel feed system was shut down when the bed temperature went above 850°C and turned on below 845°C, and the cooling air pump turned on when the bed temperature was above 820°C.

7. Thermocouple. Type K thermocouples were used to measure the burner temperatures at different heights, inlet and exit temperatures of combustion air, and cooling air at different heights of heat exchanger. The accuracy of a thermocouple is ± 2.22°C between 0–277°C and ± 0.75 percent between 277–1260°C. For example, at 800°C the maximum error is ± 6°C.

8. Instruments for measuring water vapor in the air. A dew point hygrometer, model 880 of EG&G, Walthem, New Hampshire, was used to measure the dew point temperature and a barometer was used to measure the atmospheric pressure in order to calculate the water vapor in the ambient air.

9. Data collection system. The data measured from the above instruments were transduced to a digital data collector, model 8000 of KAYE Instrument, Bedford, Mass. That equipment was connected to a magnetic tape recorder in
order to record the experimental data on a cassette tape for off line data analysis.

The prototype burner at the OARDC is shown in Figure 1.

4.2 EXPERIMENTAL PLAN

The first set of experiments were planned to examine the effect of excess air and refuse recycle ratio on the burner performance. Since the OARDC-FBC has no recycle facility, it is necessary to remove the refuse from the burner ash container and mix it with new coal, and then put the mixed fuel into the hopper for the recycle mode test. For the recycle tests, three different recycle ratios of 0.05, 0.10, and 0.15 were to be used. The recycle ratio is defined as the ratio of refuse rate to pure coal rate [ref 18. p.IV-19].

Before running the recycle test, the test without recycling refuse was conducted first. Then the recycle tests were conducted by using the refuse produced from the previous test. With three recycle ratios, a total of nine tests for one fuel rate setting value was required. Table 2 shows the amount of tests needed for each fuel rate setting. In order to investigate the effect of excess air, five different fuel rate settings were planned with each setting being tested twice. Thus a total of ninety tests would be needed for this study.
Figure 1. The Fluidized Bed Combustor at OARDC
Table 2. Experimental Plan for Recycle Test

<table>
<thead>
<tr>
<th>Recycle Ratio</th>
<th>0.05</th>
<th>0.10</th>
<th>0.15</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of Step Recycle</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
<tr>
<td>2</td>
<td>X</td>
<td>X</td>
<td>X</td>
</tr>
</tbody>
</table>

A discussion with Harold Keener revealed that one step recycle test should be enough to evaluate the recycle effect. Therefore, the second step recycle test was omitted, reducing the number of tests needed in this study to sixty.

A second set of experiments were conducted to determine the effect of fuel particle size on combustion efficiency. Coal was classified into three particle size distributions: fine, medium, and coarse coal. Each particle size was subjected to a combustion test without recycling refuse three times at the same fuel rate setting value. A total of nine tests was required.

4.3 EXPERIMENTAL PROCEDURE

The procedures to start the burner system are as follows:

The first step is to heat up the system using an electric heater. Usually, the bed temperature can reach 510°C after three hours. Then shut off the electric heater and turn on the fuel feed system to supply corncobs as fuel into the burner chamber. About twenty minutes later, the
burner should reach about 750°C. At that temperature, the fuel being fed need be manually switched to coal (Note: There are two feed hoppers). The reason for burning corncobs first is that corncobs have a lower combustion temperature and the bed temperature could be more easily elevated to the combustion temperature of coal by burning corncobs. The burner system should be kept operating about one hour in order for the system to reach pseudo-steady state.

The normal experimental procedures are as follows:

1. Shut down the burner and remove the refuse out of the burner by using an electric drill to rotate a screw auger in the bottom of refuse container.

2. Before starting the test, suck the remaining fuel out of the hopper, add the new fuel (new coal or mixed fuel with a certain recycle ratio) into the hopper, and record the weight of hopper (including fuel).

3. Start the burner and run the test for 45 minutes. During the test period, the experimental data of pressure, temperature, and gas volume ratios are recorded by the data collection system every two minutes. The ambient conditions of dew point temperature and atmospheric pressure are recorded manually.

4. Shut down the burner and record the weight of the hopper to calculate the actual fuel rate. Then remove the refuse out of the burner and take two samples to analyze the contents for unburned carbon. Finally, mix the new coal and
the refuse at a certain recycle ratio.

In order to get a similar excess air value for test with and without recycling the refuse, the fuel rate setting value was increased one scale for the recycle test. The reason for this adjustment is that the combustible material in the recycle fuel is less than that in pure coal for the same mass fuel rate.

For the experiments on the effect of fuel particle size, the first thing was to screen the coal by various meshes. Figure 2 shows the particle size distribution of a sample of coal. Three groups of particles, fine coal (diameter < 0.149mm), coarse coal (diameter > 0.149mm), and original coal were used in the tests. Due to time constraints in this study, only two tests were repeated under the same conditions.
Figure 2. Coal Particle Size Distribution
CHAPTER V

EXPERIMENTAL EVALUATION

The methods and equations for experimental data analysis are described in this chapter.

5.1 DATA ANALYSIS

The constituents entering the burner include the air (water, \( O_2 \), and \( N_2 \)) and the fuel (\( C, N_2, S, \) ash,...). The products leaving include \( CO_2, CO, N_2, SO_2, O_2, NOx, \) ash, and water. The ultimate analysis of pure coal, which was analyzed by Warner Laboratory, Inc., Cresson, Pennsylvania, is listed in Table 3.

Table 3. Ultimate Analysis of Coal

<table>
<thead>
<tr>
<th>Element</th>
<th>As Received</th>
<th>Dry Basis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moisture</td>
<td>4.37%</td>
<td>7.41%</td>
</tr>
<tr>
<td>Ash</td>
<td>7.09%</td>
<td>73.63%</td>
</tr>
<tr>
<td>Sulfur</td>
<td>2.98%</td>
<td>3.11%</td>
</tr>
<tr>
<td>Carbon</td>
<td>70.42%</td>
<td></td>
</tr>
<tr>
<td>Hydrogen</td>
<td>5.20%</td>
<td>5.43%</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>1.42%</td>
<td>1.49%</td>
</tr>
<tr>
<td>Oxygen</td>
<td>8.52%</td>
<td>8.92%</td>
</tr>
</tbody>
</table>

| HHV (Btu/lb) | 12876 | 13464 |

21
The combustible constituents of fuels are carbon, hydrogen, and sulphur and their compounds. The complete combustion reactions under considerations were:

\[
C + O_2 \rightarrow CO_2 \\
2H_2 + O_2 \rightarrow 2H_2O \\
S + O_2 \rightarrow SO_2
\]

The above reactions were used to determine the stoichiometric air which is defined as the quantity of air required to burn a unit weight of fuel completely without any free oxygen appearing in the products of combustion.

\[
\text{Supplied air} = \frac{\text{Excess air}}{} \times 100\%
\]

The air flow through the orifice is represented in Figure 3. The mass flow of air can be determined by the following equation.

\[
Q = \frac{A_2}{\sqrt{1-(A_2/A_1)^2}} \sqrt{2gS(P_1-P_2)}
\]

The orifice used to calculate air mass flow in the OARDC-FBC was derived from an available technical reference [19]. Values used were contained in the program FBCSI in Appendix B. For analysis, it was assumed that there was no air temperature change at section 1 and 2, the flow was adiabatic, and no mechanical work was transmitted through the orifice system.
5.2 The Calculation of Combustion Efficiency

It is necessary to determine the dry refuse after combustion in order to calculate the mass flow of exhaust gases and combustion efficiency.

\[
\text{Exhaust gases rate} = \text{Fuel rate} + \text{Combustion air rate} - \text{Refuse rate}
\]

\[
\text{Fuel HHV} - \text{Unburned Carbon HHV} - \text{CO HHV}
\]

\[
\text{Comb. eff} = \frac{\text{Fuel HHV}}{\text{Fuel HHV}}
\]

where HHV = high heat value (kcal/kg)

The method of removal of the dry refuse from the cyclone and analysis of the amount of carbon per unit weight of refuse can be used to determine the value of combustion efficiency. However, two factors influence the accuracy of removing the dry refuse from the cyclone: some elutriated sand may be included in the refuse and the refuse may hang
up in the cyclone and ash container. Therefore, removing refuse to determine the refuse rate can lead to large errors. From the 1946 ASME Test Code for Steam Generating Units, the weight of dry refuse may be determined by the coal and refuse analysis when it is impractical to weigh the total refuse. Since

\[
\text{Refuse} \times \frac{\text{Ash}}{\text{Coal}} = \frac{\text{Ash}}{\text{Coal}}
\]

\[A = \frac{W_r}{1-C_r}\]

where \(W_r\) = dry refuse per kg coal as fired
\(A\) = ash in coal
\(C_r\) = combustible per kg refuse

The value of \(C_r\) can be determined by putting a certain amount of refuse in an oven and keeping it at 700\(^\circ\)C for about 18 hours in order to combust all the combustible in refuse. The unburned carbon in the refuse may be calculated by assuming that all the combustible in the refuse is carbon.

5.3 ENERGY BALANCE OF THE BURNER SYSTEM

The energy balance of the burner system is shown in Figure 4. The reference temperature (Tref) used in this study was 25\(^\circ\)C, and Ti was the temperature of fuel and air into the system.
The burner efficiency is defined as the useful energy out of the system divided by the total energy into the system. In this system, the burner efficiency is the ratio of $H_3$ to $H_1$.

\[ \text{Figure 4. Energy Balance of the Burner System} \]

\[ H_1 = H_2 + H_3 + \text{OTHER} \]

\[ H_1 = \text{coal HHV} \times \text{dry coal mass flow} \]
\[ + \text{Cp of coal } \times (\text{Ti-Tref}) \times \text{dry coal mass flow} \]
\[ + \text{Cp of } H_2O \times (\text{Ti-Tref}) \times H_2O \text{ mass flow in coal} \]
\[ + \text{Cp of air } \times (\text{Ti-Tref}) \times \text{comb. air mass flow} \]
\[ + \text{Cp of } H_2O \text{ steam } \times (\text{Ti-Tref}) \times \text{water in comb. air} \]
\[ + \text{Cp of air } \times (\text{Ti-Tref}) \times \text{cooling air mass flow} \]
\[ + \text{Cp of } H_2O \text{ steam } \times (\text{Ti-Tref}) \times \text{water in cooling air} \]
\[ - \text{latent heat of } H_2O \times (\text{water mass flow in coal+water from combustion product}) \]

Since the heat capacity of gas is dependent on temperature, it is necessary to find the relationship between heat capacity and temperature first. Then integration should be used to calculate the energy carried out with exhaust gases and cooling air. The heat capacity
equation is generally of the form of a power series with absolute temperature, T, as the independent variable [15].

\[ \text{C}_p = a + (b \times 10^{-3}) T + (c \times 10^{-6}) T^2 + (d \times 10^5 T)^{-2} \]

The constants a, b, c, and d of the gases appearing in the system are listed in Table 4.

<table>
<thead>
<tr>
<th>GAS</th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>d (cal/gmole·K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>N(_2)</td>
<td>6.76</td>
<td>0.606</td>
<td>0.13</td>
<td></td>
</tr>
<tr>
<td>C(_2)</td>
<td>8.27</td>
<td>0.258</td>
<td></td>
<td>-1.877</td>
</tr>
<tr>
<td>CO(_2)</td>
<td>7.7</td>
<td>5.3</td>
<td>-0.83</td>
<td></td>
</tr>
<tr>
<td>H(_2)O</td>
<td>11.2</td>
<td>7.17</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Ar</td>
<td>4.953</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SO(_2)</td>
<td>11.4</td>
<td>1.414</td>
<td></td>
<td>-2.045</td>
</tr>
<tr>
<td>C(_O)</td>
<td>6.6</td>
<td>1.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>NO</td>
<td>7.133</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>NO(_2)</td>
<td>8.89</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

(Weast and Astle, 1980)

Since each test lasted 45 minutes, there were at least 22 data points for each test. From experimental data, the test usually reached pseudo-steady state after 20 minutes. Two samples of experimental data are shown in Figure 5 and 6. A time-averaging program developed by Keener was used to calculate the inlet and exit temperature of combustion air as well as cooling air. During the operation of this system, the combustion air was on all the time and the mass flow rate was almost constant (± 2%). In addition, the mass flow rate of the cooling air was either on (at a nearly constant value) or off.
To calculate the actual cooling air through the heat exchanger for these tests, the percentage of time that the cooling air pump was on was analyzed first. Then the actual cooling air rate at steady state was determined by multiplying the cooling air rate and the time percentage.

\[ H_2 \text{ or } H_3 = \sum_{i=1}^{n} \frac{M_i}{298} \int_{298}^{T_0} C_{pi} \, dT \]

\[ = \sum_{i=1}^{n} M_i \left[ \frac{1}{2} (T_0 - 298) + \frac{1}{2} (b \times 10^{-3}) (T_0^2 - 298^2) + \frac{1}{3} (c \times 10^{-6}) (T_0^3 - 298^3) - (d \times 10^5) \right] \]

where \( M_i \) = the computed mass flow rate of each gas
\( i \) = constituent of gas
\( n \) = number of gas
\( T_0 \) = the exit temperature of air computed by time-averaging program

The other method to calculate \( H_2 \) and \( H_3 \) is to calculate the average value of \( H_2 \) and \( H_3 \) at every data point.

\[ H_2 \text{ or } H_3 = \left( \sum_{j=1}^{m} \sum_{i=1}^{n} M_{ij} \right) \left( \frac{T_{0j}}{298} \right) \frac{C_{pi} \, dT}{m} \]

where \( M_{ij} \) = the measured mass flow rate of each gas
\( i \) = constituent of gas
\( j \) = data point
\( m \) = number of data point
\( T_{0j} \) = the measured exit temperature of air
\( j \) = data point

The difference between these two methods was compared for one case and was less than 1%. The first method was used in this study.

5.4 COMPUTER PROGRAM FOR EXPERIMENTAL DATA ANALYSIS

The data was transferred from the magnetic tape to a computer disc file. A time-averaging program developed by
Keener, for previous work, was then used to calculate the temperature and pressure at the pseudo-steady state. A computer program for the system energy balance, also previously developed by Keener, was modified and used. The program was modified to be able to calculate burner performance with a recycle mode. In addition, the units in the program were changed from English to metric.

The first part of the program allows inputting the experimental parameters. Then those values are used to calculate the mass flow rates of cooling air, combustion air, and exhaust air. Finally, the energy conservation principle is applied to compute the burner efficiency. Two output samples of the computer program are shown in Appendix C.

5.5 \textbf{HEAT LOSS FROM THE BURNER BODY}

Heat losses through the burner body were calculated from "other" heat losses by subtracting out the unburned carbon high heat value and the CO high heat value.

A computer program, based on steady state conditions, was developed in this study (Appendix D) for the OARDC burner system. The system was assumed to consist of two cylindrical shells (one for the burner chamber and the other for the cyclone). Results of that analysis suggested as cooling air rate increased from 0 to 120 kg/hr, heat loss would drop only 13 watts ($\approx 1.2\%$).
CHAPTER VI

DISCUSSIONS FOR COMBUSTION EFFICIENCY

Experimental results to investigate the effect of the recycle mode on combustion efficiency, under step recycling operation, are presented in this chapter. In addition, the results of tests are used to predict the combustion efficiency under continuous recycling operation for different recycle ratios.

6.1 STEP RECYCLING OPERATION

Results of combustion studies showed that combustion efficiency increased as excess air increased. This is a reasonable result because the fuel has more opportunity to react with the air when providing more combustion air. Figures 7, 8, 9, and 10 show data points on the plane of combustion efficiency and excess air axis.

A regression analysis was conducted for different recycle ratios based on excess air between 0-75%. Although the regression curves were expected to be similar to a logarithm curve [ref 20 p.I-37], quadratic relationships of combustion efficiency and excess air were found within this excess air range. The equations were
Figure 7. Relationship of Excess Air and Combustion Efficiency. Recycle Ratio = 0.00
Figure 8. Relationship of Excess Air and Combustion Efficiency. Recycle Ratio = 0.05
Figure 9. Relationship of Excess Air and Combustion Efficiency. Recycle Ratio = 0.10
Figure 10. Relationship of Excess Air and Combustion Efficiency, Recycle Ratio = 0.15.
Recycle Ratio = 0.00
EFFCOMB = 86.9 + 0.106*EXAIR - 0.000131*EXAIR^2

Recycle Ratio = 0.05
EFFCOMB = 86.9 + 0.0787*EXAIR - 0.000156*EXAIR^2

Recycle Ratio = 0.10
EFFCOMB = 84.2 + 0.126*EXAIR - 0.000234*EXAIR^2

Recycle Ratio = 0.15
EFFCOMB = 81.1 + 0.245*EXAIR - 0.00208*EXAIR^2

where EFFCOMB = combustion efficiency
EXAIR = the value of excess air

The R^2 values of the above equations were 82.2%, 91.7%, 89.9%, and 91.5% respectively. Because sand in the burner was gradually lost with the exhaust air, some variables could not be controlled at the same values for each test. These included bed temperature and bed pressure. This may account for some of the combustion efficiency difference within a small excess air range.

Figure 11 shows the regression curves for different recycle ratios. It appears that combustion efficiency decreased as the recycle ratio increased. This result is opposite to the fact that the recycle method can improve the combustion efficiency. The reason for this result is that the tests were under step recycling operation instead of continuous recycling operation. Although the tests were not conducted under continuous recycling operation, the results of tests under continuous recycling operation can be predicted by the results of these tests.
6.2 CONTINUOUS RECYCLING OPERATION

In order to evaluate the effect of recycle mode on combustion efficiency under continuous recycling operation, a block diagram (Figure 12) was constructed by assuming that the combustion efficiency of carbon in coal was the same for tests with and without recycling refuse, the mass flow rate of carbon monoxide was negligible, and the hydrogen and the sulfur were combusted completely.

The combustion efficiency versus recycle ratio can be predicted by solving the following equations based on mass balance.

\[ M_2(1-f_c) = M_{\text{coal}} \times \text{ASH} \]

\[ M_1 = M_2 + M_3 \]

\[ M_1 = M_{\text{coal}} \times [\text{ASH} + \text{CBN}(1-\eta_{C,\text{coal}})] + M_3 \times [f_c \times (1-\eta_{C,\text{ash}}) + (1-f_c)] \]

\[ R = \frac{M_3}{M_{\text{coal}}} \]

where

- ASH = ash content in coal
- CBN = carbon content in coal
- \( f_c \) = carbon content in refuse
- \( M_1 \) = mass flow rate of refuse into the ash tank
- \( M_2 \) = mass flow rate of refuse out the ash tank
- \( M_3 \) = mass flow rate of recycle fuel into the burner
- \( R \) = recycle ratio
- \( M_{\text{coal}} \) = mass flow rate of new coal into the burner
- \( \eta_{C,\text{coal}} \) = combustion efficiency of carbon in coal
- \( \eta_{C,\text{ash}} \) = combustion efficiency of carbon in refuse

The variables \( M_1 \), \( M_2 \), \( M_3 \), and \( f_c \) in above equations are unknown; however, the variables \( \eta_{C,\text{coal}} \) and \( \eta_{C,\text{ash}} \) which are a function of excess air can be determined from the results of tests under step recycling operation.
Figure 12. Block Diagram of Continuous Recycling Operation
From the above equations, the value of $f_C$ can be solved in terms of ASH, CBN, $\eta_{C,coal}$, $\eta_{C,ash}$, and $R$.

$$f_C = \frac{-B-\sqrt{B^2-4*A*C}}{2*A}$$  \hspace{1cm} (6.1)

where $A = R$ $\eta_{C,ash}$

$B = CBN*\eta_{C,coal} - R*\eta_{C,ash} - ASH - CBN$

$C = CBN - CBN*\eta_{C,coal}$

From the value of $f_C$, the combustion efficiency under continuous operation can be calculated. Since

$$\text{Comb. eff} = 1 - \frac{\text{HHV}_C*f_C*M_2}{\text{HHV}_{coal}*M_{coal}}$$

$$\text{Comb. eff} = 1 - \frac{\text{HHV}_C*f_C*ASH}{\text{HHV}_{coal}*(1-f_C)}$$  \hspace{1cm} (6.2)

In order to investigate the relationship of combustion efficiency and recycle ratio, the values of $\eta_{C,coal}$ and $\eta_{C,ash}$ should be calculated first.

The combustion efficiency of carbon in the coal can be calculated from the tests without recycling refuse by assuming that the hydrogen and sulfur are combusted completely.

$$\eta = \frac{\eta_{C,coal}*\text{HHV}_C*M_{C,coal} + \text{HHV}_H*M_{H,coal} + \text{HHV}_S*M_{S,coal}}{\text{HHV}_{coal}*M_{coal}}$$

$$\eta_{C,coal} = \frac{\eta*\text{HHV}_{coal}*M_{coal} - \text{HHV}_H*M_{H,coal} - \text{HHV}_S*M_{S,coal}}{\text{HHV}_C*M_{C,coal}}$$

where $\eta$ = combustion efficiency of test without recycling refuse

The experimental results and regression curve of combustion efficiency of carbon in coal are shown in Figure
13. The combustion efficiency of carbon in ash can be calculated from the tests with recycling refuse by assuming that the value of $\eta_{c,\text{coal}}$ is the same as the value of that in the test without recycling refuse.

$$\eta = \frac{\eta_{c,\text{coal}} + \eta_{c,\text{ash}}}{\frac{\text{HHV}_c + \text{HHV}_{\text{H},\text{coal}} + \text{HHV}_{\text{s},\text{coal}}}{\text{HHV}_c + \text{HHV}_{\text{coal}}}}$$

$$\eta_{c,\text{ash}} = \frac{\eta^* (\text{HHV}_c + \text{HHV}_{\text{H},\text{coal}} + \text{HHV}_{\text{s},\text{coal}})}{\text{HHV}_c}$$

where

- $\eta$ = combustion efficiency of test with recycling refuse

The experimental results of combustion efficiency of carbon in ash are shown in Figures 14, 15, and 16. It is hard to determine the value of combustion efficiency of carbon in ash because that value is widely scattered. From the above equation, the value of $\eta_{c,\text{ash}}$ is very sensitive to the value of combustion efficiency of test with recycling refuse. A smaller recycle ratio is more sensitive since the denominator is smaller. From the results, generally the value of combustion efficiency of carbon in ash is within the range of 0.3-0.7 and the average values are around 0.5.
Figure 13. Combustion Efficiency of carbon in Coal

Regression Curve:
Y = 82.5 + 0.14x + 0.000176x^2
Figure 15. Combustion Efficiency of Carbon in Ash
Recycle Ratio = 0.10
Figure 16. Combustion Efficiency of Carbon in Ash
Recycle Ratio = 0.15
In order to evaluate the effect of $\eta_{c,\text{ash}}$ on the combustion efficiency under continuous recycling operation, a set of numerical values (ASH=0.0741, CBN=0.7363, $\eta_{c,\text{coal}}=0.8525$, HHV$_c$=7823.1 kcal/kg, HHV$_{\text{coal}}$=7480 kcal/kg) were substituted into equations (6.1) and (6.2). From the results (Figure 17), higher value of $\eta_{c,\text{ash}}$ has higher combustion efficiency for a certain recycle ratio. The combustion efficiency can achieve 99% when the value of $\eta_{c,\text{ash}}$ is 0.5 for a recycle ratio of 1.7.
Figure 17. Combustion Efficiency under Continuous Recycling Operation
CHAPTER VII
RESULTS AND DISCUSSIONS

The results of experimentation to investigate the effect of the recycle mode on burner efficiency, sulphur dioxide emission, and nitrogen oxides emission are presented in this chapter. In addition, the effect of fuel particle size on combustion efficiency is also described.

6.1 BURNER EFFICIENCY

One group of experimental data were used to evaluate the effect of excess air and recycle ratio on burner efficiency. There should be an optimum excess air value for maximum burner efficiency from the experimental data of tests without recycling refuse because as the excess air increased, burner efficiency increased at first then it decreased. This trend is the same as the result of reference [18 p.I-37]. The burner efficiency increased as excess air increased at the low excess air range because combustion was more complete; burner efficiency decreased as excess air increased at the high excess air range because more heat was carried away with exhaust gases. However, the burner efficiency decreased strictly at excess air between 0-75%
when the recycle ratio was greater than 0.10. Figures 18, 19, 20, and 21 show the data points on the plane of burner efficiency and excess air axis.

A regression analysis was conducted for different recycle ratios based on the excess air between 0-75%. It appeared that a quadratic relationships described burner efficiency as a function of excess air. The equations were

Recycle Ratio = 0.00

\[
\text{EFFBURN} = 53.1 + 0.197 \times \text{EXAIR} - 0.00529 \times \text{EXAIR}^2
\]

Recycle Ratio = 0.05

\[
\text{EFFBURN} = 49.1 + 0.222 \times \text{EXAIR} - 0.00533 \times \text{EXAIR}^2
\]

Recycle Ratio = 0.10

\[
\text{EFFBURN} = 54.1 - 0.245 \times \text{EXAIR} - 0.00033 \times \text{EXAIR}^2
\]

Recycle Ratio = 0.15

\[
\text{EFFBURN} = 55.6 - 0.265 \times \text{EXAIR} - 0.00081 \times \text{EXAIR}^2
\]

where \( \text{EFFBURN} \) = burner efficiency

The \( R^2 \) values of the above equations were 64.2%, 83.1%, 84.6%, and 92.3% respectively. The first one was quite low because there were some uncertain factors in the burner system such as the FBC bed depth changing during the tests.

Figure 22 shows the regression curves of four different recycle ratios. It appears that burner efficiency became lower at high excess air (larger than 20%) and the excess air of maximum burner efficiency seemed to switch to lower values as the recycle ratio increased. In general, the burner efficiency was not improved by recycling the refuse.
Figure 18. Relationship of Excess Air and Burner Efficiency
Recycle Ratio = 0.00
Figure 19. Relationship of Excess Air and Burner Efficiency
Recycle Ratio = 0.05
Figure 20. Relationship of Excess Air and Burner Efficiency
Recycle Ratio = 0.10
Figure 21. Relationship of Excess Air and Burner Efficiency
Recycle Ratio = 0.15
Figure 22. Effect of Recycle Ratio on Burner Efficiency
with the fuel in a single step operation.

6.3 SULPHUR DIOXIDE EMISSION

The sulphur is mostly released as sulphur dioxide, when fuels are burned, which combines with water to form acids. One method used to reduce sulphur dioxide emission is to provide limestone to the burner. Since the main purpose of this research was to investigate the effect of excess air and recycle ratio on all emissions, no SO$_2$ control method was used.

Figures 23, 24, 25, and 26 show that SO$_2$ emission, in PPM, decreases as excess air increases.

A regression analysis was conducted for different recycle ratios based on the excess air between 0-75%. A linear relationship of SO$_2$ emission (PPM) and excess air was found. The equations were

Recycle Ratio = 0.00

SO$_2$ = 2439-16.3*EXAIR

Recycle Ratio = 0.05

SO$_2$ = 2532-18.7*EXAIR

Recycle Ratio = 0.10

SO$_2$ = 2247-13.8*EXAIR

Recycle Ratio = 0.15

SO$_2$ = 2257-17.8*EXAIR

where $SO_2$ = the concentration of sulphur dioxide (PPM)
Figure 23. Relationship of Excess Air and SO₂ Emission
Recycle Ratio = 0.00
Figure 24. Relationship of Excess Air and SO₂ Emission
Recycle Ratio = 0.05
Figure 25. Relationship of Excess Air and SO$_2$ Emission
Recycle Ratio = 0.10
The $R^2$ values of the above equations were 75.2%, 89.4%, 89.5%, and 83.1% respectively.

Figure 27 shows the regression curves of four different recycle ratios. In general, it appears that $SO_2$ emission was reduced as the recycle refuse was added to the burner; and the higher recycle ratio, the lower the $SO_2$ emission. The refuse of coal often contains some free lime (4043 ug/g) which can act as a sorbent to react with sulphur as calcium sulphate. The above phenomenon can be used to explain why a recycle mode can reduce sulphur dioxide emission.

6.4 NITROGEN OXIDES EMISSION

Nitrogen oxides are produced in all combustion processes using air and they are major compounds in air pollution. Therefore, it is very important to understand NOx emission during combustion. There are various parameters related to the NOx emission such as excess air, bed temperature, residence time, coal type (nitrogen chemically combined in the coal), and recycle ratio. In this study, the effect of excess air and recycle ratio were the main parameters to be evaluated because bed temperature and pressure were controlled within a certain range.

Figures 28, 29, 30, and 31 show the experimental data of NO emission and excess air. From the literature review, it has been noted that NOx reduction can be achieved by burning with low excess air. Therefore, NOx emission should
Figure 27. Effect of Recycle Ratio on SO2 Emission
Figure 28. Relationship of Excess Air and NO Emission
Recycle Ratio = 0.00
Figure 29. Relationship of Excess Air and NO Emission
Recycle Ratio = 0.05
Figure 30. Relationship of Excess Air and NO Emission
Recycle Ratio = 0.10
Figure 31. Relationship of Excess Air and NO Emission
Recycle Ratio = 0.15
increase as excess air increases. However, a contrary result was obtained from this experiment. The possible reasons for this unexpected finding are: high values of excess air reduce the bed temperature, which in turn suppresses the thermal NOx generation. Figure 32 shows steady state bed temperature as a function of excess air.

The thermal NOx can be predicted by the following equation [ref 20, p.4-33]:

$$[\text{NO}] = K_1 e^{-\frac{K_2}{T}} [\text{N}_2][\text{O}_2]^{1/2} t$$

where $[ ]$ = concentration
$T$ = temperature
$t$ = residence time
$K_1, K_2$ = constants

From this equation, temperature is more important than excess air in the control of thermal NOx because it is an exponential variable.

In addition, there are two different flame mechanisms, diffusion flame and premixed flame, in combustion shown in Figure 33. OARDC-FBC belongs to the premixed flame system. Test results of Takahashi et al. [20] gave a bell shaped curve for the relationship of NOx emission and excess air for premixed flame (Figure 34). Their results agree with the results in this study.

A regression analysis was conducted for different recycle ratios based on the excess air between 0-75%. A linear relationship of NO emission (PPM) and excess air was found. The equations were
Figure 32. Relationship of Steady State Bed Temperature and Excess Air
Figure 33. Diffusion Flame and Premixed Flame Mechanisms (Singer, 1981)

Figure 34. NOx versus Excess Air for Premixed and Diffusion Flame (Singer, 1981)
Recycle Ratio = 0.00
\[ NO = 572 - 1.39 \times EXAIR \]
Recycle Ratio = 0.05
\[ NO = 575 - 1.85 \times EXAIR \]
Recycle Ratio = 0.10
\[ NO = 569 - 1.84 \times EXAIR \]
Recycle ratio = 0.15
\[ NO = 566 - 1.75 \times EXAIR \]

where \( NO \) = concentration of nitric oxide (PPM)

The \( R^2 \) values of the above equations were 54.3\%, 82.0\%, 87.0\%, and 70.6\% respectively.

Figure 35 shows the regression curves of four different recycle ratios. It appears that combustion with recycle fuel can reduce NO emission, but there was no clear relationship between the amount of reduction and recycle ratio.

Carbon monoxide was another pollutant from the burner system. Since CO emission was very low (around 0.14\% volume fraction), no analysis was done on it.

6.5 THE EFFECT OF FUEL PARTICLE SIZE

In the tests, coal was screened into three particle size: coarse coal (diameter > 0.149 mm), fine coal (diameter < 0.149 mm), and original coal as medium coal. Each group of coal particles was subjected to the combustion test. The results of tests of the effect of particle size on combustion efficiency are shown in Table 5.
Table 5. Effect of Fuel Particle Size

<table>
<thead>
<tr>
<th>Size</th>
<th>Combustion Eff.(%)</th>
<th>Excess air (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse (D &gt; .149mm)</td>
<td>91.47</td>
<td>25</td>
</tr>
<tr>
<td>Medium</td>
<td>92.26</td>
<td>28</td>
</tr>
<tr>
<td>Fine (D &lt; .149mm)</td>
<td>95.69</td>
<td>30</td>
</tr>
</tbody>
</table>

From Table 5, it is obvious that the fine coal had a higher combustion efficiency than the coarse coal. It should be noticed that the above experiments were in different excess air conditions because with the current OARDC-FBC it was not possible to adjust air flow, only fuel rate. The above values were the average of each group of experiments. In order to compare easily, a statistical analysis was conducted by using interpolation and/or extrapolation to find the combustion efficiency at an excess air value of 30%. The modified results are shown in Table 6.

Table 6. Effect of Fuel Particle Size on Combustion Efficiency (Excess Air = 30%)

<table>
<thead>
<tr>
<th>Size</th>
<th>Coarse</th>
<th>Medium</th>
<th>Fine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combustion Eff.(%)</td>
<td>91.53</td>
<td>92.32</td>
<td>95.69</td>
</tr>
</tbody>
</table>

The fine coal has more exposed surface than coarse coal based on the same weight and the combustion reaction rate is proportional to this contact surface between coal and air. This reason can be used to explained why the combustion rate
of fine coal is higher than that of coarse coal when sufficient air (oxygen) is available.

From reference 11, the researchers found that the greater the coal particle size, the higher the combustion efficiency. Their conclusion was the opposite of the results of this study. Before figuring out the reasons for these contrary results, the following phenomenon should be indicated first: fine coal has more exposed surface areas, but it is easily blown out and has short residence time in the burner; coarse coal has less exposed surface areas, but it has long residence time to react with air. Therefore, fine and coarse coal have advantageous and disadvantageous conditions at the same time. There probably exists an optimum particle size for combustion efficiency in a FBC, but that optimum particle size is variable with different FBCs. The following reasons can be used to explain why there exists different results: particle size range and the burner configurations were different. They used a square chamber and a circular chamber was used in this study; the particle size range was much wider in their research than that in this study. In addition, the combustion air flow velocity and media size were probably different.

6.6 HEAT LOSS FROM THE BURNER BODY

Evaluation of heat losses from the burner body ranged from 100 to 1900 watts. Figure 36 is a plot of experimental
results along with the theoretical curve for heat losses. Maximum heat loss measured was less than 5 percent of the total energy into the system. The main reason for the large scatter in the results is probably the error of unburned carbon analysis.
Figure 36. Experimental Results along with the Theoretical Curve for Heat Loss from the Burner Body
CHAPTER VII

CONCLUSIONS, SUMMARY, AND RECOMMENDATIONS

The conclusions from this study are as follows:

1. The continuous recycle method can improve the combustion efficiency but not burner efficiency for the OARDC-FBC.

2. The recycle technology can suppress the pollutants emission.

3. The heat loss from burner body is about 2.5-3.5% of the total input energy and it changes only slightly within the operational cooling air rate.

In summary, the combustion efficiency of the burner is around 89%, burner efficiency around 53%, SO\textsubscript{2} emission around 4.1 lb/million Btu, and NO\textsubscript{x} emission around 1.3 lb/million Btu at an excess air level of 20%. Unfortunately, the pollutants emission are higher than the regulations of EPA. Therefore, this system will need to apply some technologies to control pollution in order to meet EPA standards. Since the fluidized bed combustor has a very high combustion efficiency, there usually is no problem with carbon monoxide emission.
Recommendations for further study are:

1. The tests of this study were not conducted under continuous recycling operation due to the present facilities. Recycle equipment should be added to the burner system and tested under continuous recycling operation.

2. There was no high excess air value to investigate the result of rich air test and there was no low excess air value to evaluate the maximum burner efficiency for recycle test. In order to get more convincing results, some high excess air values (above 80%) and low excess air values (below 0%) tests should be conducted.

3. Since the recycle mode decreases the bed temperature and the cooling air is controlled by bed temperature, decreasing the trigger temperature of cooling air would probably improve the burner efficiency with the recycle mode.

4. If it is desired to reduce pollution, tests with limestone and/or ammonia added to the burner system should be conducted.

5. More tests and wider fuel particle size range are required in order to quantify further the effect of fuel particle size on combustion efficiency.
REFERENCES


APPENDIX A

SUMMARY OF EXPERIMENTAL DATA
Experimental results are listed in this section. Tests without recycling refuse are listed in Table 7. Tables 8, 9, and 10 present the results for tests with recycle ratios of 0.05, 0.10, and 0.15 respectively.

Table 7. Experimental Data of Test without Recycling Refuse

<table>
<thead>
<tr>
<th>Excess Air (%)</th>
<th>Combustion Efficiency (%)</th>
<th>Burner Efficiency (%)</th>
<th>SO₂ Emission (PPM)</th>
<th>NO Emission (PPM)</th>
<th>Bed Temp. (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>87.14</td>
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<td>2364</td>
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Table 8. Experimental Data of Test with a Recycle Ratio of 0.05

<table>
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<tr>
<th>Excess Air (%)</th>
<th>Combustion Efficiency (%)</th>
<th>Burner Efficiency (%)</th>
<th>SO₂ Emission (PPM)</th>
<th>NO Emission (PPM)</th>
<th>Bed Temp. (°C)</th>
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</table>

Table 9. Experimental Data of Test with a Recycle Ratio of 0.10

<table>
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<th>Excess Air (%)</th>
<th>Combustion Efficiency (%)</th>
<th>Burner Efficiency (%)</th>
<th>SO₂ Emission (PPM)</th>
<th>NO Emission (PPM)</th>
<th>Bed Temp. (°C)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>2139</td>
<td>535</td>
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<table>
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<tr>
<th>Excess Air (%)</th>
<th>Combustion Efficiency (%)</th>
<th>Burner Efficiency (%)</th>
<th>$SO_2$ Emission (PPM)</th>
<th>NO Emission (PPM)</th>
<th>Bed Temp. (°C)</th>
</tr>
</thead>
<tbody>
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<td>1922</td>
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</tbody>
</table>
APPENDIX B

COMPUTER PROGRAM FOR EXPERIMENTAL DATA ANALYSIS
C THIS PROGRAM (FBCSI) IS USED TO CALCULATE BURNER
C PERFORMANCE SUCH AS COMBUSTION EFFICIENCY AND THE USEFUL
C ENERGY OUT OF THE BURNER

C

CHARACTER DATE*8
DIMENSION D(2), DP(2), SP(2), TF(2), QA(2)
DIMENSION ZW(9), ZWR(9), ZV(9)
DIMENSION XV(9), XMW(9), XWF(7), A(4, 9), P(4), H(9)
DATA S/0.1375, 0.2793/, D/1.998, 1.998/, TK/273.0/
DATA TREF/25./
DATA XMW/28.016, 32.0, 18.016, 44.011, 39.9441,
DATA XMC/.0437/
DATA XWF/.0149, .0892, .0543, .7363, .0311, .0741, 7480./
DATA XV/.7552, 2315, .0, .0004, .0128, .0, .0, .0,/.
DATA A/6.67, 0.606, 0.13, 0.827, 0.258, 0, -1.88,
# 8.86, 0, 0, 7.7, 5.3, -0.83, 0, 4.953, 0, 0, 0, 11.4, 1.414, 0,
# -2.045, 6.6, 1.2, 0, 0.7, 133, 0, 0, 0, 0.89, 0, 0, 0/
WRITE(6,*) '-----------------------------------PROGRAM
FBCSI-----------------------------------'
WRITE(6,*) '
WRITE(6,*) 'This program is designed to accept
# laboratory data and do the energy and'
WRITE(6,*) 'mass balances. It begins with a #series of
questions asked for fuel type,'
WRITE(6,*) 'fuel rate, air flow rates and specific
# temperatures'
WRITE(6,*) '
WRITE(6,*) '---------------------------TO START HIT
#RETURN-----------------------------'
read(5,*)
WRITE(6,*) '
C Orifice plate diameters are d1=0.940, d2=1.283, where 1 is
C the pipe of combustion air.
10 WRITE(6,*) 'INPUT-->DATE(Month-Day-Year), Time(Hour
# xx.xx)
WRITE(6,*) '
READ(5,100) DATE,TIME
100 FORMAT(A8,1X,F5.2)
WRITE(6,*) 'INPUT(FUEL)-->TOTAL RATE(KG/HR)
WRITE(6,*) '
READ(5,*) QC1
WRITE(6,*) 'INPUT(RECYCLE FUEL)-->RATIO, CARBON
# CONTENT(%)'
WRITE(6,*) '
WRITE(6,*) ADDRAT, RC
RC=RC/100.
RECYC=ADDRAT*QC1/(1+ADDRAT)
QC=QC1-RECYC
WRITE(6,*) 'INPUT(REFUSE)--CARBON CONTENT OF OUTPUT
# ASH(%)'
WRITE(6,*) '
READ(5,*) CCA
CCA=CCA/100.
WRITE(6,*) 'INPUT(AMBIENT AIR)-->DRY BULB, DEW POINT,
#PRESS(MM HG)'
WRITE(6,*) '   ',
READ(5,*) TAIRO,TDPO,PAIRO
WRITE(6,*) 'INPUT(COMB. AIR)-->SP1(MM H2O), DP1(MM
#H2O), Tin, Tout'
WRITE(6,*) '   ',
READ(5,*) SP1,DP1,TB1,TB2
SP1=SP1/13.595
WRITE(6,*) 'INPUT(COOLING AIR)-->SP2(MM HG), DP2(MM
#H2O), Tin, Tout, RTime'
WRITE(6,*) '   ',
READ(5,*) SP2,DP2,TC1,TC2,TR
WRITE(6,*) 'INPUT(GASES VOLUME RATIO)-->O2,CO2,SO2,
#CO,NO,NOx'
WRITE(6,*) '   ',
READ(5,*) ZV(2),ZV(4),ZV(6),ZV(7),ZV(8),ZV(9)
TAIR=1.8*TAIRO+32.
TDPO=1.8*TDPO+32.
PAIRO=PAIRO/25.4
TC=1.8*TC1+32.
SP(1)=SP1/25.4
DP(1)=DP1/25.4
SP(2)=SP2/25.4
DP(2)=DP2/25.4
AMW=0.
XV(3)=0.
DO 14 I=1,9
14 AMW=AMW+XV(I)*XMW(I)
PFR=.4912*PAIR
TF(1)=TAIR+459.6
TF(2)=TC+459.6
X=TDPO+459.6
PD=VAP(X)
XV(3)=XMW(3)*PD/(AMW*(PF-PD))
Z=1.0
FA=1.00
FM=1.00
FC=1.00
DO 18 J=1,2
HW=DP(J)
PF=PFR+.4912*SP(J)
Y=1-(FM*FM*HW/PF)*.0125
GF=(AMW*(PF-PD)+XMW(3)*PD)/(10.73*TF(J)*Z)
QA(J)=359*S(J)*D(J)*D(J)*FA*FM*FC*Y*(GF*HW)**0.5
QA(J)=QA(J)*0.4536
18 CONTINUE
QA(2)=QA(2)*TR
WRITE(6,106) DATE,TIME,TAIRO,TDPO,PAIRO,ADDRAT
DYC=(1.-XMC)*QC

85
DYM = XMC * QC
XMA = XV(3) / (1 + XV(3))
DYA = (1 - XMA) * QA(1)
DYAM = XMA * QA(1)
DYA2 = (1 - XMA) * QA(2)
DYAM2 = XMA * QA(2)

C Calculate the weight of refuse by assuming that all the combustible in the refuse is carbon
ACR = (DYC * XWF(6) + RECYC * (1 - RC)) / (DYC + RECYC)
ASH = ACR * (DYC + RECYC) / (1 - CCA)

C Calculate the oxygen needed to burn 1.0 kg dry coal completely
EOO = (XWF(3) / 4.012 + XWF(4) / 12.011 + XWF(5) / 32.033) * 32. * DYC
EOO = EOO - DYC * XWF(2) + RC * RECYC * 32. / 12.011
EXC = (DYA * XW(2) - EOO) / EOO

WRITE(6,108) DYC, DYM, TAIR0, RECYC, RC,
#DYA, DYA, YB1, TB2, DYA2, DYAM2, TC1, TC2, EXC

C Calculate the water produced as burning 1.0 kg dry coal
ZW3 = DYAM + DYM + XWF(3) * DYC * XMW(3) / 2.016
ZW5 = 0.0128 * DYA
GASRATE = QC1 + QA(1) - ASH
ANOH2O = GASRATE - ZW3
ZV(3) = 0

C Calculate the constituents of exhaust gas by supposing that the dry volume ratio of Argon is between .004-.04
DO 50 K = 4000, 4000
ZV(5) = K / 100000.
ZV(I) = 1 - ZV(2) - ZV(4) - ZV(5) - ZV(6) - ZV(7) - ZV(8) - ZV(9)
TZ = 0
DO 22 I = 1, 9
TZ = TZ + ZV(I) * XMW(I)
22 CONTINUE
DO 23 I = 1, 9
ZW(I) = ANOH2O * ZV(I) * XMW(I) / TZ
23 CONTINUE
AA = ZW(5) - ZW5
IF(ABS(AA) .LE. 0.001) GOTO 40
IF(K .NE. 4000) GOTO 50
WRITE(6,*) 'RESET ARGON VOLUME RATIO'
GOTO 20
50 CONTINUE
40 ZW(3) = ZW3
TOTWT = 0.
TOTMOL = 0.
DO 25 I = 1, 9
TOTWT = TOTWT + ZW(I)
ZV(I) = ZW(I) / XMW(I)
25 TOTMOL = TOTMOL + ZV(I)
DO 27 I = 1, 9
ZW(I) = ZW(I) / TOTWT
27 ZV(I) = ZV(I) / TOTMOL
WRITE(6,109)
WRITE(6,209) (ZW(I),ZWR(I),ZV(I), I=1,9),ASH

C Calculate the combustion efficiency
REDC=ASH-XWF(6)*DVC-(1-RC)*RECYC
ECOMB=XWF(7)
TNWCOM=ECOMB*DVC
TOTCOM=TNWCOM+RC*RECYC*7823.1
TNOCOM=REDC*7823.1+ZN(7)*943.1
EFF=(TOTCOM-TNOCOM)/TOTCOM
WRITE(6,104) EFF

104 FORMAT(12X,'COMBUSTION EFFICIENCY = ',F7.4/)
EIN=TOTCOM+.36*(TAIR0-TREF)*DVC+1.0*
##(TBI-TREF)*DYCM+.24*(TBI-TREF)*DYA+.425*
##(TBI-TREF)*DYAM+.24*(TC1-TREF)*DYA2+
##.425*(TC1-TREF)*DYAM2-583.4*(ZW3-DYAM)

C ---------------------------------------------
C Unit conversion 1.0 Kcal/hr=1.1628 Watts
C ---------------------------------------------

EIN=EIN*1.1628
TC=TB2+TK
TR=TREF+TK
P(1)=TC-TR
P(2)=(TC+TC-TR*TR)/2.*0.001
P(3)=1.0E-6*(TC**3-TR**3)/3
P(4)=-(1./TC-1./TR)*1.0E5
DO 29 I=1,9
H(I)=0.
DO 29 J=1,4
29 H(I)=H(I)+A(J,I)*P(J)
EOCOMB=0.0
DO 30 I=1,9

30 EOCOMB=EOCOMB+H(I)*ZW(I)/XMW(I)
EOCOMB=EOCOMB*1.1628
TC=TC2+TK
TR=TREF+TK
P(1)=TC-TR
P(2)=(TC+TC-TR*TR)/2.*0.001
P(3)=1.0E-6*(TC**3-TR**3)/3
P(4)=-(1./TC-1./TR)*1.0E5
DO 31 I=1,9
H(I)=0.
DO 31 J=1,4
31 H(I)=H(I)+A(J,I)*P(J)
EOCOOL=0.0
DO 32 I=1,9

32 EOCOOL=EOCOOL+H(I)*XV(I)*DYA2/XMW(I)
EOCOOL=EOCOOL*1.1628
OTHER=EIN-EOCOMB-EOCOOL
WRITE(6,107)
#TREF,-EIN,1.0,EOCOMB,EOCOMB/EIN,EOCOOL,EOCOOL/EIN,
#OTHER,OTHER/EIN
M=0
READ(5,*) M
IF (M-1) 20,10,20
106 FORMAT(33X,'BURNER PERFORMANCE'/
   $ 20X,'DATE ','A8,18X,'TIME',F6.2//
   # 12X,'AMBIENT AIR CONDITIONS'/
   # 20X,'DRY BULB,C DEW PT.,C'
  $PRESSURE,MM.HG'/
  $20X,'----------------------------------------'/*
  $ 20X,2F10.2,F15.2/'
  $ 12X,'RECYCLE RATIO =',F5.2/) 108 FORMAT(12X,'MEASURED VARIABLES DURING TEST'/
   # 30X,' FLOW,KG/HR TEMPERATURE,C'/
   # 30X,' DRY WATER IN OUT'/
   # 30X,'----------------------------------------'/*
   # 14X,'NEW FUEL',6X,2F8.2,F8.1/
   # 14X,'RECYCLE FUEL',2X,F8.2,4X,'(C CONTENT =',F6.3,')'/
   # 14X,'COMBUSTION AIR',2F8.2,2F8.1/
   # 14X,'COOLING AIR',3X,2F8.2,2F8.1/
   # 20X,'(EXCESS AIR FOR COMBUSTION IS ',F4.2,'/)')
109 FORMAT(12X,'PRODUCTS FROM COMBUSTION'/
   # 32X,' KG/HR WEIGHT VOLUME'/
   # 40X,'RATIO RATIO'/
   # 30X,'----------------------------------------'/*
209 FORMAT(14X,'NITROGEN',7X,3F8.3/
   # 14X,'OXYGEN',9X,3F8.3/
   # 14X,'WATER',10X,3F8.3/
   # 14X,'CO2',12X,3F8.3/
   # 14X,'ARGON',11X,3F8.4/
   # 14X,'SO2',13X,3F8.4/
   # 14X,'CO',14X,3F8.4/
   # 14X,'NO',14X,3F8.4/
   # 14X,'NOX',13X,3F8.4/
   # 14X,'ASH',12X,F8.3/) 107 FORMAT(12X,'ENERGY BALANCE: TREF=',F5.1/
   # 35X,' WATTS RATIO'/
   # 30X,'----------------------------------------'/*
   # 14X,'FUEL',15X,F10.1,F8.3/
   # 14X,'EXHAUST GASES',6X,F10.1,F8.3/
   # 14X,'COOLING AIR',8X,F10.1,F8.3/
   # 14X,'OTHER',14X,F10.1,F8.3)
20 STOP
END
FUNCTION VAP(T)
IF(T-491.67)1,1,2
1 ARG=23.3924-11286.6489/T-0.46057*ALOG(T)
VAP = EXP(ARG)
RETURN
2 ARG=54.6329-12301.688/T-5.16923*ALOG(T)
VAP = EXP(ARG)
RETURN
END
APPENDIX C

OUTPUT SAMPLES OF COMPUTER PROGRAM
**BURNER PERFORMANCE**

**DATE** 09-15-87  **TIME** 13:13

**AMBIENT AIR CONDITIONS**

<table>
<thead>
<tr>
<th>DRY BULB, C</th>
<th>DEW PT., C</th>
<th>PRESSURE, MM.HG</th>
</tr>
</thead>
<tbody>
<tr>
<td>25.70</td>
<td>8.70</td>
<td>740.00</td>
</tr>
</tbody>
</table>

**RECYCLE RATIO = 0.00**

**MEASURED VARIABLES DURING TEST**

<table>
<thead>
<tr>
<th>FLOW, KG/HR</th>
<th>TEMPERATURE, C</th>
</tr>
</thead>
<tbody>
<tr>
<td>DRY WATER</td>
<td>IN OUT</td>
</tr>
</tbody>
</table>

| NEW FUEL     | 4.69 | 0.21 | 25.7 |
| RECYCLE FUEL | 0.00 | (C CONTENT = 0.000) |
| COMBUSTION AIR | 56.62 | 0.40 | 38.4 | 686.2 |
| COOLING AIR  | 107.79 | 0.77 | 37.6 | 721.3 |

*(EXCESS AIR FOR COMBUSTION IS 0.20)*

**PRODUCTS FROM COMBUSTION**

<table>
<thead>
<tr>
<th>KG/HR</th>
<th>WEIGHT</th>
<th>VOLUME</th>
</tr>
</thead>
<tbody>
<tr>
<td>NITROGEN</td>
<td>44.444</td>
<td>0.727</td>
</tr>
<tr>
<td>OXYGEN</td>
<td>3.728</td>
<td>0.061</td>
</tr>
<tr>
<td>WATER</td>
<td>2.890</td>
<td>0.047</td>
</tr>
<tr>
<td>CO₂</td>
<td>8.919</td>
<td>0.146</td>
</tr>
<tr>
<td>ARGON</td>
<td>0.7237</td>
<td>0.0118</td>
</tr>
<tr>
<td>SO₂</td>
<td>0.2986</td>
<td>0.0049</td>
</tr>
<tr>
<td>CO</td>
<td>0.0877</td>
<td>0.0014</td>
</tr>
<tr>
<td>NO</td>
<td>0.0328</td>
<td>0.0005</td>
</tr>
<tr>
<td>NO₂</td>
<td>0.0512</td>
<td>0.0008</td>
</tr>
<tr>
<td>ASH</td>
<td>0.744</td>
<td></td>
</tr>
</tbody>
</table>

**COMBUSTION EFFICIENCY = 0.9092**

**ENERGY BALANCE : TREF = 25.0**

<table>
<thead>
<tr>
<th>WATTS</th>
<th>RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>FUEL</td>
<td>-39671.6</td>
</tr>
<tr>
<td>EXHAUST GASES</td>
<td>12261.3</td>
</tr>
<tr>
<td>COOLING AIR</td>
<td>22130.0</td>
</tr>
<tr>
<td>OTHER</td>
<td>5280.3</td>
</tr>
</tbody>
</table>

Table 11. Results of Test without Recycling Refuse
### BURNER PERFORMANCE

**DATE 09-25-87**

**TIME 15.92**

<table>
<thead>
<tr>
<th>AMBIENT AIR CONDITIONS</th>
<th>DRY BULB,C</th>
<th>DEW PT.,C</th>
<th>PRESSURE, MM. HG</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>30.50</td>
<td>17.25</td>
<td>739.00</td>
</tr>
</tbody>
</table>

**RECYCLE RATIO = 0.10**

### MEASURED VARIABLES DURING TEST

<table>
<thead>
<tr>
<th></th>
<th>FLOW, KG/HR</th>
<th>TEMPERATURE, C</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DRY</td>
<td>WATER</td>
</tr>
</tbody>
</table>

- **NEW FUEL**: 4.34 | 0.20 | 30.5
- **RECYCLE FUEL**: 0.45 | (C CONTENT = 0.511)
- **COMBUSTION AIR**: 55.80 | 0.70 | 43.4 | 685.3
- **COOLING AIR**: 92.64 | 1.16 | 42.3 | 717.2

*(EXCESS AIR FOR COMBUSTION IS 0.20)*

### PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th></th>
<th>KG/HR</th>
<th>WEIGHT</th>
<th>VOLUME</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>NITROGEN</strong></td>
<td>42.765</td>
<td>0.708</td>
<td>0.741</td>
</tr>
<tr>
<td><strong>OXYGEN</strong></td>
<td>3.769</td>
<td>0.062</td>
<td>0.057</td>
</tr>
<tr>
<td><strong>WATER</strong></td>
<td>3.004</td>
<td>0.050</td>
<td>0.081</td>
</tr>
<tr>
<td><strong>CO₂</strong></td>
<td>9.695</td>
<td>0.161</td>
<td>0.107</td>
</tr>
<tr>
<td><strong>ARSON</strong></td>
<td>0.7134</td>
<td>0.0118</td>
<td>0.0087</td>
</tr>
<tr>
<td><strong>SO₂</strong></td>
<td>0.2539</td>
<td>0.0042</td>
<td>0.0019</td>
</tr>
<tr>
<td><strong>CO</strong></td>
<td>0.0986</td>
<td>0.0016</td>
<td>0.0017</td>
</tr>
<tr>
<td><strong>NO</strong></td>
<td>0.0299</td>
<td>0.0005</td>
<td>0.0005</td>
</tr>
<tr>
<td><strong>NO₂</strong></td>
<td>0.0464</td>
<td>0.0008</td>
<td>0.0005</td>
</tr>
<tr>
<td><strong>ASH</strong></td>
<td>1.120</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**COMBUSTION EFFICIENCY = 0.8656**

**ENERGY BALANCE : TREF = 25.0**

<table>
<thead>
<tr>
<th></th>
<th>WATTS</th>
<th>RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>FUEL</strong></td>
<td>-36933.8</td>
<td>1.000</td>
</tr>
<tr>
<td><strong>EXHAUST GASES</strong></td>
<td>12106.6</td>
<td>0.328</td>
</tr>
<tr>
<td><strong>COOLING AIR</strong></td>
<td>19104.1</td>
<td>0.517</td>
</tr>
<tr>
<td><strong>OTHER</strong></td>
<td>5723.1</td>
<td>0.155</td>
</tr>
</tbody>
</table>

Table 12. Results of Test with a Recycle Ratio of 0.10
APPENDIX D

THEORETICAL METHOD TO ANALYZE THE HEAT LOSS
Heat out of the OARDC burner is categorized into three areas: clean air with high temperature, exhaust air, and "other" heat loss. The "other" heat loss consists of unburned fuel and heat loss from burner body. In order to understand the amount of heat loss from the burner body in the "other" area, a thermal resistance network was developed to evaluate the heat loss from burner body.

D.1 REPRESENTATION OF BURNER

The burner is constructed by two cylindrical shells (one is used as burner chamber, the other as cyclone), heat exchanger (coaxial cylindrical shell outside burner chamber), perlite as insulator, and the outer wall. Table 13 lists the dimensions of the burner. Figure 37 is a cross section of the burner showing the parameters in this theoretical study.

The heat flow of this burner is a three dimensional problem. However, it can be simplified to be a two dimensional system in the radial direction for this problem. The bed temperature is very uniform across the burner due to the high rate of heat transfer between points within a fluidized bed. This high heat transfer characteristic results from turbulent motion and rapid circulation rate of the solid particles within the fluidized air. The fluidized bed temperature was controlled between 820-850°C. A temperature of 835°C was used in this study because this
Figure 37. Sectional Sketch of the Fluidized Bed Combustor
value was close to the mean cycle temperature. For analysis, the cross section of the burner was divided into isothermals represented by node points (Figure 38). Therefore, this system was simplified by breaking up the structure into a group of elements which were interconnected by thermal resistors. The network of resistances and nodes is called an analog resistor network (Figure 39). Figure 40 is the simplified form of figure 39.

Table 13. Dimensions of the Burner

<table>
<thead>
<tr>
<th>Variable</th>
<th>Radial Distance</th>
<th>Meter</th>
</tr>
</thead>
<tbody>
<tr>
<td>V₁</td>
<td>inner side of chamber wall</td>
<td>0.0761</td>
</tr>
<tr>
<td>V₂</td>
<td>outer side of chamber wall</td>
<td>0.0841</td>
</tr>
<tr>
<td>V₃</td>
<td>inner side of heat exchanger wall</td>
<td>0.0881</td>
</tr>
<tr>
<td>V₄</td>
<td>outer side of heat exchanger wall</td>
<td>0.0901</td>
</tr>
<tr>
<td>V₅</td>
<td>inner side of outer wall (large)</td>
<td>0.3028</td>
</tr>
<tr>
<td>V₆</td>
<td>outer side of outer wall (large)</td>
<td>0.3048</td>
</tr>
<tr>
<td>V₇</td>
<td>inner side of cyclone</td>
<td>0.0996</td>
</tr>
<tr>
<td>V₈</td>
<td>outer side of cyclone</td>
<td>0.1016</td>
</tr>
<tr>
<td>V₉</td>
<td>inner side of outer wall (small)</td>
<td>0.2774</td>
</tr>
<tr>
<td>V₁₀</td>
<td>outer side of outer wall (small)</td>
<td>0.2794</td>
</tr>
<tr>
<td>DCL</td>
<td>distance between cylinder centers</td>
<td>0.2508</td>
</tr>
<tr>
<td>L</td>
<td>length of burner</td>
<td>1.524</td>
</tr>
</tbody>
</table>

Large two and three dimensional analog resistor networks often use a high speed digital computer to determine the thermal profile of complex structure. An analog resistor network can be transformed to an electrical circuit and solved by Kirchhoff's law by writing a voltage balance equation for each circuit. The simultaneous solution of those equations results in the heat flow between two
Figure 38. Isothermals of the burner
Figure 39. Analog Resistor Network for Two Dimensional Heat Flow
Figure 40. Simplified Form of Figure 39.

Diagram showing components labeled as $T_{\infty}$, $T_{\text{cyclone}}$, $T_{\text{bed}}$, $R_1$, $R_2$, $R_3$, $R_4$, $R_5$, $R_6$, $R_7$, $T_{\text{out}}$, and $T_{\text{in}}$. Connections include series and parallel configurations.
points. The heat loss of this system is the heat flow between \( T_6 \) and \( T_2 \).

The length of the burner was considered as five feet and the top and bottom areas were ignored.

**D.2 THE THERMAL RESISTANCE OF ANALOG RESISTOR NETWORK**

There exists an analogy between the diffusion of heat and electrical charge. Just as an electrical resistance may be associated with the conduction of electricity, a thermal resistance may be associated with the conduction of heat. Defining resistance as the ratio of a driving potential to the corresponding transfer rate, the thermal resistance for conduction is represented as

Cartesian system:

\[
R_{\text{cond}} = \frac{T_a - T_b}{q} = \frac{L}{K*A} \quad (D.1)
\]

Cylindrical system:

\[
R_{\text{cond}} = \frac{\ln(V_b/V_a)}{2*\pi*K} \quad (D.2)
\]

From figure D2, it is obvious that heat transfer of isothermal \( T_4-T_m \), \( T_8-T_m \), and \( T_m-T_5 \) is a two dimensional conduction problem. The resistances of \( T_4-T_m \) and \( T_8-T_m \) can be treated as a circular cylinder in a square solid. From reference 6, the resistance of this kind of configuration is

\[
R_{\text{cond}} = \frac{\ln(1.08*W/D)}{2*\pi*K} \quad (D.3)
\]
The resistance of $T_m - T_5$ can be solved by dividing the area between $T_m$ and $T_5$ into 10 parts (Figure 41). Then the resistance is equal to the equivalent resistance of 10 parallel resistances. There are six resistances which can be determined by using equation (D.1), and the remaining four using equation (D.2). However, the values of the four resistances are four times that calculated in equation (D.2). That is because it has approximately one quarter of the configuration of hollow cylinders.

A thermal resistance may also be associated with heat transfer by convection at a surface. From Newton's law of cooling,

$$q = h*A*(T_a - T_b) \quad \text{(D.4)}$$

the thermal resistance for convection is

$$R_{\text{conv}} = \frac{1}{h*A} \quad \text{(D.5)}$$

The thermal resistance in cooling air can be represented by the following two equations.

$$q = m*C_p*(T_a - T_b) \quad \text{(D.6)}$$

$$1 \quad \text{(D.7)}$$

$$R = \frac{1}{m*C_p}$$

The radiation exchange between two long concentric cylinder surface is

$$q = \frac{\sigma*A*(T_a^4 - T_b^4)}{1/E_a^*(1-E_p)/E_a^*(V_a/V_b)} \quad \text{(D.8)}$$
Figure 41. Configuration of Divided Regions in Insulator of the Burner
the thermal resistance for radiation is

\[ R_{\text{rad}} = \frac{1/E_a + (1-E_b)/E_a*(V_a/V_b)}{\sigma^*A^* (T_a^2 + T_b^2) * (T_a + T_b)} \]  

(D.9)

D.3 HEAT TRANSFER COEFFICIENT \( (h) \) FOR CONVECTION

D.3.1 Forced Convection

There are three \( h \) values which have to be determined in order to calculate the resistance of forced convection. Since the fluidized air always reaches turbulent state in a burner chamber, it is not necessary to consider the laminar state. Although the \( h \) value from bed to surface is known to be higher than that from gas alone, internal flow of pure gas is considered at this point. From reference 6, \( h \) value can be obtained by using the following equations.

\[ Re = \frac{m*D}{A_c*\mu} \]  

(D.10)

\[ Nu = 0.023*Re^{0.8}*Pr^n \]  

(D.11)

\[ Nu*K \]

\[ h = \frac{-}{D} \]  

(D.12)

Because the mass flow rate in cyclone is almost the same as that in burner chamber, equations (D.10) to (D.12) can be used again to determine the \( h \) value of exhaust gases in cyclone.

The last one is cooling air in the heat exchanger. This problem is the heat transfer in a concentric tube annulus. In addition, because cooling air can be adjusted, the air
flow in the heat exchanger could be either turbulent flow or laminar flow. Since the ratio of heat exchanger perimeter to its gap is very large, this concentric tube annulus can be treated as a rectangular cross section. The Nusselt number for fully developed laminar flow in this kind of cross section is 7.54 [ref 6, p.364].

For fully developed laminar flow:

\[
\frac{7.54 * K}{D_h} = h \quad (D.13)
\]

For fully developed turbulent flow, it can be assumed that the inner and outer convection coefficient in the heat exchanger are equal approximately. Therefore, the value may be evaluated by using equations (D.10) to (D.12) and using hydraulic diameter \((V_4-V_3)\) to replace \(D\).

D.3.2 Natural Convection

Heat transfer between the outer wall and the surroundings is due to natural convection. The convection coefficient is dependent on a characteristic length, \(L\), and the temperature difference between the surface and the surroundings. From reference 6,

\[
\frac{g* \beta * (T_a - T_L) * L^3}{\alpha \gamma} \quad \text{Ral} = \quad (D.14)
\]

\[
Nu = \frac{0.387 * \text{Ral}^{1/6}}{\left[1 + \left(0.492 / \text{Pr}\right)^{9/16}\right]^{8/27}} \quad \text{Ral} > 10^9 \quad (D.15)
\]

\[
Nu = \frac{0.67 * \text{Ral}^{1/4}}{\left[1 + \left(0.492 / \text{Pr}\right)^{9/16}\right]^{4/9}} \quad 0 < \text{Ral} < 10^9 \quad (D.16)
\]
\[ h = \frac{\text{Nu} \times K}{L} \]  

(D.17)

D.4 ITERATION SCHEME

In order to obtain the resistances of natural convection and radiation, it is necessary to assume a pair of initial values for \( T_a \) and \( T_b \) and calculate the resistances of natural convection and radiation, then to check the validity of the assumed values by analyzing the analog resistor network to determine the computed \( T_a \) and \( T_b \). Using these new values of \( T_a \) and \( T_b \), resistances of natural convection and radiation are recalculated and the process repeated until satisfactory agreement is obtained between the initial and computed values of \( T_a \) and \( T_b \). Therefore, due to the nature of the problem, it is necessary to use an iteration scheme to get an accurate result.

D.5 KIRCHHOFF'S LAW

The analog resistor network can be redrawn to be a electrical circuit (Figure 42). The current (heat flow) can be computed by using Kirchhoff's law.

\[
\begin{align*}
R_1*(q_1+q_2) + R_6*(q_1-q_5) + R_4*(q_1-q_3) + R_5*(q_1+q_4) &= T_A \\
R_1*(q_1+q_2) + R_2*(q_2+q_5) + R_7*(q_2+q_3) &= T_B \\
R_8*(q_3+q_4) + R_4*(q_3-q_1) + R_3*(q_3-q_5) + R_7*(q_3+q_2) &= T_C \\
R_8*(q_4+q_3) + R_5*(q_4+q_1) &= T_D \\
R_2*(q_5+q_2) + R_3*(q_5-q_3) + R_6*(q_5-q_1) &= 0
\end{align*}
\]

(D.18)

Rewriting equation (D.18) to get the following linear equations
Figure 42. Equivalent Electrical Circuit of the Burner
\[(\text{AW})*q_1+(\text{R}_1)*q_2+(-\text{R}_4)*q_3+(\text{R}_5)*q_4+(-\text{R}_6)*q_5 = \text{TA}\]
\[(\text{R}_1)*q_1+(\text{BW})*q_2+(\text{R}_7)*q_3+(\text{DW})*q_4+(\text{NW})*q_5 = \text{TB}\]
\[(-\text{R}_4)*q_1+(\text{R}_7)*q_2+(\text{CW})*q_3+(\text{R}_8)*q_4+(-\text{R}_3)*q_5 = \text{TC}\]
\[(\text{R}_5)*q_1+(\text{R}_8)*q_2+(\text{R}_3)*q_3+(\text{DW})*q_4+(\text{NW})*q_5 = \text{TD}\]
\[(-\text{R}_6)*q_1+(\text{R}_2)*q_2+(-\text{R}_3)*q_3+(\text{NW})*q_4+(\text{EW})*q_5 = 0\]

where
\[
\text{TA} = \text{bed temperature - ambient temperature}
\]
\[
\text{TB} = \text{TA}
\]
\[
\text{TC} = \text{cyclone temperature - ambient temperature}
\]
\[
\text{TD} = \text{TC}
\]
\[
\text{AW} = \text{R}_1+\text{R}_2+\text{R}_3+\text{R}_4
\]
\[
\text{BW} = \text{R}_1+\text{R}_2+\text{R}_7
\]
\[
\text{CW} = \text{R}_3+\text{R}_4+\text{R}_7+\text{R}_8
\]
\[
\text{DW} = \text{R}_5+\text{R}_8
\]
\[
\text{EW} = \text{R}_2+\text{R}_3+\text{R}_6
\]

To solve the above inhomogeneous set of linear equation, Gauss elimination and pivoting methods were used and they were represented in program HLBURN.

\section*{D.6 DEVELOPMENT OF COMPUTER PROGRAM}

A computer program, HLBURN, was developed to evaluate the heat loss from burner body and surface temperature of burner based on different cooling air rates. The first part of the program was to choose the value of every parameter. Some parameters used in program are listed in Appendix E. Then those values and the above equations were used to calculate the thermal resistance of the analog resistor network and substituted those thermal resistances into equation (D.18). Finally, the Gauss elimination method was used to compute the heat flow at each circuit and display the results. The program is listed in Appendix F.
D.7 RESULT OF HEAT LOSS FROM BURNER BODY

The program HLBURN was used to find the heat loss from the burner body by inputting different cooling air rates. The results are shown in Figure 43. Heat loss from the burner body decreased only slightly as cooling air rate varied between 0-120 kg/hr. The reason for this is that the thermal resistance from bed to ambient the surroundings is nearly constant. This is because the major thermal resistance is contributed by the insulation and the thermal resistance at the chamber wall to heat exchanger wall, although varying, is relatively small in value.

There was almost no change for outside wall surface temperature as cooling air rate was changed. The surface temperature calculated by the program was about 56°C (Figure 44). This value was very close to the actual measured temperature (55-65°C) during system operation.

There is another heat loss path - thermocouple well. To investigate how much heat is lost through this path, the surface temperature of the thermocouple was actually measured. When the system was operating, surface temperature of the thermocouples were about 120°C. Using equations (D.14)-(D.17), the heat loss from all eight of the thermocouples is about 63 watts. The detailed calculation is shown in Appendix G.
Figure 44. Surface Temperature of the Outer Wall
APPENDIX E

EVALUATION OF PARAMETERS IN THEORETICAL HEAT LOSS MODEL
The parameters required to calculate the thermal resistance of the burner are evaluated below.

**DIMENSION OF THE BURNER**

The dimensions of the burner were listed in Table 13. The values were obtained from the original design drawing. It should be noted that the cyclone is tapered at the bottom part, but was assumed to be a cylinder for this study.

**DIMENSION OF DIVIDED CONFIGURATION IN INSULATOR**

The insulator was divided into 12 parts in order to analyze the heat transfer from the heat exchanger wall and the cyclone wall to the outer wall. Symmetrical dimensions were chosen to simplify the calculation. The symbols in Figure D5. are listed in Table 14.

**Table 14. Dimensions of Divided Configuration in Insulator**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Meter</th>
</tr>
</thead>
<tbody>
<tr>
<td>LA</td>
<td>0.1647</td>
</tr>
<tr>
<td>LB</td>
<td>0.1433</td>
</tr>
<tr>
<td>LC</td>
<td>0.137</td>
</tr>
<tr>
<td>LD</td>
<td>0.1638</td>
</tr>
<tr>
<td>LE</td>
<td>0.2266</td>
</tr>
<tr>
<td>LF</td>
<td>0.1867</td>
</tr>
<tr>
<td>DS</td>
<td>0.2508</td>
</tr>
<tr>
<td>RS</td>
<td>0.01</td>
</tr>
</tbody>
</table>


THERMAL CONDUCTIVITY (W/m°C)

The thermal conductivities for every part of burner were found in reference 6. The perlite thermal conductivity was given in reference 13. Table 15 shows the thermal conductivities of material used in the burner.

Table 15. Thermal Conductivity of Materials in the Burner

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Material</th>
<th>Location</th>
<th>Value (W/m.K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SS316</td>
<td>AISI 316</td>
<td>Cyclone, Heat Exchange</td>
<td>24.2</td>
</tr>
<tr>
<td>SS310</td>
<td>AISI 304</td>
<td>Chamber</td>
<td>25.4</td>
</tr>
<tr>
<td>AISI</td>
<td>AISI 1010</td>
<td>Outer Wall</td>
<td>63.9</td>
</tr>
<tr>
<td>PERLITE</td>
<td>Perlite</td>
<td>Insulator</td>
<td>0.1</td>
</tr>
</tbody>
</table>

MASS FLOW OF COMBUSTION AND COOLING AIR

At normal operating conditions, the mass flow of combustion air is in the range of 55-57 kg/hr. 56 Kg/hr was used in this analysis. Mass flow of the cooling air can be changed in the theoretical model in order to find the effect of cooling air rate on heat loss.

EMISSIVITY

The emissivity values of chamber wall, cyclone wall, and outer wall (white paint) were found in reference 6.

E2 = 0.65  
E3 = 0.65  
E6 = 0.925
THERMAL PROPERTY OF AIR

Air thermal properties were needed for air in the bed, cyclone, heat exchanger, and the air film between the outer wall and ambient environment. Since the thermal property of air is dependent on temperature, the temperature at each location was required. By using trial and error method, it was found that the bed temperature was about 1100K, 1000K at the heat exchanger, 1000K at the cyclone, and 320K at the air film between the outer wall and ambient.

The properties of air at bed temperature = 1100K

\[ K = 0.0715 \text{ W/m}^0\text{C} \]
\[ \mu = 449.0\text{E-7 N*s/m}^2 \]
\[ Pr = 0.728 \]

The properties of air at heat exchanger temperature = 1000K

\[ K = 0.0667 \text{ W/m}^0\text{C} \]
\[ \mu = 424.4\text{E-7 N*s/m}^2 \]
\[ Pr = 0.726 \]
\[ Cp = 1141 \text{ Kj/Kg*K} \]

The properties of air at cyclone temperature = 1000K

\[ K = 0.0667 \text{ W/m}^0\text{C} \]
\[ \mu = 424.4\text{E-7 N*s/m}^2 \]
\[ Pr = 0.726 \]

The properties of air at film temperature = 320K

\[ K = 0.02778 \text{ W/m}^0\text{C} \]
\[ \alpha = 25.46\text{E-6 m}^2/\text{s} \]
\[ \varphi = 17.9\text{E-6 m}^2/\text{s} \]
\[ Pr = 0.704 \]
\[ g = 9.8 \text{ m/s}^2 \]
APPENDIX F

COMPUTER PROGRAM FOR HEAT LOSS ANALYSIS
C PROGRAM TO CALCULATE HEAT LOSS FROM BURNER BODY, USEFUL
C HEAT AND OUT AIR TEMPERATURE BASED ON DIFFERENT COOLING
C AIR RATE (HLBURN)
C
REAL L, LA, LB, LC, LD, LE, LF
DIMENSION A(10,11), Q(10), QL(10)
DATA V1/0.0761/, V2/0.0841/, V3/0.0881/, V4/0.0901/
DATA V5/0.3028/, V6/0.3048
DATA V7/0.0996/, V8/0.1016/, V9/0.2774/, V10/0.2794/
DATA DCL/0.2508/, L/1.524/, LA/0.1647/, LB/0.1433/
DATA LC/0.137/, LD/0.1638/, LE/0.2266/, LF/0.1867/
DATA PI/3.14159/
DATA SS316/24.2/, SS310/25.4/, AISI/63.9/, PELITE/0.1/
DATA E2/0.65/, E3/0.65/, E6/0.925/
C THE PROPERTY OF AIR AT BED TEMPERATURE 1100K
BEDK=0.0715
VSBED=449.0E-7
PRBED=0.728
C THE PROPERTY OF AIR AT HEAT EXCHANGER TEMPERATURE 1000K
GAPK=0.0667
VSGAP=424.4E-7
PRGAP=0.726
CP=1141.
C THE PROPERTY OF AIR AT FILM TEMPERATURE 320K
FREEK=0.02778
ALPHA=25.46E-6
UN=17.9E-6
PRFREE=0.704
G=9.8
C THE PROPERTY OF AIR AT CYCLONE TEMPERATURE 1000K
CLONEK=0.0667
VISCLON=424.4E-7
PRCLONE=0.726
C ASSUME T2, T3 AND T6
T2=830.
T3=920.
T6=60.
C INPUT DATA
77 TAIR=30.0
COMBAIR=56.0
TBED=835.0
TCLONE=700.0
TEST=1.0
WRITE(6,*) 'INPUT AMBIENT TEMPERATURE (C) IF OTHER
# THAN 30.0'
WRITE(6,*) '
READ(5,*) TAIR
TAIRK=TAIR+273.
WRITE(6,*) 'INPUT COMBUSTION AIR RATE (KG/HR) IF OTHER
# THAN 56.0'
WRITE(6,*) '
READ(5,*) COMBAIR
WRITE(6,*) 'INPUT BURNER TEMP. --> BED, OUTLET (C)'
WRITE(6,*) 'IF BED OTHER THAN 835 OR OUTLET OTHER THAN

$700'
WRITE(6,*) '
READ(5,*) TEBD, TCLEONE
WRITE(6,*) 'INPUT COOLING AIR RATE (KG/HR)'
WRITE(6,*) '
READ(5,*) COOLAIR
WRITE(6,*) 'INPUT TOLERANCE TEMPERATURE (C) IF OTHER

# THAN 1.0'
WRITE(6,*) '
READ(5,*) TEST
RNBED=4*COMBAIR/(2*PI*V1*VISBED)
USBED=0.023*RNBED**0.8*PRBED**0.3
HBBED=USBED*BEBK/(2*V1)
RBBED=1/(2*PI*V1*HBBED)
RCL=ALOG(V2/V1)/(2*PI*SS310)
R1=RBBED*RCL
DH=(V3-V2)*2
AC=PI*(V3**2-V2**2)
RCOND2=ALOG((V2+V3)/(2*V2))/(2*PI*GAPK)
RCOND3=ALOG(2*V3/(V2+V3))/(2*PI*GAPK)
IF (COOLAIR .EQ. 0.0) GOTO 82
RN GAP=COOLAIR*DH/(AC*VISGAP)
US GAP=0.023*RN GAP**0.8*PR GAP**0.4
IF (RN GAP .LE. 2300.0) US GAP=7.54
HGAP=US GAP*GAPK/DH
RCONV2=1/(2*PI*DH*HGAP)
R2=RCONV2
RCONV1=1/(2*PI*DH*HGAP)
R3=RCONV3
R7=3600.*L/(COOLAIR*CP)
GOTO 81
82 R2=RCOND2
R3=RCOND3
R7=L**2/(2*GAPK*AC)
81 REXG=ALOG(V4/V3)/(2*PI*SS316)
RM4=ALOG(1.08*DCL/(2*V4))/(2*PI*PERLITE)
R4=RE XG+RM4
RN CLONE=4*COMBAIR/(2*PI*V7*VIS CLONE)
US CLONE=0.023*RN CLONE**0.8*PR CLONE**0.3
HCLO NE=USCLONE*CLONEK/(2*V7)
RCLO NE=1/(2*PI*V7*HCLONE)
RTANK=ALOG(V8/V7)/(2*PI*SS316)
RM5=ALOG(1.08*DCL/(2*V8))/(2*PI*PERLITE)
R8=RCLO N E+RTANK+RM5
P=PI*(V6+V10)+2*DCL
RCASE=(V6-V5)/(PAISI)
RM5A=LA/(PERLITE*(DCL-0.01))
RM5B=LB/(PERLITE*(DCL-0.01))
RM5C=LC/(PERLITE*(DCL-0.02))
RM5D=LO/(PERLITE*(DCL-0.02))
RM5E=2*ALOG(1.08*LE/0.02)/(PI*PERLITE)
RM5F=2*ALOG(1.08*LF/0.02)/(PI*PERLITE)
RM5=2/RM5A+2/RM5B+1/RM5C+1/RM5D+2/RM5E+2/RM5F
RM5=1/RM5

42 T6K=T6+273.
TM=(T6K+TAIRK)/2
RAL=(T6-TAIR)*L**3/(TM*UN*ALPHA)
FREE=1+(0.492/PRFREE)**0.5625
IF (RAL.GT. 1.0E9) GOTO 61
USFREE=0.68+0.67*RAL**0.25/FREE**0.4444
GOTO 62

61 USFREE=(0.825+0.387*RAL**0.16667/FREE**0.2963)**2

62 HFREE=USFREE*FREEK/L
RCONV6=1/(P*HFREE)
RAD6=1/(5.67E-8*P*E6*(T6K**2+TAIRK**2)*(T6K+TAIRK))
RO=1/(RCONV6+1/RAD6)
R5=RO+RCASE+RM5
EE=1/E2+(1-E3)/E3*(V2/V3)

41 T2K=T2+273.
T3K=T3+273.
RADGAP=EE/(5.67E-8*2*PI*V2*(T2K**2+T3K**2)*(T2K+T3K))
R6= RADGAP

C SOLVING INHOMOGENEOUS SETS OF LINEAR EQUATIONS
N=5
TA=TBED-TAIR
TB=TA
TC=TCLONE-TAIR
TD=TC
A(1,1)=R1+R4+R5+R6
A(1,2)=R1
A(1,3)=-R4
A(1,4)=R5
A(1,5)=-R6
A(1,6)=TA
A(2,1)=R1
A(2,2)=R1+R2+R7
A(2,3)=R7
A(2,4)=0.
A(2,5)=R2
A(2,6)=TB
A(3,1)=-R4
A(3,2)=R7
A(3,3)=R3+R4+R7+R8
A(3,4)=R8
A(3,5)=-R3
A(3,6)=TC
A(4,1)=R5
A(4,2)=0.
A(4,3)=R8
A(4,4)=R5+R8
A(4,5)=0.
A(4, 6) = TD  
A(5, 1) = -R6  
A(5, 2) = R2  
A(5, 3) = -R3  
A(5, 4) = 0.  
A(5, 5) = R2 + R3 + R6  
A(5, 6) = 0.  
DO 10 I = 1, N-1  
IV = I  
DO 20 J = I + 1, N  
IF (ABS(A(I, IV)) .LT. ABS(A(J, I))) IV = J  
20 CONTINUE  
IF (IV .EQ. I) GOTO 11  
DO 30 JC = 1, N+1  
TT = A(I, JC)  
A(I, JC) = A(IV, JC)  
A(IV, JC) = TT  
30 CONTINUE  
C ---------------------  
C GAUSS ELIMINATION  
C ---------------------  
11 DO 40 JR = I + 1, N  
IF (A(JR, I) .EQ. 0.) GOTO 40  
R = A(JR, I)/A(I, I)  
DO 50 KC = 1 + 1, N+1  
A(JR, KC) = A(JR, KC) - R*A(I, KC)  
50 CONTINUE  
40 CONTINUE  
10 CONTINUE  
IF (A(N, N) .EQ. 0.) GOTO 98  
A(N, N+1) = A(N, N+1)/A(N, N)  
DO 60 NV = N-1, 1, -1  
VA = A(NV, N+1)  
DO 70 K = NV+1, N  
VA = VA - A(NV, K)*A(K, N+1)  
70 CONTINUE  
A(NV, N+1) = VA/A(NV, NV)  
60 CONTINUE  
DO 80 I = 1, N  
Q(I) = A(I, N+1)  
QL(I) = Q(I)*L  
80 CONTINUE  
HL = QL(1) + QL(4)  
HLCOOL = QL(2) + QL(3)  
TOUT = TAIR + R7*(Q(2) + Q(3))  
T2O = TBED - (Q(1) + Q(2))*R1  
T3O = T2O - (Q(1) - Q(5))*R6  
T6O = TAIR + R6*(Q(1) + Q(4))  
DT2 = ABS(T2 - T2O)  
DT3 = ABS(T3 - T3O)  
DT6 = ABS(T6 - T6O)  
IF (DT2 .GT. TEST) T2 = T2O
IF (DT3 .GT. TEST) T3=T30
IF (DT2 .GT. TEST .OR. DT3 .GT. TEST) GOTO 41
IF (DT6 .GT. TEST/2) T6=T60
WRITE(6,102) R1,R2,R3,R4,R5,R6,R7,R8
102 FORMAT(1X,' R1   R2   R3   R4   R5
#R6   R7   R8'/1X,8F9.6/
WRITE(6,101) (QL(I),Q(I),I=1,N)
101 FORMAT(12X,2F14.6)
WRITE(6,103) T20,T30,T60,TOUT,HL,HLC
103 FORMAT(12X,'TEMPERATURE AT PT. 2 = ',F14.3/
#12X,'TEMPERATURE AT PT. 3 = ',F14.3/
#12X,'TEMPERATURE AT PT. 6 = ',F14.3/
#12X,'OUTLET TEMPERATURE OF COOLING AIR = ',F14.3/
#12X,'HEAT LOSS FROM BURNER BODY = ',F12.4/
#12X,'HEAT LOSS FROM COOLING AIR = ',F12.4)
M=0
WRITE(6,*) 'INPUT (1) FOR ANOTHER CASE OTHERWISE TO
#STOP'
WRITE(6,*) '
READ(5,*) M
IF (M-1) 99,77,99
98 WRITE(6,*) 'MATRIX IS SINGULAR'
99 STOP
END
APPENDIX G

THE CALCULATION OF HEAT LOSS FROM THERMOCOUPLES
A discussion with H. Keener revealed that heat loss from the thermocouples might be significant. In order to estimate the amount heat loss from the thermocouples, the following calculation was conducted.

KNOWN:
There were eight thermocouples, which can be considered as horizontal pipes (Length = 0.1016m, Diameter = 0.0254 m) with an outside surface temperature of 120° C. The ambient temperature was assumed to be 30° C.

PROPERTIES:
From reference 6, air (Tf = 348K): K = 0.03 W/m*K,
\( \nu = 20.92E-6 \text{ m}^2/\text{s} \)
\( \alpha_* = 29.90E-6 \text{ m}^2/\text{s} \)
Pr = 0.7
\( \beta = 2.874E-3 \text{ K}^{-1} \)

ANALYSIS:
The total heat loss of each pipe is
\[ q = q_{\text{conv}} + q_{\text{rad}} = h*A*(T_S-T_\infty) + E*A_**(T_S^4-T_\infty^4) \]
The convection coefficient may be obtained from following equation [reference 6. p. 398]
\[ \text{Nu} = \left( 0.60 + \frac{0.387 \text{Ra}^{1/6}}{[1+(0.559/\text{Pr})^{9/16}]^{8/27}} \right)^2 \quad 10^{-5} < \text{Ra}_D < 10^{12} \]
where
\[ \text{Ra}_D = \frac{g*\beta*(T_S-T_\infty)*D^3}{\alpha_* \nu} \]
\[ \text{Ra}_D = \frac{9.8 \times 2.874 \times 10^{-3} \times (120-30) \times 0.0254^3}{20.92 \times 10^{-6} \times 29.9 \times 10^{-6}} = 66408 \]

Hence
\[ \text{Nu} = \left( 0.60 + \frac{0.387 \times 66408^{1/6}}{[1+(0.559/0.7)^{9/16}]^{8/27}} \right)^2 = 6.59 \]

and
\[ \frac{K \times \text{Nu}}{D} = \frac{0.03 \times 6.59}{0.0254} = 7.78 \]

The total heat loss per pipe is then
\[ q = 7.78 \times (\pi \times 0.0254 \times 0.1016) \times (120-30) + 0.85 \times (\pi \times 0.0254 \times 0.1016) \times 5.67 \times 10^{-8} \times (393^4 - 303^4) \]
\[ = 7.84 \text{ Watts} \]

The total heat loss from eight thermocouples is
\[ q = 7.84 \times 8 = 62.7 \text{ Watts} \]