ANALYSIS OF A FEEDBACK CONTROL SYSTEM FOR A
FLUIDIZED BED CORNCOB COMBUSTOR

A Thesis
Presented in Partial Fulfillment of the Requirements
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approved by

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Chapter I
INTRODUCTION

The 1970 fuel price increases and fuel shortages have led to an increased awareness of the exhaustibility of fossil fuels and a quest to develop alternative fuel sources. For agriculture, biomass — a renewable and probably the first source of fuel used by man — has recently regained importance as an alternative energy source.

For biomass to become a viable energy source in agriculture, the energy content of these materials must be transformed from its low grade status to a higher grade, more usable energy form, in a non-polluting fashion.

The efficient conversion of biomass materials to usable heat energy requires a different technology than that developed for traditional fossil fuels and is discussed by Claar, et al., (1980), Jenkins (1980), and Payne (1980). The major reasons for requiring a different technology are as follows:

1. Usually the material is not free flowing, and is often not in uniform discrete particles, thus it requires complex handling and feeding equipment.

2. The material contains a high proportion of volatile material which vaporizes very rapidly and can cause much smoke and loss of energy if combustion is not controlled and completed.

3. Chemicals exist in biomass materials which combine with excess oxygen to form complex ash compounds having low melting points. These compounds cause ash fouling and can also cause severe erosion/corrosion of metal components. The combustion temperature in the burner needs to be controlled closely, for at high temperatures ash in the fuel begins to melt and particles
adhere to other particles and surfaces causing slagging and eventual plugup.

4. The material has high and variable moisture content as grown and may require natural or artificial drying before it is used as a fuel in conventional furnaces.

Payne (1982), addressed the technology for burning biomass. He stated that the biomass combustor must be:

1. Clean burning - products of combustion must not contaminate crops being dried.
2. Automated - the desired temperature must be automatically maintained.
3. Reliable - the system must perform without close supervision.
4. Economical - the system must be cost competitive from both construction and operation standpoints.

The most promising method for converting biomass to heat is fluidized bed technology (Groves, 1979; Anderson, 1979). Researchers at OARDC-OSU, Wooster, Ohio started to develop a home or small size commercial burner system using fluidized bed technology and corn cobs as the fuel source (Keener, et al., 1981, 82, 83). An OARDC fluidized-bed furnace was built in 1981. An automatic control system was added in the fall of 1982 to turn the burner system off/on upon call by room thermostat and/or sand bed temperature. The furnace is still being tested and improved.

This thesis presents a study/stability analysis of the OARDC furnace burner.
Chapter II

OBJECTIVES

The overall objective of this study was to evaluate the stability and performance of the control system of the OARDC prototype fluidized-bed combustor. The prototype burner is presently equipped with an on/off fuel rate and on/off (2 speed) cooling air rates, both triggered by a set point temperature in the burner as the basis for its feedback control system.

Specific objectives of this research were to:

1. Develop mathematical equations describing the OARDC prototype fluidized bed combustion system.

2. Develop a computer model simulating the performance of the burner based on the theoretical equations.

3. Verify the computer model using experimental results.

4. Perform a stability analysis of the fluidized-bed combustor (FBC) system as the basis for studying possible types of control systems for the burner.

5. Investigate other possible types of control systems for the burner.

6. Recommend an efficient control system for the burner based on the results of the study.
Chapter III
REVIEW OF LITERATURE

An increased awareness of the exhaustibility of fossil fuels has created interest in more fully utilizing biomass as an alternative energy source, especially on the farm. For biomass to become a viable energy alternative in agricultural production, small scale methods are needed to convert the low-grade energy in these materials into more usable energy forms. This has appeal from an institutional aspect and is needed if farmers are to use their own energy sources to replace fossil fuels.

3.1 FLUIDIZED-BED COMBUSTION

Fluidized bed combustors have been found to work well with biomass materials such as corncobs because of their unique operating characteristics.

A fluidized bed consists of a chamber filled with sand-like particles usually supported by a porous floor. Air is forced through the porous floor under low pressure at a velocity which lifts and suspends the particles in the chamber in a churning mass. This churning mass absorbs and stores heat. The turbulence of the churning bed keeps the temperature uniform throughout the bed. When fuel is introduced into the FBC, it rapidly absorbs heat from the solid particles of the bed, attains ignition temperature and burns. The large quantity of inert
particles relative to fuel particles maintains stable temperatures. The advantageous characteristics of a fluidized bed are:

1. Isothermal bed temperature is maintained due to rapid mixing of a small amount of solid fuel relative to a large mass of solid bed material.

2. Accurate control of bed temperature within a narrow range below the ash fusion point of the solid fuel is possible due to uniform temperature in the bed.

3. Wide variations in type, composition, and moisture content of the feedstocks are acceptable because of the thermal inertia of the bed particles.

4. High rates of heat transfer from bed material to solid fuel and to heat exchange surfaces in contact with the agitated bed are possible.

5. Excellent restart for intermittent processes because of thermal inertia of bed (if heat transfer out is controlled).

Inherent limitations of FBCs also exist. They include:

1. Process demands expenditure of power for fluidization.

2. Operating rates are limited to those within the range over which the bed may be fluidized.

3. Hard to maintain a stable bed particle size distribution due to attrition and fines losses through elutriation.

There are two major applications of fluidized beds in converting biomass fuels to useful energy forms - gasification and combustion. Gasification is the releasing of volatiles from the fuel in the presence of insufficient oxygen for complete combustion. These gases are then further burned in another chamber to obtain the heat. During direct combustion, moisture in the fuel is evaporated, the volatile matter is gasified and burned, and finally the fixed carbon is burned. The products of complete combustion are heat energy, carbon dioxide, ash, and water.
LePori (1982), investigated direct biomass combustion in conventional furnaces and concluded that it is not the best method for obtaining usable energy forms. His experiments have shown that a fluidized bed can effectively convert biomass to energy forms which can be used in agriculture.

3.2 **EXISTING FLUIDIZED-BED FURNACES**

Gasification furnaces and two-stage combustion furnaces (both a gasification and combustion step) have been used for converting crop or forest biomass into thermal energy (Richey, et al., 1982; Payne, et al., 1980; Groves, 1979; Claar, et al., 1981; Groves, et al., 1979). These furnaces were updraft and downdraft gasifiers which have stationary beds and fluidized bed systems. This thesis will be concerned with direct combustion in a fluidized bed and, in particular, the existing unit at OARD (Keener, et al., 1981). The initial research on fluidized bed combustion of coal was carried out with the objective of developing very large water-tube boilers for power generation. The size of industrial boilers range from 500 kW to 50 MW. Howard (1983) described several existing industrial fluidized bed boilers. Most of the described industrial boilers had start up and load following fully automated (Babcock and Rists, Wires, and Cables units) based on preset air flow rates and corresponding fuel feed rates. Several were microprocessor controlled (Wallsend unit).

The National Coal Board started a stage of development of fluidized bed furnaces for direct contact drying. The first unit was a 15 MW furnace installed in 1979 to supply gas at 100°C for clay drying and
cement manufacturing. A second industrial unit of 7.5 MW was installed in 1981 for drying stone at a quarry. In these furnaces, the beds are operated as partial-gasifiers having substantial above bed combustion with secondary air. Other units such as this have been developed by Energy Equipment Co.

In order to extend the application of fluidized bed combustion furnaces to the majority of industrial drying processes, it is necessary to develop units which produce clean heated air. The design of the furnace and start up control system should be as simple as possible for reliability, and preferably built with standard components for easy maintenance. The Worsley furnace for grass drying (Howard, 1983), is one such furnace. Present Worsley Co. furnaces have a constant gas flow rate and inlet temperature and are controlled by automatically modulating the wet product feed rate to maintain the required exhaust temperature.

Howard (1983) stated also, "in order to exploit fully the advantage of FBC's in a novel heater design, it will be necessary to utilize the high heat transfer coefficient to immersed surfaces." Work in this area is presently being conducted by groups such as Fluidyne Engineering Corporation and Battelle Research Institute. Makanski and Schwieger (1982) discuss current designs of direct combustion FBCs.

The burner unit at the OARDC (patent applied for by OARDC) (Figure 1), consists of a six-inch diameter stainless steel pipe as the combustion chamber and a unique fluidized bed-to-air heat exchanger system which produces a clean high-temperature air stream for heating purposes.
The system is small (22 to 30 kW) and was designed from computer analyses on the chemical balances of the combustion process to determine heat production for different air-to-fuel ratios so that the specifications for the fluidizing pump, vibrating feeder for supplying cobs, and sizing of the sand bed (both particle size and depth) could be done. Further analysis on heat transfer from the FBC has led to specifications on the design of the vessel walls and cooling air and insulation requirements (Keener, et al., 1981). Results of early 1982 tests showed that the unit burns corncobs cleanly, delivers clean heated air efficiently and requires minimal maintenance over long periods of time if properly operated (Keener, et al., 1982). The tests confirmed that bed temperatures must not be operated above 760°C or ash from the burned cobs will melt and stick to the sand causing the burner to plug. These tests also showed that the combustion of cobs must be done above 650°C so that combustible gases are fully burned and maximum heat obtained (Keener, et al., 1982).

3.3 PREVIOUS WORK IN AUTOMATIC CONTROL OF FLUIDIZED BEDS

Little information is available in the literature concerning automatic control of small scale fluidized bed combustors/gasifiers. Payne, et al. (1982) recommended that the furnace be reliable and automated to be accepted in agricultural practices. Automation means the desired temperature is maintained automatically, and the fuel is fed to the combustor automatically, and is not contaminated by the exhaust. It appears that few small scale furnaces have been automated.
Figure 1. Schematic of Fluidized Bed Combustor.

Some work on fixed bed gasifiers, however, have addressed the question of control. Richey, et al. (1981), developed a downdraft-channel type gasifier furnace using cobs for fuel that was successfully used to dry shelled corn. He stated that it would be difficult to fully automate the furnace because of startup and other procedures required. Payne, et al. (1982), proposed a control algorithm for two-stage combustion of biomass fuels to obtain a clean exhaust for direct crop drying. They suggested that different control algorithms should be utilized for startup, shut down, and process interruption events. As of the summer of 1984, they had developed a mathematical model for the gasifier-combustor and were experimenting with the software for a microcomputer controller.
Most fluidized bed gasifiers have been tested over short periods of time and were not operated continuously. Manual operation was sufficient; automatic control was not necessary. Such was the case for the OARDC unit until late 1982.

In fluidized bed furnaces, the use of a thermal control system is similar to that of a typical home heating furnace. Miles (1965) described several control functions common to all thermal systems:

1. Control the temperature of a condition within certain limits of accuracy.

2. Control the furnace in such a manner that it is operated safely and economically.

3. Provide continuous monitoring of a temperature condition that would be difficult or impossible by manual operation.

4. Provide a safety margin so that on excessive rise in temperature, the sequence of operations providing the source of heat is brought either to a complete standstill until manually reset, or to a temporary halt until safe temperature conditions again prevail.

As mentioned previously, the ash in biomass fuels have low melting point temperatures. For instance, corn cobs used as the fuel without additives, must be burned below 760°C for the bed to operate properly (Keener, et al., 1982).

If the feed rate of fuel to the furnace is the manipulated variable in the control scheme, it is important to also control the air-to-fuel ratio in the furnace. Claar, et al. (1981), stated that excess air should be limited in a biomass furnace since excess air cools the bed and slows the combustion reaction rate, reduces overall efficiency, and increases flue gas velocities which carry partly burned particles out of the furnace. It may also accelerate corrosion in the system due to the
reactive oxygen in the hot air. On the other hand, too little air will cause unburned fuel and smoke, and result in a reducing atmosphere and free hydrogen which can also lead to high corrosion rates. Thus, the air-to-fuel ratio must be controlled to operate the burner safely and economically.

The present control system on the OARDC unit meets all of the criteria of Miles (1965). It is based on the bed temperature and it manipulates the fuel rate on-off with relays. A relay map of its operation is shown in Figure 2.

During the preheat stage, the preheater is warming the air which is fluidizing the sand bed. At 550°C, the preheater is turned off, the air lock turns on, and the cob feeder turns on, thus beginning the operational mode. As the fuel burns and the bed continues to warm, the cooling air turns on, first at low speed, then later at high speed as indicated by temperature level. From this point on, the fuel cycles on/off as required by temperature level.

3.4 DESIGN OF THERMAL CONTROL SYSTEMS

Just as in any engineering design situation, the design of feedback control systems begins with a statement of specifications to be met (Doebelin, 1962). Specifications in control systems are concerned with how well the system can reproduce commands and reject disturbances. Figure 3 (Doebelin, 1962) presents the standardized nomenclature which is used in discussing these specifications.
Development of the control diagram in this standard form for a real system requires that a mathematical model in the form of a transfer function be developed. Directly arriving at the transfer function by writing differential equations can be extremely difficult because of the thermal properties and heat transfer parameters that need to be evaluated. An alternative, or supplementary approach, is to fit the system response to a step input with an empirical equation. Yost (1966) designed a thermal control system for a muffle furnace in this manner.

Once the transfer functions are developed for the controlled system and the feedback elements, various control elements can be theoretically installed and described by a transfer function. A stability analysis
can then be made of the entire system. If the system is controlled in an on/off manner, as many furnaces are, the frequency of the limit cycle can be predicted.

Figure 3. Standard Symbology For Feedback Control Systems. (Doebelin, 1962)
Chapter IV
THEORETICAL METHODS

In developing a control diagram for the fluidized bed combustor, a mathematical model or transfer function of the combustor needs to be developed which will give the response of the system to variations in different variables. The control system can then be designed to cope with this load response.

4.1 REPRESENTATION OF PROTOTYPE BURNER

The prototype burner at the OARDC (patent applied for) is shown in Figure 4. The burner is represented as coaxial cylindrical shells (Figure 5) consisting of the bed of sand and its containing stainless steel pipe, the heat exchanger and its containing wall, the perlite insulation, and the outer wall. Table 1 lists the dimensions of the model. Figure 6 is a cutaway view of the burner showing the parameters used in this theoretical study. A complete summary of the parameters is presented in Appendix C.

Figure 6 also illustrates how the fuel and air flows enter as well as the heat flow patterns. The fluidizing/combustion air enters the bottom of the burner under the sand bed. The excess air and products of combustion pass through the burner and exit out the top. The cooling air enters the heat exchanger at the top, passes through, and exits from the bottom.
The heat flow is directed radially outward as well as vertically; however, it will be assumed one dimensional in the radial direction for this problem. The bed temperature will be assumed uniform across the burner due to the turbulent mixing of the bed and the high rate of heat transfer between particles. The temperature reading of the thermocouple used to control the furnace will be taken as the sand temperature at Location 2 in the bed (Figure 14).
Figure 5. Burner Represented as Coaxial Shells.

Figure 6. Pulled-Apart View of Burner.
Table 1. Dimensions of Model.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Radial Distance from Center of Bed to:</th>
<th>Feet</th>
</tr>
</thead>
<tbody>
<tr>
<td>r₁</td>
<td>inner side of chamber wall</td>
<td>.2527</td>
</tr>
<tr>
<td>r₂</td>
<td>outer side of chamber wall</td>
<td>.2760</td>
</tr>
<tr>
<td>r₃</td>
<td>inner side of heat exchanger wall</td>
<td>.2891</td>
</tr>
<tr>
<td>r₄</td>
<td>outer side of heat exchanger wall</td>
<td>.2956</td>
</tr>
<tr>
<td>r₅</td>
<td>inner side of outer wall</td>
<td>.9635</td>
</tr>
<tr>
<td>r₆</td>
<td>outer side of inner wall</td>
<td>.9700</td>
</tr>
<tr>
<td>L</td>
<td>Length of burner</td>
<td>5.</td>
</tr>
</tbody>
</table>

The length of the burner will be considered 5 feet to simplify the analysis by eliminating the end conditions (the cyclone and ash collection). Wall 1, the heat exchanger, wall 2, the insulation, and wall 3, will be treated as lumped masses whose temperatures are measured at their centers: T_W₁, T_EX, T_W₂, T_INS and T_W₃, respectively.

4.2 DEVELOPMENT OF THE MATHEMATICAL MODEL

To develop the differential equations which describe the dynamic response of the system, energy balances were written for each shell according to the equation

\[
\text{(energy stored)} = \text{(energy in)} - \text{(energy out)} + \text{(heat generation)} \quad (4.1)
\]

For the inner shell, the sand bed, the following equation was developed:

\[
\frac{dT_B}{dt} = \sum \frac{mc_p}{C} \cdot \left(T_{B_i} - 537\right) - \sum \frac{mc_p}{C} \cdot \left(T_{B_i} - 537\right) + \Delta H_f - Q_{\text{wall}} \quad (4.2)
\]
where \( m \) = mass of sand, lb
\( c_{psand} \) = heat capacity of the sand, Btu/lb °R
\( TB \) = burner temperature, °R
\( \mathcal{O} \) = time, hr
\( TB_i \) = temperature of entering fuel, °R
\( \Delta H_f \) = heat generated, Btu/lb
\( Q_{wall} \) = heat loss out through walls, Btu/hr
\( \Sigma n_{c_{p_{in}}} \) = sum of moles x heat capacity of items in
\( \Sigma n_{c_{p_{out}}} \) = sum of moles x heat capacity of items out
\( Q_{wall} = h_1(2\pi r_1) L(TB - TW_{1a}) \) \hspace{1cm} (4.3)

where \( h_1 \) = heat transfer coeff. between bed and wall 1
\( r_1 \) = distance from center of bed to inner wall 1, ft
\( L \) = length of bed, ft
\( TW_{1a} \) = temperature of inner wall 1, °R

To eliminate \( TW_{1a} \) from the equation, assume a lumped mass at the center of wall 1:
\( Q_{wall} = (TW_{1a} - TW_1)*(2\pi k_1) L \ln((r_1 + .5*\tau_1)/r_1) \) \hspace{1cm} (4.4)

(Eckert and Drake, 1972, pp. 70-71)

Solving eq. 4.3 and 4.4 simultaneously for \( TW_{1a} \) in terms of \( TB \) and \( TW_1 \) gives:

\[
TW_{1a} = \frac{h_1(2\pi r_1)L}{(2\pi k_1 L)/\ln((r_1 + .5\tau_1)/r_1) + h_1 2\pi r_1 L} TB + \]
\[
\frac{2\pi k_1 L/\ln((r_1 + .5\tau_1)/r_1)}{2\pi k_1 L/\ln((r_1 + .5\tau_1)/r_1) + h_1 2\pi r_1 L} TW_1 \]

(4.5)

Renaming the coefficients of \( TB \) and \( TW_1 \) as \( \gamma_1 \) and \( \gamma_2 \), respectively,
\[ T_{W1a} = \dot{q}_1 TB + \dot{q}_2 T_{W1} \quad (4.6) \]

Substituting 4.6 into 4.2, dividing by \( mc_{psand} \) and renaming coefficients, the differential equation for the burner can now be written as

\[ \frac{dT_B}{d\Theta} = \alpha_1 (TB_1 - 527) - \alpha_2 (TB - 527) + Q_{gen} - \alpha_3 TB + \alpha_4 T_{W1} \quad (4.7) \]

where
\[ \alpha_1 = \frac{\dot{m}_{c, pin}}{(mc_{psand})}, \text{ l/hr} \]
\[ \alpha_2 = \frac{\dot{m}_{c, pout}}{(mc_{psand})}, \text{ l/hr} \]
\[ \alpha_3 = h_1 (2\pi r_1)(1-\delta_1)/(mc_{psand}), \text{ l/hr} \]
\[ \alpha_4 = h_1 (2\pi r_1)(\delta_2)/(mc_{psand}), \text{ l/hr} \]
\[ Q_{gen} = \Delta H_f*\text{fuelrate}/(mc_{psand}), \text{ °R/hr} \]

Similarly, equations (see Appendix A) were developed for each shell of the model. Results are:

\[ \frac{dT_{W1}}{d\Theta} = -\alpha_8 TB - \alpha_9 T_{W1} + \alpha_{10} T_{W2} + \alpha_{11} T_{EX} \quad (4.8) \]

\[ \frac{dT_{EX}}{d\Theta} = -\alpha_{12} T_{EX} + \alpha_{13} T_{W1} + \alpha_{14} T_{W2} + \alpha_{15} T_{EX_i} \quad (4.9) \]

\[ \frac{dT_{W2}}{d\Theta} = \alpha_{16} T_{EX} + \alpha_{17} T_{W1} - \alpha_{18} T_{W2} + \alpha_{19} T_{INS} \quad (4.10) \]

\[ \frac{dT_{INS}}{d\Theta} = \alpha_{20} T_{W2} - \alpha_{21} T_{INS} + \alpha_{22} T_{W3} \quad (4.11) \]

\[ \frac{dT_{W3}}{d\Theta} = -\alpha_{23} T_{W3} + \alpha_{24} T_{INS} + \alpha_{25} T_{\infty} \quad (4.12) \]

In deriving equations 4.8, 4.9, and 4.10, heat transfer by radiation was incorporated into the equation by linearization of the
temperature term \( T^4 \) using the Taylor series expansion (see Appendix A).

\[
T^4 = T_0^4 + 4T_0^3 (T - T_0)
\]  

(4.13)

Coefficients \( \alpha_1 \) through \( \alpha_{25} \) are described in Appendix B.

4.3 SOLVING FOR TRANSFER FUNCTION

To solve the set of 6 linear differential equations developed for TB and TEX as a transfer function, the equations were written in matrix form and solved using matrix algebra. To avoid a 6th-order differential equation for the system the transient heat storage in wall 2 and wall 3 were assumed to be negligible because their total heat capacity was small compared to that for the burner. The same was assumed to be true for the insulation. The validity of these assumptions based on laboratory and computer simulation results is discussed in the summary.

Figure 7 is a matrix of the differential equations formed using the coefficients of the variables TB, TW₁, TEX, TW₂, TINS, TW₃ and F, the forcing function. Any of the variables can be solved for by replacing the column representing the variable in question with the column representing the forcing function and solving the new matrix.

Please note that the coefficients of all the temperatures in the equations are listed in Figure 7. The assumption of no storage in wall 2, wall 3, and the insulation causes the underlined terms to disappear from the equations. However, heat storage capacities are still required to evaluate \( \alpha_{16} \) through \( \alpha_{25} \) in the equations as now construed.
<table>
<thead>
<tr>
<th>TB</th>
<th>TW₁</th>
<th>TEX</th>
<th>TW₂</th>
<th>TINS</th>
<th>TW₃</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
<td>(D + \alpha_3)</td>
<td>(-\alpha_4)</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0 (537(\alpha_2 - \alpha_1)) + Q_{gen} + \alpha_1 TB_{i1}</td>
</tr>
<tr>
<td>(-\alpha_8)</td>
<td>(D + \alpha_9)</td>
<td>(-\alpha_{11})</td>
<td>(-\alpha_{10})</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>
| 0    | \(-\alpha_{13}\) | \(D + \alpha_{12}\) | \(-\alpha_{14}\) | 0    | 0    | \(
| \alpha_{15}T_{EX,i}\) |
| 0    | \(-\alpha_{17}\) | \(-\alpha_{16}\) | \(D + \alpha_{18}\) | \(-\alpha_{19}\) | 0    | 0               |
| 0    | 0    | 0    | \(-\alpha_{20}\) | \(D + \alpha_{21}\) | \(-\alpha_{22}\) | 0               |
| 0    | 0    | 0    | 0    | \(-\alpha_{24}\) | \(D + \alpha_{23}\) | \(\alpha_{25}T_{\infty}\) |

Figure 7. Coefficients of Temperatures in Differential Equations.

To solve for bed temperature, TB, the following matrices were solved.

\[
\begin{align*}
\begin{bmatrix}
537(\alpha_2 - \alpha_1) \\
+ Q_{gen} + \alpha_1 TB_{i1} \\
-\alpha_4 \\
0 \\
D + \alpha_9 \\
-\alpha_{11} \\
-\alpha_{10} \\
0 \\
\alpha_{15}T_{EX,i} \\
\alpha_{13} \\
D + \alpha_{12} \\
-\alpha_{14} \\
0 \\
\alpha_{17} \\
-\alpha_{16} \\
\alpha_{18} \\
\alpha_{19} \\
0 \\
\alpha_{25}T_{\infty} \\
0 \\
0 \\
\alpha_{24} \\
\alpha_{23}
\end{bmatrix}
\end{align*}
\]

\[
TB = \frac{1}{D} \begin{bmatrix}
D + \alpha_3 + \alpha_3 \\
-\alpha_8 \\
D + \alpha_9 \\
\alpha_{11} \\
\alpha_{10} \\
0 \\
\alpha_{13} \\
D + \alpha_{12} \\
-\alpha_{14} \\
0 \\
\alpha_{17} \\
-\alpha_{16} \\
\alpha_{18} \\
-\alpha_{19} \\
0 \\
0 \\
0 \\
-\alpha_{20} \\
\alpha_{21} \\
-\alpha_{22} \\
0 \\
0 \\
0 \\
-\alpha_{24} \\
\alpha_{23}
\end{bmatrix}
\] (4.14)
Similarly, to solve for \( \text{TEX} \), the matrix \( [D_2] \) was solved where

\[
[D_2] = \begin{bmatrix}
D + \alpha_2 + \alpha_3 & -\alpha_4 & +Q\text{gen} & 0 & 0 & 0 \\
-\alpha_8 & D + \alpha_9 & 0 & -\alpha_{10} & 0 & 0 \\
0 & -\alpha_{13} & \alpha_{15,\text{TEX}_1} & -\alpha_{14} & 0 & 0 \\
0 & -\alpha_{17} & 0 & -\alpha_{18} & -\alpha_{19} & 0 \\
0 & 0 & \alpha_{25,\text{TE}} & -\alpha_{20} & \alpha_{21} & -\alpha_{22}
\end{bmatrix}
\]

\[
\text{TEX} = \frac{[D_2]}{[D]} = \begin{bmatrix}
0 & 0 & 0 & 0 & -\alpha_{24} & \alpha_{23}
\end{bmatrix}
\]

(4.15)

The solutions to \( [D_1] \), \( [D_2] \), and \( [D] \) are given in Appendix D. Both \( \text{TB} \) and \( \text{TEX} \) are solved for separately because of the desire to use both in control of the burner; \( \text{TB} \) from the standpoint of fuel combustion efficiency and \( \text{TEX} \) from the standpoint of operating a hot gas turbine (Henry, et al., 1984). A standard control block diagram can now be constructed.

4.4 Control Diagrams

The control diagram using \( \text{TB} \) as the controlled variable is shown in Figure 8 and that using \( \text{TEX} \) is shown in Figure 9. The disturbances to the system are seen to be the temperature of the cooling air in, \( \text{TEX}_1 \), the temperature of the surroundings, \( T \), the temperature of the fuel and fluidizing air into the bed, and the reference temperature used.

The transfer function for the control element, \( G_n(D) \), is left general so it can be replaced by on/off, proportional, or whatever. The term \( e^{-T_1D} \) represents the dead time that it takes for the fuel to go
from the vibrating feeder to the point of burning in the bed. $Q_{gen}$ is the amount of heat being generated from combustion. The transfer function for the feedback elements which in this case are thermocouples, has a first order lag term in it, $\tau_2$, due to the nature of the well it is placed in. Previously, as seen in Figure 9, the feedback element transfer function is unity because the thermocouple is not inserted in a well in the heat exchanger. Currently, $TEX$ is not used for control, and further analysis of this mode of control is left to future studies. $Te$ is the error signal, that is, the difference between the bed set point temperature and the bed temperature being sensed.
FIGURE 8. CONTROL DIAGRAM BASED ON BED TEMPERATURE AS CONTROLLED VARIABLE.
FIGURE 9. CONTROL DIAGRAM BASED ON EXCHANGER TEMPERATURE AS THE CONTROLLED VARIABLE.
4.5 STABILITY ANALYSIS

Doebelin (1962) states:

"... In the application of a linear time-invariant mathematical model to the analysis of control systems leads to the prediction that in absolutely unstable systems the controlled variable approaches infinity." This is occurring because as the controlled variable becomes larger, the system equations become nonlinear. This results in the system either coming to a dead stop or going into a sustained oscillation of finite amplitude which is called a limit cycle and it is a feature of nonlinear systems. "In some nonlinear systems, such as an on/off controller in a home heating system, it is known beforehand that the system will oscillate continuously in a limit cycle during its normal operation." This system need not be called unstable since it is performing its task according to specification.

The on/off controller of the automatic burner, therefore, has a limit cycle and the system could be called unstable. The question of stability; however, was investigated here for several reasons:

1. To gain insight into how the system is expected to behave by predicting its limit cycle.

2. To determine how stable the system is. When a system can not be adequately approximated as linear, stability is no longer a system characteristic and may depend on the form and magnitude of commands and disturbances. So far, no rigorous evaluation of stability has been performed, which is important before installing the system in the field.

3. To investigate the value of a proportional control system, as opposed to an on/off control system, on the burner performance.

The describing function method approach was used to study the nonlinear on/off system. It is an approximate method and its predictions are not exact; however, good results are often obtained.

In a describing function analysis, a sinusoidal input is applied to the nonlinearity. The output is periodic with the same period as the input but it contains many harmonics in addition to the fundamental component. Because most control systems act as low pass filters, only
the fundamental component is used. The describing function is defined as the ratio of the fundamental component of the output of the nonlinearity to that of the sinusoidal input.

The on/off controller is assumed to work as shown in Figure 10. The error signal, $te$, ranges from $-15$ to $+15^\circ C$ and the fuel is either on or off (Note: the value $d = 30$ could just as easily be $d = 10$). This on/off signal is represented as $\pm 1/2$ in Figure 10. This comes about by assuming that the system has forced heating and forced cooling although the actual system has forced heating and free cooling. If the burner is off, it doesn't come on until the error signal reaches $30^\circ C$. When it comes on, it stays on until the error is eliminated, at which point it turns off and the cycle begins again.

![Figure 10. Behavior of On/Off Controller.](image-url)
Now application of a sinusoidal input to this on/off controller gives the nonlinear response shown in Figure 11.

\[ x(t) = A_i \sin(\omega t) \]

The output is

\[ y(t) = \begin{cases} 
-1/2 & 0 \leq \omega t \leq B \\
+1/2 & B \leq \omega t \leq \pi + B 
\end{cases} \]

where \( B = \sin^{-1} \frac{d}{A_i} \)

which is the time where \( d = A_i \sin(\omega t) \)

Figure 11. Nonlinear Response of On/Off Controller.
Results of a Fourier Series on $y(t)$, the output, gives for the fundamental component

$$4b \sin(wt)/\pi$$ \hspace{1cm} (4.16)

Using this output results in a describing function which is dependent on the amplitude of the input signal.

$$G_n(D) = \frac{4b}{\pi A_1}$$ \hspace{1cm} (4.17)

For predicting the limit cycle of the on/off controller, the characteristic equation

$$G_1G_2H + 1 = 0$$ \hspace{1cm} (4.18)

from the control diagram is solved.

To obtain the characteristic equation define:

$$G_1 = G_n e^{-\tau D} Q_{gen}(MD^2 + ND + P)$$ \hspace{1cm} (4.19)

$$\frac{1}{aD^3 + bD^2 + cD + e}$$ \hspace{1cm} (4.20)

$$H = \frac{1}{\tau 2D + 1}$$ \hspace{1cm} (4.21)

where $a$, $b$, $c$, $e$, $M$, $N$, $P$, and $Q_{gen}$ are as defined in Appendix D.

The deadtime $e^{-\tau D}$ is approximated as $-\tau D + 1$, resulting in

$$G_1G_2H = \frac{G_n(-\tau D + 1)(Q_{gen})(MD^2 + ND + P)}{(\tau 2D + 1)(aD^3 + bD^2 + cD + e)}$$ \hspace{1cm} (4.22)
From the control diagram, one can write

\[(1+G_1 G_2 H)TB = G_1 G_2 \text{Tr} + G_2 (\text{TEX}_1 N_1 + \text{TE}_N N_2 + N_3 (\text{TB}_1 N_3 + 537 N_4))\]  \hspace{1cm} (4.23)

where

\[N_1 = \alpha_{15} R\]

\[N_2 = \alpha_{25} (SD + T)\]

\[N_3 = \alpha_1\]

\[N_4 = \alpha_2 - \alpha_1\]

\[N_5 = MD^2 + ND + P\]

The resulting characteristic equation is

\[\tau_2 a d^4 + (\tau_2 b + (G_2 Q + LM)) d^3 + (\tau_2 c + (G_2 Q + LN) + G_2 QM) d^2 + (\tau_2 e + c - G_2 QLP + G_2 QN) d + (e + G_2 QP) = 0\]  \hspace{1cm} (4.24)

Equation 4.18 can now be rewritten as

\[\frac{(-\tau_2 D + 1)}{(\tau_2 D + 1)} (Q_{gen}) (MD^2 + ND + P) = -1\]  \hspace{1cm} (4.25)

Figure 12 shows the control diagram with the describing function representing the on/off controller.

Both sides of Eq. 4.25 can be plotted. If the two functions intersect, a limit cycle is implied. The frequency at the intersection point is the limit cycle frequency and this point is used to predict the limit cycle magnitude.
Figure 12. Modified Control Diagram.

Figure 13 is a representative plot of both sides of Eq 4.25 showing how the two functions intersect in the third quadrant. The solution values are given on the figure for d-values of 10 and 30. Both cases predict limit cycles. For d = 10, the predicted limit cycle frequency is .41 cycles/min and the amplitude of the limit cycle is 33°C. For d = 30, the predictions are .26 cycles/min and 104°C.
FIGURE 13. PLOT OF EQUATION 4.25.
For a proportional controller, rather than an on/off controller, a Routh Criterion stability analysis was performed using the characteristic equation of the system (4.24). *K* was used to represent the controller and substituted into the characteristic equation for *G_n*. Using a fixed fuel rate of 18 lb/hr the Routh Criterion was set up as follows:

\[
\begin{array}{ccc}
A & C & E \\
B & D & 0 \\
(BC-AD)/B & E & 0 \\
((BC-AD)/B)D-BC & 0 & 0 \\
(BC-AD)/B & 0 & 0 \\
E & 0 & 0 \\
0 & 0 & 0 \\
\end{array}
\]

where \( A = \tau_2 a \)
\[
B = \tau_2 b + a - KQ_{gen} \tau_1 M
\]
\[
C = \tau_2 c + b - KQ_{gen} \tau_1 N + KQ_{gen} M
\]
\[
D = \tau_2 e + c - KQ_{gen} \tau_1 P + KQ_{gen} N
\]
\[
E = e + KQ_{gen} P
\]

where *a*, *b*, *c*, *d*, *e*, *Q_{gen}*, *M*, *N*, and *P* are as defined in Appendix D. *K* was allowed to vary from 1 to 1000 hr\(^{-1}\) until a sign change was found in column 1 of the stability criteria. This sign change indicates instability in the system. At a *K* value of 238–240, the term "\(\tau_2 b + a - KQ_{gen} \tau_1 M\)" changed sign. This would indicate that a *K* value between
238-240 would be marginally stable for a fuel rate of 18 lbs/hr. More discussion of this can be found in Chapter 6.

4.6 DEVELOPMENT OF COMPUTER MODELS

The differential equations developed in section 4.2 were the basis for the simulation model used in this study. First, a program was developed to evaluate the numerous constants of the model. This program, PARAM, is listed in Appendix E. PARAM was then integrated into another program, SIMMOD, that solved the differential equations. Keener and Meyer, (1982) discussed the method used to solve the differential equations. The on/off controller was simulated by turning the fuel on if the error signal was greater than 30° C and off when the signal fell to 0 or less. The program SIMMOD is also listed in Appendix E.

The proportional controller was simulated by adjusting the fuel rate according to the error signal. The equation used was based on the average amount of heat required during the on/off control segment. This was determined by multiplying the total amount of heat available, based on 8000 Btu/hr * fuel rate, by 80 percent, the average time the burner was on in a cycle. This value was then decreased in proportion to the error signal of the burner. The entire equation was divided by \( \frac{m_c p_s}{\text{sand}} \) to fit into the simulation program, resulting in a proportional controller transfer equation of \( (14000 - 600t_e) \ ^\circ \text{R/hr} \).

This program is called SIMMOD2 and is also listed in Appendix E. Two other adaptations of PARAM, DESCRIB and PROPCON, were used in the describing function analysis and proportional controller stability analysis. These additions to PARAM are also in Appendix E.
Chapter V

EXPERIMENTAL EVALUATION

Some assumptions were made in developing the computer model for simulating the prototype burner. The fuel was assumed to be completely combusted; heat flow was assumed one-dimensional; and the masses were assumed lumped at the centers of each shell.

The validity of these assumptions was evaluated comparing the computer model predictions with experimental results. The prototype unit was subjected to a series of step inputs and the thermal response of the system recorded. The experimental data was used to evaluate the computer model with an on/off controller.

The experimental testing was also used to determine such parameters as the time constant of the thermocouple well which was needed in the theoretical analysis.

5.1 EXISTING CONTROL OF THE BURNER

The existing control of the system is based on the bed temperature. When starting the burner cold, the sand bed is preheated by an electric heater. This process is automatic and is started by pushing a start button. The preheater comes on and the heated air flows through the bed. This continues for several hours until the bed temperature reaches approximately 550°C. At this point the burner is triggered manually or automatically and the preheater goes off; the fluidizing air flow is increased, the airlock comes on; the cob feeder comes on; and the fuel begins to burn as it enters the bed of hot sand. As the bed continues
to heat up, the cooling air comes on at its low flow rate at 600°C and then on high flow at 700°C. At 750°C/720°C the fuel will cycle off/on, respectively, upon call by sand bed temperature. This control scheme was illustrated previously in Figure 3. There is also a low limit to open the relays upon failure of the bed.

Before testing began, the burner was allowed to operate 4-5 hours to warm up the insulation and approach pseudo-steady state conditions. In this state, the fuel rate and air flow rates are such that the cobs cycle on and off to maintain the burner set point temperature. Throughout the testing, the set points were maintained at 734°C and 724°C.

5.2 TEST OBJECTIVES

1. To determine the thermal response (cycle frequency and magnitudes of overshoot and undershoot) of the system to a step-input excitation.

2. To compare the observed response with the computer prediction.

3. To determine the time constant associated with the thermocouple well.

5.3 TEST PROCEDURES

The first set of experiments was aimed at determining the conduction heat loss of the burner. The burner was operated for several hours until steady-state conditions were obtained, then the entire burner was shut down. There was no fuel burning, no fluidizing air running, and no cooling air running. A cover was placed over the exhaust of the heat
exchanger to prevent natural convection losses. The only heat loss present then was conduction through the walls and through the top and bottom of the burner. The thermal response of both the bed and the heat exchange area was monitored during this time. Data was collected continuously for the first 5-10 minutes, then every one minute thereafter. A time constant for conduction was obtained from the cooling curve. For statistical purposes, the tests were repeated 3 times under the same conditions.

The second group of tests was concerned with the entire system operation. The temperature decay curve from the combined effect of conduction losses, cooling air induced convection losses, and fluidization air induced convection losses. Again, the burner was heated to steady state conditions, the cooling air was left on, the fluidizing air was left on, and the system was allowed to operate in its normal cycle with the fuel cycling on and off, simulating step inputs to the system, to maintain the set point. Temperatures, in both the bed and the heat exchanger, were recorded as often as the data logger could print them through each cycle. These tests were run at three different air-to-fuel ratios - 1.13, 1.18, and 1.23, and then at three different cooling air rates - 287, 197, and 87 pounds per hour. For statistical purposes, the burner was allowed to cycle several times during each test with each cycle representing a test itself.

Recently, the thermocouples located in the bed have been inserted into thermowells in order to protect them from attrition. Placing the thermocouples in thermowells adds additional lags to the measurement
device. This lag, or resistance, arises from the material of the well, the air space involved, the depth of immersion, etc. This lag plays a role in the control of a system because, in general, the more lag in a feedback loop, the more unstable a system gets.

To get an idea of what magnitude range the time constant is in, the following third experimental plan was conducted.

A bare thermocouple in equilibrium with the air, was plunged into a temperature controlled hot/cold water bath and its temperature rise-decay curve was recorded. The reverse situation (water-to-air) was also conducted getting a temperature curve for the thermocouple in air. The time constant for the thermocouple in each situation was calculated from the curves.

The same experiment was repeated for a thermocouple enclosed in a sheath, obtaining time constants for both air environment and water environment. The results were used to predict the time constant in a fluidized bed. It should be pointed out, however, that the conditions in a fluidized bed, in particular, the heat transfer coefficient, differ significantly from those of air or water alone. The water environment time constants are probably the closest to that of the FBC's but one would still expect the time constant of the sheathed thermocouple in the bed to be smaller because of the high heat transfer coefficient in the bed.

Due to time restraints in this project, only one test in water of the thermocouple in the sheath was run and an evaluation of the time constant made. However; the simulation program was used to test the
effect of time constant on stability. The results are in Chapter 6 and will be discussed there.

In addition to the previous experiments, two sets of tests were run, but not analyzed for parameter values in this study.

The fourth group of tests examined the convection heat losses created by the fluidizing air. The burner was heated up to steady state, the cooling air was shut off, and a cover was placed over the exhaust of the heat exchanger to prevent natural convection losses. The fluidizing air was left on and the burner allowed to cycle (the cobs were off then suddenly were on, or they were on then suddenly were off). Under these conditions the burner heat loss is due to conduction through the burner walls, top, and bottom and by convection due to the fluidizing air. The temperatures were again recorded continuously in order to get a temperature decay curve. The point of cob turn on and off was also recorded on the data strip in order to get a cycling frequency. For statistical purposes, several cycles or step inputs of the cobs were recorded during each test. The tests were conducted at 3 different air-to-fuel ratios. They were 1.44, 1.32, and 1.03.

The fifth and final group of tests examined the contribution of cooling-air-induced convection losses. In these tests, the burner was heated up to steady state, then everything was shut off except the cooling air. The burner heat loss under these conditions is due to conduction through the burner walls and cooling-air-induced convection losses. Three different cooling air flow rates were tested: 287, 197, and 87 pounds per hour. The temperatures in both the bed and heat exchanger
were recorded continuously at the beginning of the test and only one minute intervals thereafter. The burner was allowed to cool down to approximately 650°C during each test. For statistical purposes, 3 tests were run at each cooling air flow rate.

5.4 INSTRUMENTATION AND DATA COLLECTION

The fuel used for all experimental testing was corn cobs (the burner has previously been used extensively as a corn cob combustor). Even though the burner was operated below the ash fusion point two percent lime by weight (Henry, et al., 1984), was added to the cobs as a way of insuring that the ash would not melt and cause plugging. Previous ultimate and proximate analysis of corn cobs were done (Sukup, 1982; Keener, 1981; and others). The results that Keener et al. (1981), used in their analyses were used here also and are presented in Appendix C.

During these tests the fuel rate was determined in two different ways. One method was to determine the weight of fuel used during an entire test, and record when it was off and when it was on during the test to determine its feed rate when it was on. The second method was to remove the spout from the feeder when the burner did not require fuel, let the feeder run, time it and catch the dispensed fuel and weigh it. The average of 5 runs was used. The second method was more accurate and was used in the energy balances.

The moisture content of the cobs for each test was determined by placing 3 samples of approximately 200g each in a hot oven at 103°C for 72 hours and calculating moisture loss during that period.
Ambient air properties were obtained using a EG&G 880 dewpoint hygrometer, a barometer, and dry bulb thermometer. Pressure drops for the cooling air and fluidizing air were measured with 4 manometers installed on the system. The manometer used to measure the pressure drop across the bed served as a good indicator of the extent of fluidization. When the manometer fluctuated about the mean bed pressure the bed was well fluidized while a stagnant pressure reading indicated poor fluidization.

The gas leaving the fluidized bed was sampled for CO₂, CO, and O₂ content using MSA analyzers. The O₂ content is a fairly direct measure of excess air. The CO content of the flue gas is a direct measure of the completeness of combustion and, therefore, is an objective measure of combustion efficiency that doesn't depend on the burner type or fuel being burned.

The fluidized bed burner had 5 chromel/alumel thermocouples inserted at various heights throughout the bed as can be seen in Figure 14. There were also thermocouples in the heat exchanger at the same position as those in the bed. The bed thermocouples were inserted in thermowells to prevent attrition. Those in the heat exchanger were not in thermowells. Two different sizes of wells were used which had different time constants.

The data was recorded by a Kaye System 8000, 30-channel data logger and on an MPE 2500 magnetic tape recorder. The data was then transferred from the magnetic tape to a computer data file for further analyses.
The temperatures and pressures, when recorded "continuously", were being recorded as fast as the data logger could print through the channels each time. For energy balance calculations, all measurements of temperature, differential pressures, and gas sampling were time-averaged over the test period to average the cyclic fluctuations in the system caused by the intermittent feeding of cubes into the chamber.

Each instrument used was calibrated against standards referenced to the National Bureau of Standards.
A summary of the data collected includes fuel rate, temperatures in the bed and heat exchanger, including the combustion air in and out, and the cooling air in and out, pressure drops, barometric pressure, and dewpoint temperature, and combustion air and cooling air flow rates. Ash samples were also collected from each test but no elemental analysis was performed for chemical content or unburned carbon (Schonauer et al., 1984).

5.5 **DATA ANALYSIS**

The data was transferred from magnetic tape to the computer for time-averaging to minimize fluctuations due to the cobs cycling. A time-average program developed by Keener (1984 Personal Communication) for previous work was modified and utilized to account for different amounts of data and used in the analysis.

A computer program for the system energy balance, also previously developed by Keener, was modified (to account for the addition of lime to the fuel) and used. The lime was treated as ash and the weight percentages of the corncobs constituents were recalculated (Appendix F). The program took into account energy inputs from both the fluidizing pump and cooling fan and was used to determine burner efficiency in each test. Inputs to the program were taken from the computer printouts of the measured variables.

The cycle times were computed based on the cob feeder on-off operation recorded by the Kaye data logger. Some estimation was required here because the cob feeder sometimes came on while the logger was printing other data.
Chapter VI

RESULTS AND DISCUSSION

6.1 SIMULATION - ON/OFF

The computer model was developed so that the effect of various parameters could be investigated. The parameters under investigation were emissivities, cooling air flow rates, fuel rates, time constant of the thermowell, and the deadtime. Table 2 shows the values used in a typical run of the real burner.

<table>
<thead>
<tr>
<th>run</th>
<th>emissivity</th>
<th>ht trnsi coeff</th>
<th>cool air</th>
<th>time</th>
<th>deadt</th>
<th>fuel rate</th>
</tr>
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<tbody>
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<td>9</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
<tr>
<td>2</td>
<td>.01</td>
<td>9</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
<tr>
<td>3</td>
<td>.99</td>
<td>9</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
<tr>
<td>4</td>
<td>.65</td>
<td>9</td>
<td>300</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
<tr>
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<td>.65</td>
<td>9</td>
<td>100</td>
<td>.017</td>
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<td>.017</td>
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<tr>
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</tr>
<tr>
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<td>.65</td>
<td>9</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
<tr>
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<td>.65</td>
<td>9</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
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</tr>
<tr>
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<td>.65</td>
<td>1000</td>
<td>200</td>
<td>.017</td>
<td>.0008333</td>
<td>18</td>
</tr>
</tbody>
</table>
The raw data from these runs is shown in Appendix G.

6.1.1 Emissivity

The effect of the emissivity value for the radiation from the heat exchanger walls is illustrated in Figures 15, 16, 17 and Table 3. The lower emissivity value of .01 resulted in lower steady state temperatures in the insulation and the outside wall, as well as a lower average temperature in the heat exchanger and higher average temperatures in both the burner and wall 1, because less heat was emitted from wall 1 to the exchanger. With a low emissivity, the exchanger and wall 2 temperatures were nearly the same and much lower than that of wall 1. As the emissivity increased, the heat exchanger temperature increased and that of wall 2 approached that of wall 1. It was rather unusual that the temperature increased, decreased, then increased radially outward from the center of the burner; however, this behavior was also observed by Keener in previous burner tests. There was very little difference in either temperatures or cycle time for an emissivity value between .65 and .99 although the same trend in temperature was found in comparing results using emissivities of .65 and .99 as .01 and .65. The values given in the Perry and Chilton, 1973, vary depending on the surface of the metal wall. A value of .65 is estimated for the burner.

Table 3. Effect of Emissivity.

<table>
<thead>
<tr>
<th>run</th>
<th>cycle (hr)</th>
<th>peak (R)</th>
<th>low (R)</th>
<th>TB (R)</th>
<th>TW₁ (R)</th>
<th>TEX (R)</th>
<th>TW₂ (R)</th>
<th>TINS (R)</th>
<th>TW₃ (R)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>e=.65</td>
<td>.076</td>
<td>1892</td>
<td>1704</td>
<td>1892</td>
<td>1880</td>
<td>1503</td>
<td>1709</td>
<td>958</td>
</tr>
<tr>
<td>2</td>
<td>e=.01</td>
<td>.072</td>
<td>1910</td>
<td>1703</td>
<td>1910</td>
<td>1899</td>
<td>1348</td>
<td>1294</td>
<td>828</td>
</tr>
<tr>
<td>3</td>
<td>e=.99</td>
<td>.076</td>
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<td>1714</td>
<td>1893</td>
<td>1881</td>
<td>1532</td>
<td>1782</td>
<td>974</td>
</tr>
</tbody>
</table>
FIGURE 15. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - STANDARD TEST. FUEL RATE = 18 LBS/HR, AIRFLOW ≈ 200 LBS/HR, EMISSIVITY = 0.65, TIME CONSTANT (THERMOWELL) = 0.017 HR, DEAD TIME (FUEL) = 0.000833 HR, HEAT TRANSFER COEFFICIENT (EXCHANGER) = 9.0 BTU/HR °F FT².
PLOT OF TEMPERATURES INSIDE THE FGC
SIMULATION RUN 3

FIGURE 17. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF EMISSIVITY. EMISSIVITY = .99
6.1.2 Cooling Air Rate

Three different cooling air flow rates were tested by the computer model, 100, 200, and 300 lb/hr. This was done to determine whether it would be beneficial to use the cooling air as a controlled variable in the control system. The results are shown in Figures 18, 15, 19, and Table 4.

The cooling air flow rate did not have any effect on the steady state temperature in wall 3 but the temperatures of wall 2 and the insulation were slightly higher at lower flow rates. This is expected because more heat was being removed in the heat exchanger as the flow rate was increased, thus less was available to move radially outward. With the lower flow rate, a much shorter cycle time was obtained with more overshoot but less undershoot. As the cooling air traveled down the length of the bed it got hotter. At low flow rates, the cooling air could not remove heat from the bed fast enough to keep the temperature from overshooting after the fuel was shut off, which caused the larger overshoot. Because there was less air to heat at the low cooling air rate, the temperature rise was faster after the fuel shut off and the temperature fell slower after the fuel turned on because there was less air cooling occurring. This can be seen by the increase in time that the burner was on per cycle, .071 hrs at 300 lb/hr compared to .039 hrs at 100 lb/hr, as it took longer for the burner to catch up to itself again with so much heat being removed at the high flow rate. In going from 100 to 300 lb/hr, the undershoot increased 21°R, (29%), the
FIGURE 19. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF AIRFLOW. AIRFLOW = 300 LBS/HR
Table 4. Effect of Cooling Air Flow Rate.

<table>
<thead>
<tr>
<th>run (lb/hr)(hr)</th>
<th>cycle</th>
<th>peak</th>
<th>low</th>
<th>on/off</th>
<th>off/on</th>
<th>TE (R)</th>
<th>TW₁ (R)</th>
<th>TSX (R)</th>
<th>TW₂ (R)</th>
<th>TIN₅ (R)</th>
<th>TW₃ (R)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5 100</td>
<td>0.069</td>
<td>1913</td>
<td>1715</td>
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<td>0.03</td>
<td>1913</td>
<td>1903</td>
<td>1660</td>
<td>1762</td>
<td>979</td>
<td>576</td>
</tr>
<tr>
<td>1 200</td>
<td>0.076</td>
<td>1892</td>
<td>1704</td>
<td>0.053</td>
<td>0.023</td>
<td>1892</td>
<td>1880</td>
<td>1503</td>
<td>1709</td>
<td>958</td>
<td>575</td>
</tr>
<tr>
<td>4 300</td>
<td>0.091</td>
<td>1876</td>
<td>1694</td>
<td>0.071</td>
<td>0.02</td>
<td>1876</td>
<td>1863</td>
<td>1385</td>
<td>1671</td>
<td>942</td>
<td>575</td>
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</table>

Overshoot decreased 37°R, (51%), and the cycle time increased .022 hr, (32%).

6.1.3 Fuel Rate

It was desirable to know the effect of fuel rate because the controlled variable is presently controlled by turning the fuel on/off and it is possible that the fuel rate could be put on a proportioned rate. Three fuel rates, 15, 18, and 21 lb/hr were run. The results can be seen in Figures 20, 15, 21 and Table 5.

As shown in the figures, the fuel rate had little effect on the steady state temperatures of the outside wall, and insulation. As the fuel rate was increased, the cycle time became shorter with more overshoot and less undershoot because more heat was being generated than the cooling air could remove quickly enough. From the results of the cooling air effects and these results with fuel rate, it appears that a combination of cooling air and fuel rate would be a good possibility for better control of the system. One has to be careful, though, because the feeder has a limit on fuel throughput. The air-to-fuel ratio also has to be monitored to avoid less than stoichiometric or very excessive air.
FIGURE 20. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF FUEL RATE. FUEL RATE = 15 LBS/HR
FIGURE 21. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF FUEL RATE. FUEL RATE = 21 LBS/HR
### Table 5. Effect of Fuel Rate.

<table>
<thead>
<tr>
<th>run</th>
<th>fuel (lb/hr)</th>
<th>cycle (hr)</th>
<th>peak (R)</th>
<th>low (R)</th>
<th>on (hr)</th>
<th>off (hr)</th>
<th>TB (R)</th>
<th>TW₁ (R)</th>
<th>TEX (R)</th>
<th>TW₂ (R)</th>
<th>TINS (R)</th>
<th>TW₃ (R)</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
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<td>0.092</td>
<td>1876</td>
<td>1715</td>
<td>0.07</td>
<td>0.022</td>
<td>1876</td>
<td>1865</td>
<td>1498</td>
<td>1709</td>
<td>959</td>
<td>575</td>
</tr>
<tr>
<td>1</td>
<td>18</td>
<td>0.076</td>
<td>1892</td>
<td>1704</td>
<td>0.053</td>
<td>0.023</td>
<td>1892</td>
<td>1880</td>
<td>1503</td>
<td>1709</td>
<td>958</td>
<td>575</td>
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<tr>
<td>7</td>
<td>21</td>
<td>0.066</td>
<td>1911</td>
<td>1701</td>
<td>0.042</td>
<td>0.024</td>
<td>1911</td>
<td>1898</td>
<td>1511</td>
<td>1714</td>
<td>960</td>
<td>575</td>
</tr>
</tbody>
</table>

#### 6.1.4 Thermowell Time Constant

The thermocouples were recently inserted into thermowells in the bed to protect them. The value of the time constant depends on several things such as size of thermowell, material of well, distance thermocouple is inserted, flow conditions, etc.

Three different time constants were tested, .008, .017, and .08 hours. The length of the time constant appears to be very important in the control of the burner. It added a delay to the feedback sensing element, which, if large enough, could possibly cause instability. The results are given in Figures 22, 15, 23, and Table 6.

The cycle times changed dramatically by increasing the time constant, becoming larger by 282% when the time constant was increased tenfolds. The overshoot and undershoot also increased with increasing time constant, 455% and 268%, respectively. This is to be expected because by the time the error signal has reached the controller, something else has occurred in the burner.
FIGURE 22. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF TIME CONSTANT. THERMOWELL TIME CONSTANT = 0.008 HR
FIGURE 23. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF TIME CONSTANT. THERMOWELL TIME CONSTANT = 0.08 HR
Table 6. Effect of Thermowell Time Constant.

<table>
<thead>
<tr>
<th>run</th>
<th>tau2 (hr)</th>
<th>cycle (hr)</th>
<th>peak (R)</th>
<th>low (R)</th>
<th>TE (R)</th>
<th>TW1 (R)</th>
<th>TEX (R)</th>
<th>TW2 (R)</th>
<th>TINS (R)</th>
<th>TW3 (R)</th>
</tr>
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<tbody>
<tr>
<td>8</td>
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<td>1740</td>
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<td>1694</td>
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<td>1704</td>
<td>1892</td>
<td>1880</td>
<td>1503</td>
<td>1709</td>
<td>958</td>
<td>575</td>
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<td>9</td>
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<td>.187</td>
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<td>1990</td>
<td>1587</td>
<td>1819</td>
<td>979</td>
<td>576</td>
</tr>
</tbody>
</table>

6.1.5 Dead Time

The effect of the "dead" time the fuel takes to reach the bed once it leaves the feeder is expected to be similar to that of increasing the thermowell time constant because the longer it takes the cobs to get to the burner the more change there can be between error signal command and action. The value chosen for the standard test was 8.33x10^-4 hr and was based on doubling of the dead time to .0017 hr which increased the period of cycling to .079 hr (4% increase), the overshoot 2°R and undershoot 4°R. The results can be seen in Figures 24 and 15 and Table 7.

Table 7. Effect of Dead Time.

<table>
<thead>
<tr>
<th>run</th>
<th>dt (hr)</th>
<th>cycle (hr)</th>
<th>peak (R)</th>
<th>low (R)</th>
<th>TB (R)</th>
<th>TW1 (R)</th>
<th>TEX (R)</th>
<th>TW2 (R)</th>
<th>TINS (R)</th>
<th>TW3 (R)</th>
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</thead>
<tbody>
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<td>1704</td>
<td>1892</td>
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<td>1709</td>
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<td>10</td>
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<td>.079</td>
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<td>1700</td>
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<td>1882</td>
<td>1504</td>
<td>1711</td>
<td>957</td>
<td>575</td>
</tr>
</tbody>
</table>
FIGURE 24. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF DEAD TIME. DEAD TIME (FUEL INTO BURNER) = 0.0017 HR
6.1.6 Heat Transfer Coefficient at Exchanger Walls

The effect of having an extremely high heat transfer coefficient at the heat exchanger walls can be seen in Figures 25 and 15 and Table 8. If it is extremely high, the temperature profile in the bed, wall 1, heat exchanger, and wall 2 would be nearly identical. Since this is not the case in real practice, this suggests that radiation plays a much larger role in heat transfer in the heat exchanger than convection.

Table 8. Effect of Heat Transfer Coefficient at Exchanger Walls.

<table>
<thead>
<tr>
<th>run</th>
<th>coeff (hr)</th>
<th>cycle</th>
<th>peak (R)</th>
<th>low (R)</th>
<th>TB (R)</th>
<th>TW1 (R)</th>
<th>TEX (R)</th>
<th>TW2 (R)</th>
<th>TINS (R)</th>
<th>TW3 (R)</th>
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</thead>
<tbody>
<tr>
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<td>.076</td>
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<td>1704</td>
<td>1992</td>
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<td>1503</td>
<td>1709</td>
<td>958</td>
<td>575</td>
</tr>
<tr>
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<td>1000</td>
<td>.099</td>
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<td>1708</td>
<td>1877</td>
<td>1864</td>
<td>1845</td>
<td>1846</td>
<td>983</td>
<td>576</td>
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</table>

6.2 EXPERIMENTAL

The experimental results are given in this section. The parameters under investigation were conduction losses, fuel rate effect, cooling air flow rate effect, thermowell time lag, and convection losses induced by the fluidizing and cooling air flows.

6.2.1 Conduction

To determine the conduction losses of the burner system, a set of tests was conducted in which the burner was allowed to reach steady state, then totally shut off and allowed to cool. This 'state' of the burner is the same as that of the night cycle where the burner was allowed to cool to some preset point, maybe a 40 degree span between on and off cycles. If an overall heat transfer coefficient can be obtained
FIGURE 25. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION - EFFECT OF HEAT TRANSFER. HEAT TRANSFER COEFFICIENT (EXCHANGER) = 1000 BTU/HR °F FT²
from these curves, it can be determined what temperature span is required to obtain a specific cycle time. This transfer coefficient is found using the formula:

\[
\frac{(T_2 - T)}{(T_{20} - T)} = e^{-\frac{T}{T_0}}
\]

where \( T_2 \) = burner temperature

\( T \) = surrounding temperature

\( T_{20} \) = initial temperature

The results are given in Table 9.

Table 9. Burner Conduction Time Constant.

<table>
<thead>
<tr>
<th>Test</th>
<th>Time Constant (1/hr)</th>
<th>Average (1/hr)</th>
<th>Stand. Dev. (1/hr)</th>
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<tbody>
<tr>
<td>7</td>
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<td>1.52</td>
<td>.37</td>
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<td>12</td>
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</tr>
<tr>
<td>13</td>
<td>1.12</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

A representative temperature profile of the FBC during these tests is given in Figure 26.

6.2.2 Fuel Rate

One group of tests was run to test the effect of fuel rate on cycle time, overshoot, and undershoot. The entire system was tested under normal operating conditions. Several cycles were run at each fuel rate and these values, as well as averages and standard deviations can be found in Appendix H. The average values are given in Table 10.
FIGURE 26. EXPERIMENTAL COOLING CURVE TO FLUIDIZED BED BURNER FOLLOWING SHUTOFF.
Table 10. Effect of Fuel Rate - Experimental.

<table>
<thead>
<tr>
<th>Test</th>
<th>Fuel (lb/hr)</th>
<th>On/On (min)</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>Under (°C)</th>
<th>Over (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>14.2 (DOB)</td>
<td>6.3</td>
<td>5.8</td>
<td>.5</td>
<td>14.9</td>
<td>1.3</td>
</tr>
<tr>
<td>1</td>
<td>15.1</td>
<td>4.2</td>
<td>3.7</td>
<td>.5</td>
<td>15.4</td>
<td>1.0</td>
</tr>
<tr>
<td>3</td>
<td>16.8</td>
<td>2.4</td>
<td>1.9</td>
<td>.5</td>
<td>13.2</td>
<td>2.3</td>
</tr>
</tbody>
</table>

Notice that as the fuel rate increased, cycle time decreased, and there was less undershoot and more overshoot. Figures 27, 28, and 29 show the increase in cycling time as fuel rate is decreased. The same thing is illustrated in a complete temperature profile of the burner in Appendix G. This is the same trend predicted by the simulation model. A more thorough comparison between the simulation and experimental results will be given in section 6.3.

A regression analysis was conducted on the variables cycle time, undershoot, and overshoot with fuel rate as the predictor. It appears that fuel rate and cycle time are linearly related because an $R^2$ of 95.3% was obtained. The relationship found was:

$$\text{cycle time} = 26.3 - 1.43 \times \text{fuelrate}$$

This may be useful to know because, if desired, the fuel rate can be adjusted to give a particular cycle time. The correlation between air-to-fuel ratio and cycle time was low with an $R^2$ of 21.3%. More tests are required to evaluate the effect of fuel rate on undershoot and overshoot behavior, because fuel rate should be more linearly related to overshoot and undershoot than indicated.
FIGURE 27. EXPERIMENTAL RESPONSE CURVE FOR CONTROL TEMPERATURE OF FBC - EFFECT OF FUEL RATE. FUEL RATE = 14.2 LBS/HR
FIGURE 28. EXPERIMENTAL RESPONSE CURVE FOR CONTROL TEMPERATURE OF FBC - EFFECT OF FUEL RATE. FUEL RATE = 15.1 LBS/HR
FIGURE 29. EXPERIMENTAL RESPONSE CURVE FOR CONTROL TEMPERATURE OF FBC - STANDARD TEST. FUEL RATE = 16.9 LBS/HR., AIRFLOW = 287 LBS/HR
It is also interesting to note that as the air-to-fuel ratio decreased with increasing fuel rate, the efficiency of the burner dropped from 63.7% to 54% (Appendix H). This was expected because the amount of excess air decreased and more unburned fuel could have been leaving the system. Also, because the fuel rate increased, more heat was generated in the bed but the cooling air did not increase proportionally to remove the extra heat, thus heat energy was lost in the exhaust.

6.2.3 Cooling Air Rate

Experimental testing was also conducted to test the effect of cooling air flow rate on cycle time, undershoot, and overshoot. The average values for all the cycles are given in Table 11.

Table 11. Effect of Cooling Air - Experimental

<table>
<thead>
<tr>
<th>Run</th>
<th>Flow (lb/hr)</th>
<th>On/On (min)</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>Under (°C)</th>
<th>Over (°C)</th>
<th>Fuel Rate (lb/hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>287.1</td>
<td>2.43</td>
<td>1.94</td>
<td>.50</td>
<td>13.1</td>
<td>2.3</td>
<td>16.9</td>
</tr>
<tr>
<td>18</td>
<td>197</td>
<td>1.91</td>
<td>1.35</td>
<td>.56</td>
<td>10.1</td>
<td>3.4</td>
<td>16.8</td>
</tr>
<tr>
<td>19</td>
<td>86.8</td>
<td>1.66</td>
<td>.87</td>
<td>.79</td>
<td>7.5</td>
<td>6.3</td>
<td>16.9</td>
</tr>
</tbody>
</table>

As can be seen in Figures 29, 30, and 31, the cycle time increased with cooling air flow rate, that is, the frequency of cycling increased 32% as flow rate went from 300-100 lb/hr. Also, the amount of undershoot decreased and overshoot increased with decreasing cooling air flow rate. Regression analysis was performed to predict cycle time, undershoot, and overshoot dependence on cooling air flow rate. Good
FIGURE 30. EXPERIMENTAL RESPONSE CURVE FOR CONTROL TEMPERATURE OF FBC - EFFECT OF AIR FLOW. AIRFLOW = 137 LBS/HR.
FIGURE 31. EXPERIMENTAL RESPONSE CURVE FOR CONTROL TEMPERATURE OF FBC - EFFECT OF AIR FLOW. AIRFLOW = 86.8 LBS/HR.
correlation, ($r^2$ greater than 93%), was obtained in all cases. The relations found were

undershoot = 4.89 + .0283 * cooling air

overshoot = 7.84 - .0202 * cooling air

cycle time = 1.28 + .0038 * cooling air

Again, the behavior observed in the experimental results follow the same trend as that predicted by the simulation model and a more complete comparison will be given in section 6.3.

As expected, when maintaining the air-to-fuel ratio and fuel rates at 1.2 and 16.9 lb/hr respectively, while decreasing the cooling air flow rate, the efficiency of the burner dropped from 54% to 19% (Appendix H) because the low flow rate could not remove all the available heat.

6.2.4 Time Constant

One experimentally obtained value for the thermal time constant associated with the thermowells used in the bed, was obtained from H.M. Keener. The value was for a thermocouple, in a sheath, in a water bath and was evaluated to be 14-23 sec (.0039-.0064 hrs). A value of 1.4-4.5 sec (.00039-.00125 hrs) for a bare thermocouple in a water bath was obtained from Thermoelectric, 1975. These values illustrate that there is a thermal lag associated with the thermowell. The time constant of the thermowell in the fluidized bed would be expected to be somewhat smaller than those given for water due to the higher heat transfer coefficient in the bed.
The values used in the simulation model were .008, .017, and .08 hrs. These values are all larger than the ones estimated for the bed. Since the estimated values are so small in comparison to the simulated values, the time lag would not be expected to have too large an effect on the stability of the burner system providing the model sufficiently describes the burner.

6.2.5 Additional Testing

In the fourth group of tests, the effect of fluidizing induced convection losses were to be investigated. The burner was allowed to cycle but no cooling air was used to remove heat from the bed. Figures 32, 33, 34 and Table 12 show the results of the tests. Notice that the temperatures began to increase in the upper part of the bed since no heat was removed other than by conduction and fluidizing air losses. A fuel rate effect is apparent here, also, cycle time decreased, undershoot increased, and overshoot increased with increased fuel rate. The undershoot effect was opposite that obtained when looking at the total system - in that case decreasing with increased fuel rate. This could be due to an offsetting effect of cooling air in the overall system.

Table 12. Fluidizing Air Heat Losses.

<table>
<thead>
<tr>
<th>A/F</th>
<th>Run</th>
<th>Fuel (lb/hr)</th>
<th>On/On (min)</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>Under (°C)</th>
<th>Over (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.44</td>
<td>6</td>
<td>13.9</td>
<td>1.83</td>
<td>1.10</td>
<td>.72</td>
<td>7.1</td>
<td>3.4</td>
</tr>
<tr>
<td>1.32</td>
<td>8</td>
<td>15.5</td>
<td>1.65</td>
<td>1.01</td>
<td>.63</td>
<td>7.14</td>
<td>4.3</td>
</tr>
<tr>
<td>1.03</td>
<td>14</td>
<td>19.6</td>
<td>1.59</td>
<td>.86</td>
<td>.72</td>
<td>8.7</td>
<td>6.4</td>
</tr>
</tbody>
</table>
FIGURE 32. EXPERIMENTAL RESPONSE CURVES FOR BED TEMPERATURE OF FBC WHEN NO COOLING AIR. FUEL RATE = 13.9.
FIGURE 33. EXPERIMENTAL RESPONSE CURVE FOR BED TEMPERATURE OF FBC WHEN NO COOLING AIR. FUEL RATE = 15.5
FIGURE 34. EXPERIMENTAL RESPONSE CURVES FOR BED TEMPERATURE OF FBC WHEN NO COOLING AIR. FUEL RATE = 19.6
FIGURE 35. EXPERIMENTAL RESPONSE CURVES FOR BED TEMPERATURES - NO BURNING OF FUEL. COOLING AIRFLOW = 287 LBS/HR.
FIGURE 36. EXPERIMENTAL RESPONSE CURVES FOR HEAT EXCHANGER - NO BURNING OF FUEL. COOLING AIRFLOW = 287 LBS/HR.
FIGURE 37. EXPERIMENTAL RESPONSE CURVE FOR BED TEMPERATURE - NO BURNING OF FUEL. COOLING AIRFLOW = 197 LBS/HR.
FIGURE 38. EXPERIMENTAL RESPONSE CURVE FOR HEAT EXCHANGER – NO BURNING OF FUEL. COOLING AIRFLOW = 197 LBS/HR.
FIGURE 39. EXPERIMENTAL RESPONSE CURVE FOR BED TEMPERATURE - NO BURNING OF FUEL. COOLING AIRFLOW = 87 LBS/HR.
FIGURE 40.  EXPERIMENTAL RESPONSE CURVE FOR HEAT EXCHANGER - NO BURNING OF FUEL.  COOLING AIRFLOW = 87 LBS/HR.
However; these tests were run under abnormally extreme air-to-fuel ratios and more investigation is required before any definite conclusions can be drawn.

The fifth group of tests examined the cooling-air-induced convective losses in the burner. A series of nine tests was run, 3 each at 3 cooling air flow rates, 287 lb/hr, 197 lb/hr, and 87 lb/hr. The cooling air alone was running during these tests, allowing conduction and cooling-air-induced heat loss to occur. The temperature profiles in both the bed and the heat exchanger for each test are given in Figures 35-40 and in Appendix H (statistical repetitions).

No significant analysis was conducted with the data other than comparing the time for the burner temperature to drop from 735°C to 700°C in these tests with the time for the same to occur in the conduction loss only tests. These results are given in Table 13.

**Table 13. Cooling Air Heat Losses.**

<table>
<thead>
<tr>
<th>Flow</th>
<th>Avg. Time for Drop</th>
<th>Std. Dev.</th>
<th>Cond.</th>
<th>% Change</th>
</tr>
</thead>
<tbody>
<tr>
<td>lb/hr</td>
<td>min</td>
<td>min</td>
<td>min</td>
<td></td>
</tr>
<tr>
<td>-------</td>
<td>--------------------</td>
<td>-----------</td>
<td>-------</td>
<td>----------</td>
</tr>
<tr>
<td>287</td>
<td>2.4</td>
<td>.4</td>
<td>5.2</td>
<td>54</td>
</tr>
<tr>
<td>197</td>
<td>3.4</td>
<td>.7</td>
<td></td>
<td>35</td>
</tr>
<tr>
<td>87</td>
<td>11.4*</td>
<td>.6</td>
<td></td>
<td>---</td>
</tr>
</tbody>
</table>

*This result indicates it takes longer to cool the bed with an air flow present than with conduction loss alone. One should note the different temperature profile in the bed at the test start compared to the other tests in regard to Locations 1 & 2. This could explain the increased time as Location 2 gains heat from the hotter bed bottom (Location 1).
6.3 COMPARISON

Two types of comparisons were made; one between the experimental and the simulation results and one between the theoretical predictions of the describing function and the actual experimental and simulation results.

A comparison between the experimental results and simulation results was done to determine how adequate the model is at describing the true burner behavior.

Fuel rate was varied under both experimental and simulation conditions. The results are tabulated in Table 14.

Table 14. Comparing Fuel Rate Effect.

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Cycle</th>
<th>Over</th>
<th>Under</th>
<th>Cool</th>
<th>Cycle</th>
<th>Over</th>
<th>Under</th>
<th>Cool</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>lb/hr</td>
<td>min</td>
<td>°C</td>
<td>°C</td>
<td>lb/hr</td>
<td>min</td>
<td>°C</td>
<td>°C</td>
</tr>
<tr>
<td>14</td>
<td>6.3</td>
<td>1.3</td>
<td>14.9</td>
<td>287</td>
<td>15</td>
<td>5.52</td>
<td>18.9</td>
<td>40</td>
</tr>
<tr>
<td>15</td>
<td>4.2</td>
<td>.99</td>
<td>13.4</td>
<td>285</td>
<td>18</td>
<td>4.56</td>
<td>27.8</td>
<td>46</td>
</tr>
<tr>
<td>16</td>
<td>2.4</td>
<td>2.3</td>
<td>13.2</td>
<td>287</td>
<td>21</td>
<td>3.96</td>
<td>38.3</td>
<td>48</td>
</tr>
</tbody>
</table>

The simulation model indicates a linear relation exists between fuel rate and cycle time, overshoot, and undershoot. The model predicts the same trends as those actually observed but the magnitudes are quite different in terms of hi-lo swings and cycle times. Only the cycle time was linear with fuel rate in the experimental results but, as pointed out earlier, more testing should be done to prove or disprove the overshoot and undershoot behavior obtained for the burner. The linear relations and their $R^2$ values are listed below.
There are several differences in the testing conditions that could be contributing to the discrepancies in results. The fuel rates are higher in the simulation model than in the experimental runs and as is the trend in both cases, the overshoot increased with increase in flow rate. Therefore, it is expected that the overshoots and undershoots of the model will be larger than the experimental values obtained just because the fuel rate is higher. The linear relations should still have similar slopes if the effect is the same. This is not the case. The experimental data has a slope of 1.43 which is 4.5 times larger than that of the model data which has a slope of .26. Secondly, the setpoint temperatures for on/off determination in the model were 30° apart (750-720°C) vs. the experimental case of (734-724°C) only a 10° swing. For this reason, the cycle times would be expected to be longer in the simulation model because of the 20° extra temperature correction required per each cycle.

The trend in cycle time occurs because as fuel rate increased, more heat was produced in the bed and it takes a shorter time to make up the temperature deficiency, thus, shortening the cycle time. Over and undershoots are larger because given the deadtime in the burner, a larger amount of fuel is carried into the bed after the feeder cycles causing more heat production. Also at the higher rate of temperature increases
the actual bed temperature will be higher when the fuel rate shuts off because of the thermocouple lag.

Finally, the cooling air rate was 287 lb/hr in the experiments and kept at 200 lb/hr in the model. There is a definite effect of cooling air flow rate on burner thermal response as discussed below and shown in Table 15. Regression analysis on both experimental and simulation data, predict a highly correlated linear relationship between cycle time, overshoot, and undershoot with cooling air flow rate.

Table 15. Comparing Cooling Air Effect.

<table>
<thead>
<tr>
<th>Experimental</th>
<th>Simulation</th>
</tr>
</thead>
<tbody>
<tr>
<td>cool cycle over</td>
<td>under fuel</td>
</tr>
<tr>
<td>lb/hr</td>
<td>min °C</td>
</tr>
<tr>
<td>-------</td>
<td>-------</td>
</tr>
<tr>
<td>287</td>
<td>2.43</td>
</tr>
<tr>
<td>197</td>
<td>1.91</td>
</tr>
<tr>
<td>87</td>
<td>1.65</td>
</tr>
</tbody>
</table>

Again, the same trend is predicted by the model as that observed experimentally. A decrease in cooling air flow rate causes a decrease in cycle time and undershoot and an increase in overshoot. This occurs because at low flow rates, not as much heat is removed from the bed after the on or off signal occurs, thus, not cooling it off as much after turning on before the temperature turns around but also not removing the heat that is produced after shutting off, thus raising the temperature. Although cooling air flow rates are similar in these tests, the fuel
rate is larger in the model which could explain why the magnitudes of over and under shoots are so large compared to the experimental results.

The experimental time constant study resulted in a quite small value of less than .0039 hours as a prediction for the lag associated with the thermocouple in a thermowell. This value is quite a bit smaller than the lowest used in the simulation, .008 hrs. At .008 hrs, the amplitude of oscillation was greatly reduced compared to that using larger values in the simulation. The value of time constant used in most of the simulation tests was .017 hrs which had a lot greater oscillation associated with it than what the burner apparently has. This value should be changed to something smaller in the model so it is more accurate. Note that the model did predict the limit cycle as was obtained in the experimental work.

Table 16 presents the results of the describing function predictions and the experimental and simulation results. For the case of d=10 (experimental conditions) the predicted cycle frequency, 2.45 min/cycle, compares quite closely to that actually obtained, 2.4 cycles/min. The peak-peak amplitude predicted for the measured burner temperature was 23.8°C vs. 25.5°C actually obtained. The measured temperature, Tm, is compared rather than TB because of the uncertainty in the value of the thermowell time constant. For d=30 (simulation conditions) the predictions were again quite close to those obtained in the simulation tests. Here TB values were compared because the time constant used in both prediction and simulation methods was the same value. The predicted
limit cycles were symmetric about the time axis, whereas the real situation had biased limit cycles either in overshoot or undershoot.

Table 16. Theoretical Predictions vs. Actual Results.

<table>
<thead>
<tr>
<th></th>
<th>Theoretical</th>
<th>Experimental</th>
<th>Theoretical</th>
<th>Simulation</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$\Delta_T$</td>
<td>$\Delta_T$</td>
<td>$\Delta_T$</td>
<td>$\Delta_T$</td>
</tr>
<tr>
<td>Cycle freq. (min/cycle)</td>
<td>2.45</td>
<td>2.4</td>
<td>3.85</td>
<td>4.56</td>
</tr>
<tr>
<td>Peak-peak amplitude ($^\circ$C)</td>
<td>23.8</td>
<td>25.5</td>
<td>102</td>
<td>104</td>
</tr>
</tbody>
</table>

6.4 SIMULATION - PROPORTIONAL

As presented earlier, (sections 4.4 & 4.5), a proportional controller was substituted for the on/off controller in the simulation program, where the fuel rate was adjusted in proportion to the error signal according to the equation $14000 - 600*te (^\circ$R/hr).

The results of the simulation are shown in Figure 41. The bed temperature reached its steady state value of $1803^\circ$R, ($729^\circ$C), in .058 hrs. The system's stability has been improved since the oscillations damp out, with the peak overshoot of $51^\circ$R, (2.8%), occurring at .01 hrs and undershoot of $27^\circ$R, (1.5%), at .032 hrs. The gain used was determined by deciding how much fluctuation in te would be allowed.

From a theoretical standpoint, a Routh Stability test predicted that a gain or $K$ value of 238-240 would cause marginal stability. This discrepancy in gain (600 vs 238) may be due to the assumption of no storage in the insulation, outer wall, and wall 2 when developing the characteristic equation but still including them in the model. Another
FIGURE 41. BURNER TEMPERATURE PROFILES, RADIAL DIRECTION – PROPORTIONAL CONTROLLER.
explanation is that the controller is supposed to be adjusting the fuel rate in proportion to the error signal but no adjustment was being made on the parameters that are dependent on fuel rate in the model. These parameters are shown in Appendix C to be negligible compared to the heat of combustion but were included in the model. The term that is limiting in the stability analysis is $\tau_2 b + a - KQ_{gen}TLM$. These parameters such as $\tau_2$ could be incorrectly estimated which would result in a gain that would not be compatible with the real system gain.

Further simulation work is indicated here if one would want to implement proportional control on the burner. For one to upgrade the control mode from on/off to proportional, the benefit gained in stability and control should be well worth the added investment and complexity of the controller required. In this particular case, the on/off controller appears to do an adequate job of control and to upgrade it would not be worthwhile.
Chapter VII

CONCLUSIONS, SUMMARY, AND RECOMMENDATIONS

The conclusions from this study are:

1. Mathematical and computer simulation models have been developed for the FBC system to predict dynamic response. They can predict trends but need further adjustments to predict values more accurately.

2. The fluidized-bed system is stable with on/off feedback control.

3. No need for proportional feedback control is indicated. The added expense and complexity is not necessary to get better control although the bed temperature would be controlled much closer without the oscillations present with an on/off controller.

4. Cooling air flowrate is indicated as a good possibility for the manipulated variable in a control scheme, either alone or in combination with fuel rate manipulation.

In summary, the model is found to at least predict the correct trends in thermal response as those actually observed but several things could be changed to improve the model. They are:

1. From the present model it was found that the mass of air in the heat exchanger is so small that storage in it could be neglected when solving the original matrices. If neglected, storage in wall 2 could be included in the model and this may be closer to what is actually occurring.
2. Applying the technique of Payne and Dunlap, 1982, Payne, Ross, Walker, 1980, or Sukup, 1982, for modeling the energy in and out of the burner rather than what was done, may be more accurate. A constant heat value was used in this analysis and several things were neglected that could affect the accuracy of the model. The approach by Payne et al. used the chemical formula of the fuel from the proximate analysis, then, without ignoring moisture in the fuel or air, the combustion equations were written as functions of moisture contents, thereby determining stoichiometric and excess air requirements as well as gross heat of combustion.

3. The thermowell time constant should be adjusted down to at least .006 hours.

The following are recommendations for further study:

1. The present simulation model predicts almost exactly off setting effects on undershoot, overshoot, and cycle time for the cooling air flow rate increasing from 200 to 300 lb/hr as for when the fuel rate increasing from 15 to 21 lb/hr. This, as well as the other results on fuel rate and cooling air rate tests, indicate that manipulation of both variables in a control scheme could give very tight control of the bed temperature and this needs further investigation.

2. Using cooling air as the basis for a control scheme should be investigated further using the results obtained thus far, particularly if the temperature of the heat exchanger exhaust needs to be kept constant.
3. To better compare the prediction ability of the present model to experimental observations, more simulations need to be run under exactly the same conditions as those in the experimental tests. Too many parameters were being varied at one time in this study to allow easy comparison.

4. More experimental evaluation of the thermowell time constant should be done because the model indicates that this additional lag increases instability if large enough. It is possible that in the bed, the lag is small enough that it can be neglected.

5. If it is desired to determine fluid air induced losses, a test similar to that used in cooling air induced loss determination should be ran. Everything should be shut off except fluidizing air and a cooling curve obtained. Then a percent increase over that of conduction alone, and a comparison between cooling air and fluidizing air can be made.


Appendix A

DERIVATION OF MATHEMATICAL MODEL EQUATIONS
As derived in Chapter 4, Section 4.2, the differential equation for the bed is

\[
\frac{dTB}{d\Theta} = \alpha_1(TB_1 - 537) - \alpha_2(TB - 537) + Q_{gen} - \alpha_3TB + \alpha_4TW_1
\]

where the definitions of all coefficients are summarized in Appendix B.

Continuing on, an equation for wall 1 can be written as follows:

\[
\frac{dTW_1}{d\Theta} = Q_{in} - Q_{out}
\]

where \(Q_{in} = mC \rho_{sand}(\alpha_3TB - \alpha_4TW_1)\)

\(Q_{out} = Q_{conv} + Q_{rad}\)

\[
= h_2(2\pi r_2)L(TW_{1b} - TEX) + \frac{G(TW_{1b}^4 - TW_{2a}^4)}{1/(2\pi r_2Le_1) + (1/2\pi r_3L)(1/e_2 - 1)}
\]

Assuming a lumped mass at the center, gives:

\[
\chi_3(TW_1 - TW_{1b}) = \chi_4(TW_{1b} - TEX) + \chi_5(TW_{1b}^4 - TW_{2a}^4)
\]

(1)

where

\[
\chi_3 = \frac{2\pi k_1L}{\ln(r_2/(r_2 + 5t_1))} \quad \text{(Btu/h F)}
\]

\[
\chi_4 = h_2(2\pi r_2)L \quad \text{(Btu/h F)}
\]

\[
\chi_5 = \frac{G}{1/(2\pi r_2Le_1) + (1/2\pi r_3L)(1/e_2 - 1)} \quad \text{(Btu/h F)}
\]

for the other side:

\[
\chi_6(TW_{2a} - TW_2) = \chi_7(TEX - TW_{2a}) + \chi_5(TW_{1b}^4 - TW_{2a}^4)
\]

(2)

where
\[ \xi_6 = \frac{(2\pi k_2)L}{\ln(r_3^*.5t_2/r_3)} \quad \text{(Btu/h F)} \]

\[ \xi_7 = h_3(2\pi r_3)L \quad \text{(Btu/h F)} \]

To linearize the fourth order term do Taylor expansion about the same point, \( T_o \), for both terms: (all temperatures are Rankine)

\[ TW_{1b}^4 = T_o^4 + 4T_o^3 (TW_{1b} - T_o) \]
\[ TW_{2a}^4 = T_o^4 + 4T_o^3 (TW_{2a} - T_o) \]

or

\[ TW_{1b}^4 - TW_{2a}^4 = 4T_o^3 (TW_{1b} - TW_{2a}) \]

Rewriting eq. 1 and 2,

\[ \xi_3(TW_{1b} - TW_{1b}) = \xi_4(TW_{1b} - \text{TEX}) + \xi_5(4T_o^3)(TW_{1b} - TW_{2a}) \] \quad (3)
\[ \xi_6(TW_{2a} - TW_{2a}) = \xi_7(\text{TEX} - TW_{2a}) + \xi_8(4T_o^3)(TW_{1b} - TW_{2a}) \] \quad (4)

Solving eq 4 for \( TW_{2a} \) gives

\[ TW_{2a} = B_1 TW_{1b} - B_2 \text{TEX} - B_3 TW_1 \]

Where

\[ B_1 = \frac{\xi_4 + \xi_3 + 4\xi_5 T_o^3}{485T_o^5} \quad \text{where } T_o \text{ is absolute temp, R} \]

\[ B_2 = \frac{\xi_4}{485T_o^5} \]

\[ B_3 = \frac{\xi_3}{485T_o^5} \]

Similarly, solving eq. 3 for \( TW_{1b} \) and substituting in \( TW_{2a} \):

\[ TW_{1b} = B_4 TW_{2a} - B_5 TW_2 - B_6 \text{TEX} \]
$$TW_{1b} = B_4 TW_{2a} - B_5 TW_2 - B_6 TEX$$

where

$$B_4 = \frac{86+4\delta_5 T_o^3}{4\delta_5 T_o}$$

$$B_5 = \frac{86}{4\delta_5 T_o}$$

$$B_6 = \frac{7}{4\delta_5 T_o}$$

Substituting $TW_{1b}$ and $TW_{2a}$ back into eq 1 and 2 results in

$$TW_{1b} = B_7 TEX + B_8 TW_1 + B_9 TW_2$$

$$TW_{2a} = B_{10} TEX + B_{11} TW_1 + B_{12} TW_2$$

where

$$B_7 = \frac{B_4 B_2 + B_6}{(B_4 B_1 - 1)}$$

$$B_8 = \frac{B_4 B_3}{(B_4 B_1 - 1)}$$

$$B_9 = \frac{B_5}{(B_4 B_1 - 1)}$$

$$B_{10} = B_1 B_7 - B_2$$

$$B_{11} = B_1 B_8 - B_3$$
\[ B_{12} = B_1 B_9 \]

After substituting back into eq 1 and eq 2 the differential equation for wall 1 can be written as:

\[ \frac{dT}{d\Theta} = \alpha 8TB - \alpha 9TW_1 + \alpha 10TW_2 + \alpha 11TEX \]

where

\[ \alpha 3 = \frac{h_1(2\pi r_1)L(1-\delta_1)}{m_{C_p} \text{sand}} \]

\[ \alpha 4 = \frac{h_1(2\pi r_1)L\delta_2}{m_{C_p} \text{sand}} \]

\[ \alpha 5 = h_2(2\pi r_2)L(B_7-1) + 4\delta S_7^3(B_7-B_{10}) \quad \text{(Btu/h F)} \]

\[ \alpha 6 = h_2(2\pi r_2)L(B_8) + 4\delta S_8^3(B_8-B_{11}) \quad \text{(Btu/h F)} \]

\[ \alpha 7 = h_2(2\pi r_2)LB_9 + 4\delta S_9^3(B_9-B_{12}) \quad \text{(Btu/h F)} \]

\[ \alpha 8 = \frac{\alpha 3 mCp_{\text{sand}}}{m_{bwL} C_{pL}} \]

\[ \alpha 9 = \frac{mC_p \text{sand} \alpha 4 + \alpha 6}{m_{bwL} C_{pL}} \]

\[ \alpha 10 = \frac{-\alpha 7}{m_{bwL} C_{pL}} \]

\[ \alpha 11 = \frac{-\alpha 5}{m_{bwL} C_{pL}} \]
For the heat exchanger, one can write

\[ m_{\text{pex}} \frac{d\text{TEX}}{d\Theta} = Q_{\text{in}} - Q_{\text{out}} \]  

(7)

where \( Q_{\text{in}} = h_2(2\pi r_2)L(T_{Wb} - \text{TEX}) + m_{\text{pair}} \text{TEX}_i \)

\( Q_{\text{out}} = h_3(2\pi r_3)L(\text{TEX} - T_{W2a}) + m_{\text{pair}} \text{TEX} \)

Substituting in for \( T_{W1b} \) and \( T_{W2a} \), results in the differential equation for the heat exchanger.

\[ \frac{d\text{TEX}}{d\Theta} = -\alpha_{12}\text{TEX} + \alpha_{13}T_{W1} + \alpha_{14}T_{W2} + \alpha_{15}\text{TEX}_i \]

where

\[ \alpha_{12} = \frac{h_2(2\pi r_2)L}{m_{\text{pex}}(1-B_7)} + \frac{h_3(2\pi r_3)L}{m_{\text{pex}}(1-B_{10})} + \frac{m_{\text{pair}}}{m_{\text{pex}}} \]

\[ \alpha_{13} = \frac{h_2(2\pi r_2)L}{m_{\text{pex}}} \frac{B_8}{m_{\text{pex}}} + \frac{h_3(2\pi r_3)L}{m_{\text{pex}}} \frac{B_{11}}{m_{\text{pex}}} \]

\[ \alpha_{14} = \frac{h_2(2\pi r_2)L}{m_{\text{pex}}} \frac{B_9}{m_{\text{pex}}} + \frac{h_3(2\pi r_3)L}{m_{\text{pex}}} \frac{B_{12}}{m_{\text{pex}}} \]

\[ \alpha_{15} = \frac{m_{\text{pair}}}{m_{\text{pex}}} \]

Equation 8 can be written for wall 2

\[ m_{w2}C_{pw2} \frac{dT_{W2}}{d\Theta} = Q_{\text{in}} - Q_{\text{out}} \]  

(8)

where \( Q_{\text{in}} = h_3(2\pi r_3)L(\text{TEX} - T_{W2a}) + \psi_5(T_{W1b} - T_{W2a}) \)

\( Q_{\text{out}} = 2\pi k_2L(T_{W2} - T_{\text{INSa}})/\ln(r_4/(r_3+.5t_2)) \)  

(9)
To get rid of TINSa, assume a lumped mass in the center of the
insulation:

\[ Q_{out} = (TINSb - TINSa) \left( \frac{2\pi k_3 L}{\ln(\frac{r_4 + 0.5 t_3}{r_4})} \right) \]  

(10)

Solving eq 9 and 10 simultaneously gives:

\[ TINSa = (8 BTW_2 + 89 TINS) / (88 + 89) \]

where

\[ \gamma_8 = \frac{(2\pi k_2) L}{\ln(\frac{r_4}{r_3 + 0.5 t_2})} \quad \text{(Btu/h F)} \]

\[ \gamma_9 = \frac{(2\pi k_3) L}{\ln(\frac{r_4 + 0.5 t_3}{r_4})} \quad \text{(Btu/h F)} \]

Substituting back into eq 8, the differential equation for wall 2
can be written as follows.

\[ \frac{dTW_2}{d\theta} = \alpha_16 \text{TEX} + \alpha_17 \text{TW}_1 - \alpha_18 \text{TW}_2 + \alpha_19 \text{TINS} \]

where

\[ \alpha_{16} = \frac{h_3 (2\pi r_3) L}{m_2 C_{pw2}} - \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} + \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} \]

\[ \alpha_{17} = \frac{h_3 (2\pi r_3) L}{m_2 C_{pw2}} - \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} + \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} \]

\[ \alpha_{18} = \frac{h_3 (2\pi r_3) L}{m_2 C_{pw2}} - \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} + \frac{\Delta T_0^{\gamma_5}}{m_2 C_{pw2}} + \frac{\gamma_8 \gamma_9}{m_2 C_{pw2}} \]
\[ \alpha_{19} = \frac{\alpha_{89}}{(\alpha_{8} + \alpha_{9}) m_{ins} C_{pins}} \]

Writing the energy balance for the insulation results in:

\[ m_{ins} C_{pins} \frac{dT_{INS}}{d\Theta} = Q_{in} - Q_{out} \]  \hspace{1cm} (11)

where

\[ Q_{in} = \alpha_{8} \alpha_{9} (T_{W2} - T_{INS})/(\alpha_{8} + \alpha_{9}) \]
\[ Q_{out} = \alpha_{10}(T_{INSb} - T_{W3}) \]
\[ = \alpha_{11}(T_{INS} - T_{INSb}) \] (using lumped parameter analysis)  \hspace{1cm} (12)

where

\[ \alpha_{10} = \frac{(2\pi k_{4})L}{\ln((r_{5} + 0.5t_{4})/r_{5})} \] (Btu/h F)

\[ \alpha_{11} = \frac{(2\pi k_{3})L}{\ln(r_{5}/(r_{4} + 0.5t_{3}))} \] (Btu/h F)

Solving eq 12 and eq 13

\[ T_{INSb} = (\alpha_{10} T_{W3} + \alpha_{11} T_{INS})/(\alpha_{10} + \alpha_{11}) \]

Substituting back into eq 11 the differential eq for the insulation can be written as

\[ \frac{dT_{INS}}{d\Theta} = \alpha_{20} T_{W2} - \alpha_{21} T_{INS} + \alpha_{22} T_{W3} \]

where

\[ \alpha_{20} = \frac{\alpha_{89}}{(\alpha_{8} + \alpha_{9}) m_{ins} C_{pins}} \]
\[ \alpha_{21} = \frac{\delta \delta_9}{(\delta_8 + \delta_9)} + \frac{\delta \delta_10 \delta_11}{(\delta 8 + \delta_10 + \delta_11)} \times \left(\frac{1}{m_{ins}C_{pins}}\right) \]

\[ \delta_{10} \delta_{11} \]

\[ \alpha_{22} = \frac{\delta \delta_{10}}{m_{ins}C_{pins}(\delta 10 + \delta 11)} \]

Writing the energy balance for wall 3 gives

\[ m_{w3}C_{pw3} \frac{dT_{w3}}{dT} = Q_{in} - Q_{out} \quad (14) \]

where \( Q_{in} = \delta_{10}(T_{INSb} - T_{w3}) \quad \text{Note: } (T_{INSb} = T_{w3a}) \)

\[ Q_{out} = \delta_{12}(T_{w3b} - T_{w3}) \quad (15) \]

\[ = \delta_{13}(T_{w3} - T_{w3b}) \quad \text{(using lumped mass analysis)} \quad (16) \]

where

\[ \delta_{12} = h_4(2\pi r_6) \quad \text{(Btu/h F)} \]

\[ \delta_{13} = \frac{(2\pi k_4)\delta L}{\ln (r_6/(r_5 + 0.5t_4))} \quad \text{(Btu/h F)} \]

Solving for \( T_{w3b} \) gives

\[ T_{w3b} = \frac{(\delta_{12}T_{w} + \delta_{13}T_{w3})}{(\delta_{12} + \delta_{13})} \]

Substituting back into \( Q_{in} \) and \( Q_{out} \) and eq 14 results in the differential equation for wall 3.

\[ \frac{dT_{w3}}{dT} = -\alpha_{23}T_{w3} + \alpha_{24}T_{INS} + \alpha_{25}T \]

where

\[ \alpha_{23} = \frac{\delta_{10} \delta_{11}}{m_{w3}C_{pw3}(\delta_{10} + \delta_{11})} + \frac{\delta_{12} \delta_{13}}{m_{w3}C_{pw3}(\delta_{12} + \delta_{13})} \]
\[ \alpha_{24} = \frac{\xi_{10}^6 \xi_{11}}{m_{\nu_3} c_{\nu_3} (\xi_{10} + \xi_{11})} \]

\[ \alpha_{25} = \frac{12 \ 13}{m_{\nu_3} c_{\nu_3} (\xi_{12} + \xi_{13})} \]
Appendix B

SUMMARY OF CONSTANTS IN THEORETICAL MODEL
A summary of the constants from the theoretical development of the mathematical model complete with the units associated with them will be found here. The computer program Param, used in evaluating them, can be found in Appendix E.

\[ \beta_1 = \frac{h_1(2\pi r_1)L}{(2\pi k_1)L} \]  
\[ \beta_1 = \frac{1}{\ln((r_1+.5t_1)/r_1)} \]  
\[ \beta_2 = \frac{h_1(2\pi r_1)L}{(2\pi k_1)L} \]  
\[ \beta_2 = \frac{1}{\ln((r_1+.5t_1)/r_1)} + h_1(2\pi r_1)L \]  
\[ \beta_3 = \frac{(2\pi k_1)L}{\ln(r_2/(r_1+.5t_1))} \]  
\[ \beta_4 = h_2(2\pi r_2)L \]  
\[ \beta_5 = \frac{\sigma}{1/((2\pi r_2)Le_1) + (1/((2\pi r_3)L))(1/e^2-1)} \]  
\[ \beta_6 = \frac{(2\pi k_2)L}{\ln((r_3+.5t_2)/r_3)} \]  
\[ \beta_7 = h_3(2\pi r_3)L \]  
\[ \beta_8 = \frac{(2\pi k_2)L}{\ln(r_4/(r_3+.5t_2))} \]  
\[ \beta_8 = \frac{(2\pi k_2)L}{\ln(r_4/(r_3+.5t_2))} \]
\( q_9 = \frac{(2\pi k3)L}{\ln((r4+.5t3)/r4)} \)  
\( (Btu/h \ F) \)

\( q_{10} = \frac{(2\pi k4)L}{\ln((r5+.5t4)/r5)} \)  
\( (Btu/h \ F) \)

\( q_{11} = \frac{(2\pi k3)L}{\ln(r5/(r4+.5t3))} \)  
\( (Btu/h \ F) \)

\( q_{12} = b4(2\pi r6)L \)  
\( (Btu/h \ F) \)

\( q_{13} = \frac{(2\pi k4)L}{\ln(r6/(r5+.5t4))} \)  
\( (Btu/h \ F) \)

The following "B" terms are all dimensionless.

\( B_1 = \frac{q4+q3+465To^3}{465To^3} \) where \( To \) is absolute temp, \( R \)

\( B_2 = \frac{q4}{465To^3} \)

\( B_3 = \frac{q3}{465To^3} \)

\( B_4 = \frac{q6+485To^3+q7}{465To^3} \)

\( B_5 = \frac{q6}{465To^3} \)
$B_6 = \frac{1}{\text{45}T_3}$

$B_7 = \frac{B_4B_2 + B_6}{(B_4B_1-1)}$

$B_8 = \frac{B_4B_3}{(B_4B_1-1)}$

$B_9 = \frac{B_5}{(B_4B_1-1)}$

$B_{10} = B_1B_7 - B_2$

$B_{11} = B_1B_8 - B_3$

$B_{12} = B_1B_9$

The following $\alpha$ values are used in the final differential equations. The units are (1/hr) unless otherwise noted.

$$\alpha_1 = \frac{\text{nCp in}}{\text{mCps and}}$$

$$\alpha_2 = \frac{\text{nCp out}}{\text{mCps and}}$$

$$Q_{\text{gen}} = \frac{\Delta H_f \times \text{fuel rate}}{\text{mCps and}} \quad (\text{W/hr})$$
\[ \alpha_3 = \frac{h(2n1)L(1-\delta)}{mCpsand} \]

\[ \alpha_4 = \frac{h(2n1)L \delta}{mCpsand} \]

\[ \alpha_5 = h(2n2)L(B7-1) + 4\delta T_0^3 (B7-B10) \quad (\text{Btu/h F}) \]

\[ \alpha_6 = h(2n2)L(B8) + 4\delta T_0^3 (B8-B11) \quad (\text{Btu/h F}) \]

\[ \alpha_7 = h(2n2)L B9 + 4\delta T_0^3 (B9-B12) \quad (\text{Btu/h F}) \]

\[ \alpha_8 = \frac{3mCpsand}{mW*Cpw1} \]

\[ \alpha_9 = \frac{mCpsand\alpha_4 + \alpha_6}{mW*Cpw1} \]

\[ \alpha_{10} = \frac{-\alpha_7}{mW*Cpw1} \]

\[ \alpha_{11} = \frac{-\alpha_5}{mW*Cpw1} \]

\[ \alpha_{12} = \frac{h(2n2)L}{mCpex} (1-B7) + \frac{h(2n3)L}{mCpex} (1-B10) + \frac{\dot{m}C_{pair}}{mCpex} \]

\[ \alpha_{13} = \frac{h(2n2)L B8 + h(2n3)L B11}{mCpex} \]
\[ h_2(2\pi r_2)L \ B_9 + h_3(2\pi r_3)L \ B_{12} \]
\[ \mu_{\text{Cpex}} \ \\
\mu_{\text{Cpex}} \]
\[ \mu_{\text{Cpair}} \]
\[ \mu_{\text{Cpex}} \]
\[ h_3(2\pi r_3)L \ 4\tau \ \gamma_5 \]
\[ \mu_{\text{w}2\text{Cpw}2} \ \\
\mu_{\text{w}2\text{Cpw}2} \]
\[ \mu_{\text{w}2\text{Cpw}2} \ \\
\mu_{\text{w}2\text{Cpw}2} \]
\[ h_3(2\pi r_3)L \ 4\tau^3 \ \gamma_5 \]
\[ \mu_{\text{w}2\text{Cpw}2} \ \\
\mu_{\text{w}2\text{Cpw}2} \]
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\mu_{\text{w}2\text{Cpw}2} \]
\[ \mu_{\text{w}2\text{Cpw}2} \ \\
\mu_{\text{w}2\text{Cpw}2} \]
$\alpha_{24} = \frac{w_{0611}}{m_{3c_{p_{w3}}} (\xi_{10} + \xi_{11})}$

$\alpha_{25} = \frac{\xi_{12} \xi_{13}}{m_{3c_{p_{w3}}} (\xi_{12} + \xi_{13})}$
Appendix C

EVALUATION OF PROGRAM PARAMETERS IN THEORETICAL MODEL
The parameters required in the derivation of the differential equations are defined and evaluated below.

**Heat Transfer Coefficients (BTU/h F ft)**

- \( h_1 = 25,000 \) (bed-wall 1)
- \( h_2 = 9 \) (wall 1-exchanger)
- \( h_3 = 9 \) (exchanger-wall 2)
- \( h_4 = 4.5 \) (wall 3-surroundings)

The heat transfer coefficients were obtained from H. Keener's previous work via personal communication. A sample derivation is given for \( h_2 \). The transfer between the bed and wall 1 is so large because of the nature of fluidized beds.

From the figure in Kreith (p. 334) of Nusselt number vs. Reynolds number for air flowing in a pipe, the following equations were developed.

\[
\begin{align*}
\text{if } Re &< 2100 \quad \text{then } \text{nue} = .453 \times Re^{.3} \\
\text{if } 2100 < Re < 8000 \quad \text{then } \text{nue} = .5 \times (\text{nue} + .02 \times Re^{.8}) \\
\text{if } Re > 8000 \quad \text{nue} = .02 \times Re^{.8}
\end{align*}
\]

where \( Re = \text{Reynolds Number} \)

\( Nu = \text{Nusselt Number} \)

and \( Re = \frac{(4 \times \text{dmas})}{(\pi \times \text{u})} \)

where \( \text{dmas} = \text{flowrate of air} = 300 \text{ lb/hr} \)

\( \text{ss} = \text{surface area} = (d_1 + d_2) \)

\( u = .000025 + .00000001 \times (t_{air} - 100) \text{ lb/(ftsec)} \)

Using \( \text{dmas} = 300 \) and \( \text{ss} = 3.55 \text{ ft} \), a Reynolds number of 3800 is obtained which gives a Nusselt number of 9.99.

The heat transfer coefficient, \( hc \), is equal to
hc = \( x_k \times \frac{x_{nu}}{dhs} \)

where \( x_k \) = thermal conductivity of the air

\[ = 0.0314 + 0.01625(t_{air} - 1000)/1000 \]

dhs = hydraulic diameter

\[ = 4 \text{ as/ ss} \]

where as = cross sectional area.

Evaluating these terms using the dimensions given previously, a heat transfer coefficient of 12.1 Btu/hr°F ft is obtained. This value varies depending on flow rate and a value of 9 is used in the analysis.

<table>
<thead>
<tr>
<th>Thermal Conductivities (Btu/hr°F ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td>k1 = 9.4</td>
</tr>
<tr>
<td>k2 = 9.4</td>
</tr>
<tr>
<td>k3 = 0.058</td>
</tr>
<tr>
<td>k4 = 30</td>
</tr>
</tbody>
</table>

The thermal conductivity values for the steel walls were found in Perry and Chilton, 1973, pp. 23-38, 39. The perlite insulation conductivity was given in Perlite, 1983.

Dimensions of Burner (ft)

The dimensions of the burner were given previously in section Table 1. The values were obtained from Drawing FEC-4A, OARDC FEC Project-FBC Vessel, 8.81, RIA, as well as from personal communication with J. Henry and H. Keener. (Note: Patent filed September 24, 1984.) It should be noted that the heat exchanger outer wall is tapered from 1/8" at the top to 3/16" at the bottom. A discussion with H. Keener
revealed it was sufficient to use the average value of \((1/8 + 3/16)/2\)" for the thickness of the heat exchanger.

**Masses (lb) and Heat Capacities (BTU/lb F)**

The amount of sand in the bed varies because of loss of fines and blow overs into the cyclone, but the bed was maintained at approximately 36 lb.

The steel masses were calculated from volume and density data. The density and heat capacities for steel were found in Perry and Chilton, pp. 23-38, 39. The perlite data was taken from Perlite, 1983 and the sand data from Bauer, 1961.

**Sand**
- \(w_{msand} = 36\)
- \(C_{psand} = .194\)

**Wall 1 (316 ss)**
- \(\text{density} = .29\)#/in
- \(\text{volume} = 334.9\) in
- \(\text{mv1} = 97.11\) lb
- \(C_{pw1} = .12\) BTU/lb F

**Wall 2 (304ss)**
- \(\text{density} = .29\)#/in
- \(\text{volume} = 103.1\) in
- \(\text{mv2} = 29.91\) lb
- \(C_{pw2} = .12\)

**Wall 3 (mild steel)**
- \(\text{density} = .284\)#/in
- \(\text{volume} = 341.1\) in
- \(\text{mv3} = 96.9\) lb
- \(C_{pw3} = .107\)

**Insulation**
- \(w_{mins} = 150\) lb
- \(C_{pins} = .21\)
Emissivities

\[ e_1 = 0.65 = \text{emissivity} \]
\[ e_2 = 0.65 = \text{emissivity} \]
\[ \sigma = 0.1714 \times 10^{-8} \text{ Btu/hr ft}^2 \text{ R}^4 \]

The emissivity value for radiation from wall 1 and wall 2 to the heat exchanger was found in Bird, Stewart, and Lightfoot, 1960, p. 432; and Perry and Chilton, 1973, pp. 10-46.

The operating temperature, \( T_0 \), was taken to be 1824°F, and all temperatures were used in degrees R.

The heat capacity of air (Kreith, 1963, p. 35) was taken as 0.24 Btu/lb°F for both that of the exchanger and that in the bed. The variation of heat capacity with temperature was ignored for this study.

The mass flow rate of the cooling air, \( \dot{m}_{\text{air}} \), was taken at a typical value of 200 lb/hr while the mass of air in the exchanger, \( m_{\text{ex}} \), was calculated as 0.00312 lb using the volume of the exchanger and density of air at 1824°F.

The sum of heat capacities of constituents in and out of the bed, \( \Sigma c_{\text{pin}} \) and \( \Sigma c_{\text{pout}} \), and the heat of combustion were calculated as follows. When beginning the theoretical analysis on the burner, the following energy balance was performed for the combustion chamber itself.

\[ \text{storage} = (\text{enthalpy in-enthalpy out}) + \text{generation} - \text{Qwall loss} \]

The change in enthalpy for a chemical reaction can be written as

\[ \Delta H_1 - \Delta H_2 = \Sigma \Delta N (h_1 - h_0) - H_r - \Sigma N (h_2 - h_0) \]

where \( \Delta H_1 - \Delta H_2 \) = the change in enthalpy

\( N \) = moles of each constituent
\[ h = \text{enthalpy of constituent at inlet or outlet} \]
\[ h_0 = \text{enthalpy of constituents at a reference temperature} \]
\[ H_r = \text{heat of combustion at reference temperature} \]

The reference temperature used is 537°F. The enthalpies, \( h_1 \) and \( h_2 \), can be determined by multiplying the average specific heat of each constituent by the difference in temperature from that of the reference, either \( (T_{B_i} - 537) \) or \( (T_B - 537) \), since all combustion products are assumed to leave at the burner temperature.

The combustion reactions under consideration were:

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

The constituents entering the burner include the air, \( O_2, N_2 \), and its moisture, the fuel, \( C, N_2, O \), etc, and the products leaving will include \( CO_2, N_2, \text{ash, water, etc.} \). There will be no unburned carbon or CO leaving because it is assumed that the fuel is being completely combusted.

Stoichiometric air is that air which is required for complete combustion of a fuel with no excess oxygen in the exhaust. This assumes that the ash, nitrogen, etc. in the fuel are inert constituents. The ultimate analysis data is given by Keener, et al., (1981).
In addition to the chemical analysis, a differential thermal analysis (DTA) and thermogravimetric analysis (TGA) had been run. These analyses indicate how heat and gases are released at various temperatures. For coals, it was found the volatiles are released at temperatures of 300 to 400°C. The ash samples showed minor endotherms at 590 to 800°C, indicating that low melting point compounds are present and that serious slagging of the ash could be expected at temperatures above 760°C (Keener, et al., 1981).

The following balances were done to determine stoichiometric air requirements on a per lb of fuel basis.

\[
\text{Theoretical} \quad \text{O}_2 \text{ required per lb of fuel}
\]

\[
\begin{align*}
\text{C} + \text{O}_2 & \rightarrow \text{CO}_2 \quad \text{.0403 moles} \\
2\text{H}_2 + \text{O}_2 & \rightarrow 2\text{H}_2\text{O} \quad \text{.0139 moles}
\end{align*}
\]

The \text{O}_2 available from the fuel is .0138 moles, so the net \text{O}_2 requirement is .04039 moles \text{O}_2/lb fuel.

The desired level of excess air is 20%. This is desirable since burner efficiency is greatest in this range (Keener, et al., 1981). If 20% excess air is required, then 1.55 lb \text{O}_2, or .0484 moles \text{O}_2/lb fuel which is .2304 moles air/lb fuel or 6.7 lb air/lb fuel.

To do the heat capacities, the following formulas were used from Perry and Chilton, 1973, pp. 3-119. The average heat capacity was used and is defined as

\[
C_p = \frac{\int_T^T \text{C}_p \text{ d}T}{T - T_{\text{ref}}} \quad \text{(cal/mol K)}
\]
Oxygen \[ C_p = 8.27 + 0.000258T - 187700/T^{**2} \]

Nitrogen \[ C_p = 6.5 + 0.00100T \]

CO \[ C_p = 6.6 + 0.00120T \]

H₂ \[ C_p = 6.62 + 0.00081T \]

H₂O \[ C_p = 8.22 + 0.00015T + 0.00000134T^{**2} \]

CO₂ \[ C_p = 10.34 + 0.00274T - 195500/T^{**2} \]

C \[ C_p = 2.673 + 0.002617T - 116900/T^{**2} \]

The values in Table C1 were calculated from the integrated form of the above equations.

Table C1. Values of \( \int C_p \, dt \) (cal/mol)

<table>
<thead>
<tr>
<th>Temperature (K)</th>
<th>CO₂</th>
<th>N₂</th>
<th>CO</th>
<th>H₂</th>
<th>H₂O</th>
<th>CO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>26 = 299.15</td>
<td>6.24</td>
<td>6.8</td>
<td>6.95</td>
<td>6.86</td>
<td>8.67</td>
<td>8.97</td>
</tr>
<tr>
<td>30 = 303.15</td>
<td>31.35</td>
<td>34.0</td>
<td>34.8</td>
<td>34.32</td>
<td>43.4</td>
<td>45.0</td>
</tr>
<tr>
<td>35 = 308.15</td>
<td>63.05</td>
<td>68.03</td>
<td>69.64</td>
<td>68.66</td>
<td>86.7</td>
<td>90.43</td>
</tr>
<tr>
<td>720 = 993.15</td>
<td>5422.9</td>
<td>4966.2</td>
<td>5125.5</td>
<td>4964.40</td>
<td>6386.0</td>
<td>7956.9</td>
</tr>
<tr>
<td>740 = 1013.15</td>
<td>5589.7</td>
<td>5116.3</td>
<td>5281.5</td>
<td>5113.0</td>
<td>6580.5</td>
<td>8214.8</td>
</tr>
<tr>
<td>760 = 1033.15</td>
<td>5756.8</td>
<td>5266.8</td>
<td>54381.1</td>
<td>5262.00</td>
<td>6775.6</td>
<td>8474.0</td>
</tr>
</tbody>
</table>

Finally the average heat capacities, Table C2, were calculated.
Table C2. Average Heat Capacities, (cal/mol*K)

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>T-25</th>
<th>O2</th>
<th>N2</th>
<th>CO</th>
<th>H2</th>
<th>H2O</th>
<th>CO2</th>
<th>C</th>
</tr>
</thead>
<tbody>
<tr>
<td>26</td>
<td>1</td>
<td>6.24</td>
<td>6.8</td>
<td>6.96</td>
<td>6.86</td>
<td>8.67</td>
<td>8.97</td>
<td>2.14</td>
</tr>
<tr>
<td>30</td>
<td>5</td>
<td>6.27</td>
<td>6.8</td>
<td>6.96</td>
<td>6.86</td>
<td>8.68</td>
<td>9.0</td>
<td>2.17</td>
</tr>
<tr>
<td>35</td>
<td>10</td>
<td>6.31</td>
<td>6.8</td>
<td>6.96</td>
<td>6.86</td>
<td>8.67</td>
<td>9.04</td>
<td>2.19</td>
</tr>
<tr>
<td>740</td>
<td>715</td>
<td>7.82</td>
<td>7.16</td>
<td>7.39</td>
<td>7.15</td>
<td>9.20</td>
<td>11.49</td>
<td>4.02</td>
</tr>
<tr>
<td>760</td>
<td>735</td>
<td>7.83</td>
<td>7.17</td>
<td>7.40</td>
<td>7.16</td>
<td>9.22</td>
<td>11.53</td>
<td>4.05</td>
</tr>
</tbody>
</table>

Figure C1 illustrates the constituents entering and leaving the bed. The water in the combustion air, ash and other components were ignored except in the energy balances. See the discussion, Chapter 7, for an alternative approach.

![Figure C1. Constituent Flow In and Out of Bed.](image-url)
The enthalpy table is given in Table C3.

Table C3. Enthalpy Values.

<table>
<thead>
<tr>
<th>element</th>
<th>mol/lb fuel</th>
<th>Cp</th>
<th>nCp</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>cal/mol K</td>
<td>(BTU/F)/(lb fuel)</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>.0403</td>
<td>2.17</td>
<td>.27</td>
</tr>
<tr>
<td>N2</td>
<td>-----</td>
<td>-----</td>
<td>-----</td>
</tr>
<tr>
<td>fuel</td>
<td>2H2</td>
<td>.01389</td>
<td>6.86</td>
</tr>
<tr>
<td>S</td>
<td>-----</td>
<td>-----</td>
<td>-----</td>
</tr>
<tr>
<td>O2</td>
<td>.0138</td>
<td>6.27</td>
<td></td>
</tr>
<tr>
<td>air in</td>
<td>O2</td>
<td>.0484</td>
<td>6.27</td>
</tr>
<tr>
<td></td>
<td>N2</td>
<td>.182</td>
<td>6.8</td>
</tr>
<tr>
<td>prod.</td>
<td>CO2</td>
<td>.0403</td>
<td>11.49</td>
</tr>
<tr>
<td>out</td>
<td>2H2O</td>
<td>.01389</td>
<td>9.2</td>
</tr>
<tr>
<td></td>
<td>O2</td>
<td>.0097</td>
<td>7.82</td>
</tr>
<tr>
<td></td>
<td>N2</td>
<td>.182</td>
<td>7.16</td>
</tr>
</tbody>
</table>

Since the net enthalpy change for the constituents in and out of the bed is so small, the gross heating value, Hr=8000 BTU/lb, was used to represent the energy change in the bed, except in the simulation model where the above numbers were used.
Appendix D

MATRIX SOLUTIONS
The following are the solutions to the matrices $[D]$, $[D1]$, and $[D2]$ 
where $TB = [D1]/[D]$ and $TEX = [D2]/[D]$.

$$D = sD^3 + bD^2 + cD + e$$

where

$$a = \alpha_{18}K_1 + \alpha_{19}K_2$$

$$b = (\alpha_{18}(\alpha_9 + \alpha_{12} + \alpha_{2} + \alpha_{3}) - \alpha_{14}\alpha_{16} - \alpha_{10}\alpha_{17})K_1 + (\alpha_{19}(\alpha_9 + \alpha_{12} + \alpha_{2} + \alpha_{3}))K_2$$

$$c = (\alpha_{18}\alpha_{12}\alpha_{18} - \alpha_{9}\alpha_{14}\alpha_{16} - \alpha_{11}\alpha_{13}\alpha_{14} - \alpha_{11}\alpha_{14}\alpha_{17} - \alpha_{12}\alpha_{13}\alpha_{16} - \alpha_{16}\alpha_{17}\alpha_{12} - \alpha_{14}\alpha_{8}\alpha_{18} + (\alpha_2 + \alpha_3)(\alpha_{18}\alpha_{9} + \alpha_{18}\alpha_{12} - \alpha_{14}\alpha_{16} - \alpha_{10}\alpha_{17})K_1 + (\alpha_{9}\alpha_{12}\alpha_{19} - \alpha_{17}\alpha_{12}\alpha_{19} + (\alpha_2 + \alpha_3)(\alpha_{18}\alpha_{9} + \alpha_{19}\alpha_{12} - \alpha_{14}\alpha_{19})K_2$$

$$e = ((\alpha_2 + \alpha_3)(\alpha_{9}\alpha_{12}\alpha_{18} - \alpha_{9}\alpha_{14}\alpha_{16} - \alpha_{11}\alpha_{13}\alpha_{18} - \alpha_{11}\alpha_{14}\alpha_{17} - \alpha_{10}\alpha_{13}\alpha_{16} - \alpha_{10}\alpha_{17}\alpha_{12}) - \alpha_{14}\alpha_{8}\alpha_{18}\alpha_{12} + \alpha_{14}\alpha_{8}\alpha_{14}\alpha_{16})K_1 + (\alpha_{9}\alpha_{12}\alpha_{19} - \alpha_{11}\alpha_{13}\alpha_{19} - \alpha_{14}\alpha_{8}\alpha_{19}\alpha_{12})K_2$$

$$K_1 = \alpha_{18}\alpha_{12}-\alpha_{14}\alpha_{17}$$

$$K_2 = -\alpha_{20}\alpha_{23}$$

$$D1 = (MD^2 + ND + P) F_1 + R F_2 + (SD + T) F_3$$

where

$$F_1 = \alpha_{18}537 + \alpha_{1}(TB_1 - 537) + Qgen = (\alpha_{2} - \alpha_{1})537 + TB_1 + Qgen$$

$$F_2 = \alpha_{15}TEX_i$$

$$F_3 = \alpha_{25}T$$

$$M = K_1(\alpha_{18}) + K_2(\alpha_{19})$$

$$N = (\alpha_{18}(\alpha_9 + \alpha_{12}) - \alpha_{14}\alpha_{16} - \alpha_{10}\alpha_{17})K_1 + K_2(\alpha_9 + \alpha_{12})$$

$$P = (\alpha_{18}\alpha_{9}\alpha_{12} - \alpha_{9}\alpha_{14}\alpha_{16} - \alpha_{11}\alpha_{13}\alpha_{18} - \alpha_{11}\alpha_{14}\alpha_{17} - \alpha_{10}\alpha_{13}\alpha_{16} - \alpha_{10}\alpha_{14}\alpha_{12})K_1 + (\alpha_{9}\alpha_{9}\alpha_{12} - \alpha_{11}\alpha_{13}\alpha_{19})K_2$$
\[ R = (\alpha_6 \alpha_{11} \alpha_{18} + \alpha_4 \alpha_{10} \alpha_{16})K_1 + (\alpha_4 \alpha_{11} \alpha_{19})K_2 \]

\[ S = \alpha_4 \alpha_{10} \alpha_{19} \alpha_{22} \]

\[ T = \alpha_4 \alpha_{11} \alpha_{14} \alpha_{19} 22 + \omega \alpha_4 \alpha_{10} \alpha_{19} \alpha_{12} \alpha_{22} \]

\[ K_1 = \alpha_{21} \alpha_{23} - \alpha_{22} \alpha_{24} \]

\[ K_2 = -\alpha_{20} \alpha_{23} \]

and the \( \alpha' \)'s are as defined in Appendix B.

\[
DZ = VF_1 + (WD^2 + XD + Y)F_2 + (UD^2 + ZD + Q)F_3
\]

where

\[ V = K_1(\alpha_8 \alpha_{13} \alpha_{18} + \alpha_8 \alpha_{14} \alpha_{17}) + K_2(\alpha_8 \alpha_{13} \alpha_{19}) \]

\[ W = \alpha_{18} K_1 + \alpha_{19} K_2 \]

\[ X = K_1(\alpha_1 \alpha_8 \alpha_9 - \alpha_1 \alpha_7 + (\alpha_2 + \alpha_3) \alpha_8) + \]
\[ + K_2(\alpha_1 \alpha_9 \alpha_9 + (\alpha_2 + \alpha_3)(\alpha_1 \alpha_9)) \]

\[ Y = K_1((\alpha_2 + \alpha_3)(\alpha_1 \alpha_8 \alpha_9 - \alpha_1 \alpha_7) + \alpha_4 \alpha_8 \alpha_{18}) \]
\[ + K_2((\alpha_2 + \alpha_3)(\alpha_1 \alpha_9 \alpha_9) - \alpha_4 \alpha_8 \alpha_{19}) \]

\[ U = \alpha_{22} \alpha_{19} \alpha_{14} \]

\[ Z = \alpha_{22} \alpha_{19} \alpha_{14} \alpha_{19} + \alpha_{22} \alpha_{10} \alpha_{13} \alpha_{19} + \alpha_{22} \alpha_{19} \alpha_{14} \alpha_{19} \]

\[ Q = (\alpha_{22} \alpha_{19})(\alpha_2 + \alpha_3)(\alpha_{14} \alpha_9 + \alpha_{10} \alpha_{13}) - \alpha_{22} \alpha_{8} \alpha_{14} \alpha_{19} \alpha_4 \]

\[ F_1 = 537(\alpha_2 - \alpha_1) + \alpha_1 TB_i + Q_{gen} \]

\[ F_2 = \alpha_{15} T \]

\[ F_3 = \alpha_{25} T_{\infty} \]

\[
TB = \frac{(MD^2 + ND + P)F_1 + RF_2 + (SD + T)F_3}{aD^3 + bD^2 + cD + e}
\]

\[
TEX = \frac{VF_1 + (WD^2 + XD + Y)F_2 + (UD^2 + ZD + Q)F_3}{aD^3 + bD^2 + cD + e}
\]
Appendix E

SIMULATION MODEL PROGRAMS
PARAM

$CONTROL USLUNIT,NOSOURCE
C PROGRAM TO CALCULATE PARAMETERS IN EQUATIONS
DIMENSION H(4),TC(4),R(6),T(4),WM(5),CP(5)
DATA H/25000.,9.,9.,4.5/
DATA TC/9.,4.,9.,4.,058,30/
DATA R/1257.,2760.,289.,2956.,9635.,970/
DATA T/023.,0065.,668.,0065/
DATA WM/36.,97.21,29.91,96.89,150/
DATA CP/194.,12.,12.,107.,21/
DISPLAY"INPUT---FUEL RATE"
ACCEPT FUEL,,
L=5
SIGMA=.1714*(10.**(-8))
T=1824
SCF1N=1.88*FUEL,,
SCFOUT=.97*FUEL,,
HF=8000*FUEL,,
E1=.65
E2=.65
WMAIR=200.
WMEX=00312
CPEX=.24
CPAIR=.24
G1=H(1)*2*3.141*R(1)*L/(2*3.141*TC(2)*L)/ALOG((R(1)+.5*T(1))/
#R(1))+H(1)*2*3.141*R(1)*L)
G2=2*3.141*TC(1)*L/ALOG((R(1)+.5*T(1))/R(1))/(2*3.141*TC(1)*
#L/ALOG((R(1)+.5*T(1))/R(1)))+H(1)*2*3.141*R(1)*L)
G3=2*3.141*TC(1)*L/ALOG(R(2)/(R(1)+.5*T(1)))
G4=H(2)*2*3.141*R(2)*L
G5=SIGMA/(1/(2*3.141*R(2)*L)*E1)+1/(2*3.141*R(3)*L)*(1/E2-1)
G6=2*3.141*TC(2)*L/ALOG((R(3)+.5*T(2))/R(3))
G7=H(3)*2*3.141*R(3)*L
G8=2*3.141*TC(3)*L/ALOG(R(4)/(R(3)+.5*T(2)))
G9=2*3.141*TC(3)*L/ALOG(R(4)+.5*T(3))/R(4))
G10=1*3.141*TC(4)*L/ALOG((R(5)+.5*T(4))/R(5))
G11=2*3.141*TC(3)*L/ALOG(R(5)/(R(4)+.5*T(3)))
G12=H(4)*2*3.141*R(6)*L
G13=2*3.141*TC(4)*L/ALOG(R(6)/(R(5)+.5*T(4)))
WRITE,(*,*)G1,G2,G3,G4,4,G5,G6,G7,G8,G9,G10,G11,G12,G13
B1=(G4+G3+G4*G5*0**3)/(4*G5*0**3)
B2=G4/(4*G5*0**3)
B3=G3/(4*G5*0**3)
B4=(G6+G5*0**3+G7)/(4*G5*0**3)
B5=G6/(4*G5*0**3)
B6=G7/(4*G5*0**3)
B7=(B4*B2+B6)/(B4*B1-1)
B8=(B4*B3)/(B4*B1-1)
B9=B5/(B4*B1-1)
B10=B1*B7-B2
B11=B1*B8-B3
B12=B1*B9
WRITE(6,*)(B1,B2,B3,B4,B5,B6,B7,B8,B9,B10,B11,B12)
A1=SCPINC(/(WM(1)*CP(1)))
A2=SCPDCP(/(WM(1)*CP(1))
QGDN=HF/(WM(1)*CP(1))
A3=H(1)*2*3.141*R(1)*L*(1-G1)/(WM(1)*CP(1)))
A4=H(1)*2*3.141*R(1)*L*G2/(WM(1)*CP(1))
A5=H(2)*2*3.141*R(2)*L*(B7-1)+4*G5*(T0**3)*(B7-B10)
A6=H(2)*2*3.141*R(2)*L*B8+4*G5*(T0**3)*(B8-B11)
A7=H(2)*2*3.141*R(2)*L*B9+4*G5*(T0**3)*(B9-B12)
A8=3/(WM(2)*CP(2))*(WM(1)*CP(1))
A9=(WM(1)*CP(1)*A4+A6)/(WM(2)*CP(2))
A10=-A7/(WM(2)*CP(2))
A11=-A5/(WM(2)*CP(2))
A12=(H(2)*2*3.141*R(2)*L*(1-B7)+H(3)*2*3.141*R(3)*L*(1-B10)+
#WMAIR*CPAIR)/(WMEX*CPEX)
A13=(H(2)*2*3.141*R(2)*L*B8+H(3)*2*3.141*R(3)*L*B11)/(WMEX*
#CPEX)
A14=(H(2)*2*3.141*R(2)*L*B9+H(3)*2*3.141*R(3)*L*B12)/(WMEX*
#CPEX)
A15=WMAIR*CPAIR/(WMEX*CPEX)
A16=(H(3)*2*3.141*R(3)*L*(1-B10)+4*(T0**3)*G5*(B7-B10))/
#(WM(3)*CP(3))
A17=(H(3)*2*3.141*R(3)*L*B11+(-1)+4*(T0**3)*G5*(B8-B11))/
#(WM(3)*CP(3))
A18=H(3)*2*3.141*R(3)*L*(B12)/(WM(3)*CP(3)) + 4*(T0**3)*G5
#*(B12-B9)/(WM(3)*CP(3))*GB/(WM(3)*CP(3))+(1-(G8/(G8+G9))
A19=G8/(WM(3)*CP(3))*G9/(G8+G9)
A20=G8/(WM(5)*CP(5))*(1-G8/(G8+G9))
A21=(G8*G9/(G8+G9)+G10*G11/(G10+G11))/(WM(5)*CP(5))
A22=(c10/(G10+G11)-1)*(-1)*c10/(WM(5)*CP(5))
A23=(G10*(1-G10/(G10+G11))+G12*G13/(G12+G13))/(WM(4)*CP(4))
A24=G10*G11/(G10+G11)/(WM(4)*CP(4))
A25=G12*(1-G12/(G12+G13))/(WM(4)*CP(4))
WRITE(6,*)(A1,A2,A3,A4,A5,A6,A7,A8,A9,A10,A11,A12,A13,A14
WRITE(6,*)(A15,A16,A17,A18,A19,A20,A21,A22,A23,A24,A25
XX1=A21*A23-A22*A24
XX2=(-1)*A20*A23
AA=A18*XX1+AA*XX2
BB=(A18*(A9+A12+A2+A3)-A14*A16-A10*A17)*XX1+(A19*(A9+A12+
#A2+A3))*XX2
#A16-A10*A17*A12-A4*A8+A18+(A2+A3)*(A18*A9+A18*A12-A14*A16-
#A12-A4*A8*A19))*XX2
EE = ((A2*A3)*(A9*A12*A18-A9*A14*A16-A11*A13*A18-A11*A14*A17-
A10*A13*A16-A10*A17*A12)-A4*A8*A18*A12+A4*A8*A14*A16)*XX1+
WRITE(6,*)(AA, BB, CC, EE
XMM=XX1*A18+XX2*A19
XNN=(A18*(A9+A12)-A14*A16-A10*A17)*XX1+XX2*A13*(A9+A12)
XPP=(A18*A9*A12-A9*A14*A15-A11*A13*A18-A11*A14*A17-A10*A13-
A16-A10*A17*A12)*XX1+(A19*A9*A12-A11*A13*A19)*XX2
RR=(A4*A11*A18+A4*A10*A16)*XX1+A4*A11*A19*XX2
SS=A4*A10*A19*A22
TT=A4*A11*A14*A19*A22+A4*A10*A19*A12*A22
WRITE(6,*)(XMM, XNN, XPP, RR, SS, TT
WRITE(6,*)(A2, A1, A15, A25, QGEN
STOP
END
The following additions to the end of the program, Param, were made to do the describing function analysis and the Routh Criteria Stability Analysis. They are Describ and Propcon, respectively.

**DESCRIPT**

```
TAU1 = .00083
TAU2 = .017
DO 100 I=1,100
    W=700 + I
    W2=W*W
    W3=W2*W
    W4=W2*W2
    XRR = AA*TAU2*W4 - CC*TAU2*W2 - BB*W2 + EE
    XRS = TAU2*EE*W + CC*W - TAU2*BB*W3 - AA*W3
    XRV = QGEN*(TAU1*XNN*W2 + XPP - XMM*W2)
    XRW = QGEN*(TAU1*XMM*W3 - TAU1*XPP*W + XNN*W)
    XREAL = (XRV*XRR + XRW*XRS)/(XRR*XRR + XRS*XRS)
    XIMAG = (XRW*XRR - XRV*XRS)/(XRR*XRR + XRS*XRS)
    PHASE=ATAN(XIMAG/XREAL)
    XMAG=(XREAL*XREAL+XIMAG*XIMAG)**.5
    WRITE(6,*) W,XMAG,PHASE,XIMAG,XREAL
100 CONTINUE
    GO TO 10
900 STOP
END
```
PROPCON

TAU1 = .00083
TAU2 = .017
WRITE(6,*)A2,A1,A15,A25,QGEN
DO 100 I=1,1000
K=I*2
RCA = TAU2 * AA
RCB = TAU2 * BB+AA-K*QGEN*TAU1*XMM
RCC = TAU2*CC +BB-K*QGEN*TAU1*XNN + K*QGEN*XMM
RCD = TAU2*EE+CC-K*QGEN*TAU1*XPP+K*QGEN*XNN
RCE = EE +K *QGEN*XPP
FIRST = (RCB*RCC - RCA*RCD)/RCB
SECOND=(FIRST*RCD-RCB*RCE)/FIRST
THIRD = RCE
WRITE(6,*)K,RCA,RCB,FIRST,SECOND,THIRD
100 CONTINUE
STOP
END
SIMMOD

$CONTROL USLINIT,NOSOURCE
C THIS PROGRAM SIMULATES THE FBC SYSTEM.
CHARACTER*30 TITLE
DIMENSION H(4), TC(4), R(6), T(4), WM(5), CP(5)
DATA H/25000.,9.,9.,4.5/
DATA TC/9.4,9.4,.058,.30/
DATA R/.2527,.2760,.289,.2956,.9635,.970/
DATA T/.023,.0065,.668,.0065/
DATA WM/36.,97.11,29.91,96.89,150./
DATA CP/194.,12.,12.,107.,21/
C CONSTANTS
TAU2= .017
C INPUT
DISPLAY "INPUT--† TITLE"
ACCEPT TITLE
DISPLAY "INPUT--†FINTIM,DELT,OUTDEL"
ACCEPT FINTIM,DELT,OUTDEL
DISPLAY "INPUT--†TIN,TAIR,TSURR"
ACCEPT TIN,TAIR,TSURR
DISPLAY "INPUT--†FUEL RATE"
ACCEPT FUELR
C EVALUATE PARAMETERS
C PROGRAM TO CALCULATE PARAMETERS IN EQUATIONS
L=5
SIGMA=.1714*(10.**(-8))
T0=1824
SCPIN=1.88*FUELR
SCPOUT=1.97*FUELR
HF=8000.*FUELR
E1=.65
E2=.65
WMAIR=200.
WMEX=.00312
CPFX=.24
CPAIR=.24
G1=H(1)*2.3.141*R(1)*L/(2*3.141*TC(1)*L)*ALOG((R(1)+.5*T(1))/
#R(1))+H(1)*2*3.141*R(1)*L
G2=2*3.141*TC(1)*L*/ALOG((R(1)+.5*T(1))/R(1))/(2*3.141*TC(1)*
#L*/ALOG((R(1)+.5*T(1))/R(1))+H(1)*2*3.141*R(1)*L
G3=2*3.141*TC(1)*L*/ALOG(R(2)/(R(1)+.5*T(1)))
G4=H(2)*2*3.141*R(2)*L
G5=SIGMA/(1/(2*3.141*R(2)*L*E1)+1/(2*3.141*R(3)*L)*/(1/E2-1))
G6=2*3.141*TC(2)*L*/ALOG((R(3)+.5*T(2))/R(3))
G7=H(3)*2*3.141*R(3)*L
G8=2*3.141*TC(2)*L*/ALOG(R(4)/(R(3)+.5*T(2)))
G9=2*3.141*TC(3)*L/ALOG((R(4)+.5*T(3))/R(4))
G10=2*3.141*TC(4)*L/ALOG((R(5)+.5*T(4))/R(5))
G11=2*3.141*TC(3)*L/ALOG(R(5)/(R(4)+.5*T(3)))
G12=R(4)*2*3.141*R(6)*L
G13=2*3.141*TC(4)*L/ALOG(R(6)/(R(5)+.5*T(4)))
B1=(G4+G2+4*G5*TO**3)/(4*G5*TO**3)
B2=G4/(4*G5*TO**3)
B3=G3/(4*G5*TO**3)
B4=(G6+4*G5*TO**3+G7)/(4*G5*TO**3)
B5=G6/(4*G5*TO**3)
B6=G7/(4*G5*TO**3)
B7=(B4*B2+B6)/(B4*B1-1)
B8=(B4*B3)/(B4*B1-1)
B9=B5/(B4*B1-1)
B10=B1*B7-B2
B11=B1*B8-B3
B12=B1*B9
A1=SCPIN/(WM(1)*CP(1))
A2=SCPOUT/(WM(1)*CP(1))
QGEN=HF/(WM(1)*CP(1))
A3=H(1)*2*3.141*R(1)*L*(1-G1)/(WM(1)*CP(1))
A4=H(1)*2*3.141*R(1)*L*G2/(WM(1)*CP(1))
A5=H(2)*2*3.141*R(2)*L*(B7-1)+4*G5*(TO**3)*(B7-B10)
A6=H(2)*2*3.141*R(2)*L*B8+4*G5*(TO**3)*(B8-B11)
A7=H(2)*2*3.141*R(2)*L*B9+4*G5*(TO**3)*(B9-B12)
A8=A3/(WM(2)*CP(2))*(WM(1)*CP(1))
A9=WM(1)*CP(1)*A4*A6/(WM(2)*CP(2))
A10=A7/(WM(2)*CP(2))
A11=A5/(WM(2)*CP(2))
A12=(H(2)*2*3.141*R(2)*L*(1-B7)+H(3)*2*3.141*R(3)*L*(1-B10)+
#WMAIR*/CPAIR*/(WMEX*CPFX)
A13=(H(2)*2*3.141*R(2)*L*B8+H(3)*2*3.141*R(3)*L*B11)/(WMEX*
#CPFX)
A14=(H(2)*2*3.141*R(2)*L*B9+H(3)*2*3.141*R(3)*L*B12)/(WMEX*
#CPFX)
A15=WMAIR*/CPAIR*/(WMEX*CPFX)
A16=(H(3)*2*3.141*R(3)*L*(1-B10)+4*(TO**3)*G5*(B7-B10))/
#(WM(3)*CP(3))
A17=(H(3)*2*3.141*R(3)*L*B11*(-1)+4*(TO**3)*G5*(B8-B11))/
#(WM(3)*CP(3))
A18=H(3)*2*3.141*R(3)*L*(B12)/(WM(3)*CP(3))+4*(TO**3)*G5
#*(B12-B9)/(WM(3)*CP(3))+G8/(WM(3)*CP(3))*(1-(G8/(G8+G9))
A19=G8/(WM(3)*CP(3))*G9/(G8+G9)
A20=G8/(WM(5)*CP(5))*(-1-G8/(G8+G9))
A21=(G8*G9/(G8+G9)+G10*G11/(G10+G11))/(WM(5)*CP(5))
A22=(G10/(G10+G11)-1)*(-1)*G10/(WM(5)*CP(5))
A23=(G10*(1-G10)/(G10+G11)+G12*G13/(G12+G13))/(WM(4)*CP(4))
A24=G10*G11/(G10+G11)/(WM(4)*CP(4))
A25=G12*(1-G12/(G12+G13))/(WM(4)*CP(4))

C INITIALIZE
DISPLAY "INPUT--↑TB,TW1,TEX,TW2,TINS,TW3(R)"
ACCEPT TB,TW1,TEX,TW2,TINS,TW3
TM=TB
TIME=0.
DT=0.
TPRINT=0.
CK=0.
TSTART =FINTIM
TSTOP=FINTIM
CKOLD=0.
CKD=0.
DEADT=.0008333
C DYNAMIC
50 TE=TM-1787.
   CKOLD=CK
   IF(TE.LE.0.)CK=1
   IF(TE.GT.54.)CK=0.
   IF(CKOLD.EQ.CK) GO TO 55
   IF(CK.GT.CKOLD)TSTART=TIME+DEADT
   IF (CK.LT.CKOLD)GO TO 54
   TSTOP=FINTIM
   GO TO 55
54 TSTOP=TIME+DEADT
   TSTART=FINTIM
55 CONTINUE
   IF(TIME.GE.TSTART)CKD=1
   IF(TIME.GE.TSTOP)CKD=0.
   QHEAT=CKD*QGEN
   TBDOT=A1*(TIN-537)-A2*(TB-537)+QHEAT-A3*TB+A4*TW1
   WRITE(6,*) CK,CKD,QHEAT,TBDOT
   TB=INTGRL(TB,TBDOT,DT)
   TW1DOT=A8*TB-A9*TW1+A10*TW2+A11*TEX
   TW1=INTGRL(TW1,TW1DOT,DT)
   TAU=A12
   F=A13*TW1+A14*TW2+A15*TAIR
   TEX=TEZ(TEX,TAU,F,DT)
   TW2DOT=A16*TEX+A17*TW1-A18*TW2+A19*TINS
   TW2=INTGRL(TW2,TW2DOT,DT)
   TINS=INTGRL(TINS,TINSDOT,DT)
   TINS=INTGRL(TINS,TINSDOT,DT)
   TW3=INTGRL(TW3,TW3DOT,DT)
   TMDOT=(TB-TM)/TAU2
   TM=INTGRL(TM,TMDOT,DT)
   DT=DELT
   IF(TIME.GE.TPRINT) GO TO 65
   GO TO 70
65 WRITE(6,*) TIME,TB,TW1,TEX,TW2,TINS,TW3,TM
   WRITE(9,250)TIME,TB,TW1,TEX,TW2,TINS,TW3
250 FORMAT(F10.4,8F10.2)
TPRINT = TPRINT + OUTDEL
70 TIME = TIME + DT
   IF (TIME .LE. FINTIM) GO TO 50
   STOP
END
FUNCTION INTEGRAL (Y, DY, DT)
   INTEGRAL = Y + DY * DT
   RETURN
END
FUNCTION TEZ(Y, TAU, F, DT)
   TAU = TAU * DT
   IF (TAU > 160.) 2, 2, 1
1  TEZ = F / TAU
   RETURN
2  TEZ = F / TAU + (Y - F / TAU) * EXP(- TAU * DT)
   RETURN
END
$CONTROL
USLIMIT,NOSOURCE
C THIS PROGRAM SIMULATES THE FBC SYSTEM.
   CHARACTER*30 TITLE
   DIMENSION H(4),TC(4),R(6),T(4),WM(5),CP(5)
   DATA H/25000.,9.,9.,4.5/
   DATA TC/9.4,9.4,.058,30/
   DATA R/,.2527,.2760,.289,.2956,.9635,.970/
   DATA T/,.023,.0065,.668,.0065/
   DATA WM/36.,.9711,.2991,.9689,.150/
   DATA CP/,.194,.12,.12,.107,.21/
C CONSTANTS
   TAU2= .017
C INPUT
   DISPLAY "INPUT--→ TITLE"
   ACCEPT TITLE
   DISPLAY "INPUT--→FINTIM,DELT,OUTDEL"
   ACCEPT FINTIM,DELT,OUTDEL
   DISPLAY "INPUT--→TIN,TAIR,TSURR"
   ACCEPT TIN,TAIR,TSURR
   DISPLAY "INPUT--→FUEL RATE"
   ACCEPT FUEL R
C PROGRAM TO CALCULATE PARAMETERS IN EQUATIONS
L=5
SIGMA=.1714*(10.**(4.))
TO=1824
SCP1N=1.88*FUEL R
SCP1OUT=1.97*FUEL R
HF=8000.*FUEL R
E1=.65
E2=.65
WMAIR=200.
WMEX=.00312
CPEX=.24
CPAIR=.24
G1=H(1)*2*3.141*R(1)*L/(2*3.141*TC(1)*L)/ALOG((R(1)+.5*T(1))/
#R(1))+H(1)*2*3.141*R(1)*L)
G2=2*3.141*TC(1)*L/ALOG((R(1)+.5*T(1))/R(1))/(2*3.141*TC(1)*
#L)/ALOG((R(1)+.5*T(1))/R(1))+H(1)*2*3.141*R(1)*L)
G3=2*3.141*TC(1)*L/ALOG(R(2)/(R(1)+.5*T(1)))
G4=H(2)*2*3.141*R(2)*L
G5=SIGMA/(1/(2*3.141*R(2)*L*E1)+1/(2*3.141*R(3)*L)*(1/E2-1))
G6=2*3.141*TC(2)*L/ALOG((R(3)+.5*T(2))/R(3))
G7=H(3)*2*3.141*R(3)*L
G8=2*3.141*TC(2)*L/ALOG(R(4)/(R(3)+.5*T(2)))
G9=2*3.141*TC(3)*L/ALOG((R(4)+.5*T(3))/R(4))
G10=2*3.141*TC(4)*L/ALOG((R(5)+.5*T(4))/R(5))
G11=2*3.141*TC(3)*L/ALOG(R(5)/(R(4)+.5*T(3)))
G12=H(4)*2*3.141*R(6)*L
G13=2*3.141*TC(4)*L/ALOG(R(6)/(R(5)+.5*T(4)))
B1=(G4+G3+4*G5*TO**3)/(4*G5*TO**3)
B2=G4/(4*G5*TO**3)
B3=G3/(4*G5*TO**3)
B4=(G6+4*G5*TO**3+G7)/(4*G5*TO**3)
B5=G6/(4*G5*TO**3)
B6=G7/(4*G5*TO**3)
B7=(B4*B2+B6)/(B4*B1-1)
B8=(B4*B3)/(B4*B1-1)
B9=B5/(B4*B1-1)
B10=B1*B7-B2
B11=B1*B8-B3
B12=B1*B9
A1=SCPIN/(WM(1)*CP(1))
A2=SCPOUT/(WM(1)*CP(1))
QGEN=HF/(WM(1)*CP(1))
A3=H(1)*2*3.141*R(1)*L*(1-G1)/(WM(1)*CP(1))
A4=H(1)*2*3.141*R(1)*L*G2/(WM(1)*CP(1))
A5=H(2)*2*3.141*R(2)*L*(B7-1)+4*G5*(TO**3)*(B7-B10)
A6=H(2)*2*3.141*R(2)*L*B8+4*G5*(TO**3)*(B8-B11)
A7=H(2)*2*3.141*R(2)*L*B9+4*G5*(TO**3)*(B9-B12)
A8=A3/(WM(2)*CP(2))]*(WM(1)*CP(1))
A9=(WM(1)*CP(1)*A4+A6)/(WM(2)*CP(2))
A10=-A7/(WM(2)*CP(2))
A11=-A5/(WM(2)*CP(2))
A12=(H(2)*2*3.141*R(2)*L*(1-B7)+H(3)*2*3.141*R(3)*L*(1-B10)+
#WMAIR*CPAIR)/(WMEX*CPEX)
A13=(H(2)*2*3.141*R(2)*L*B8+H(3)*2*3.141*R(3)*L*B11)/(WMEX*#CPEX)
A14=(H(2)*2*3.141*R(2)*L*B9+H(3)*2*3.141*R(3)*L*B12)/(WMEX*#CPEX)
A15=WMAIR*CPAIR/(WMEX*CPEX)
A16=(H(3)*2*3.141*R(3)*L*(1-B10)+4*(TO**3)*G5*(B7-B10))/
#(WM(3)*CP(3))
A17=(H(3)*2*3.141*R(3)*L*B11*(-1)+4*(TO**3)*G5*(B8-B11))/
#(WM(3)*CP(3))
A18=H(3)*2*3.141*R(3)*L*(B12)/(WM(3)*CP(3)) + 4*(TO**3)*G5
#*(B12-B9)/(WM(3)*CP(3)) + G8/(WM(3)*CP(3))*(1-(G8/(G8+G9)))
A19=G8/(WM(3)*CP(3)) G9/(G8+G9)
A20=G8/(WM(5)*CP(5)) *(1-G8/(G8+G9))
A21=(G8+G9)/(G8+G9) + G10*(G10+G11)/(WM(5)*CP(5))
A22=(G10/(G10+G11)-1)*(-1)*G10/(WM(5)*CP(5))
A23=(G10*(1-G10)/(G10+G11)+G12*G13/(G12+G13))/(WM(4)*CP(4))
A24=G10*G11/(G10+G11)/(WM(4)*CP(4))
A25=G12*(1-G12/(G12+G13))/(WM(4)*CP(4))

C INITIALIZE
DISPLAY "INPUT--+TB,TW1,TEX,TW2,TINS,TW3\(r\)"
ACCEPT TB,TW1,TEX,TW2,TINS,TW3
TM=TB
TIME=0.
DT=0.
TPRINT=0.
TSTART =FINTIM
TSTOP=FINTIM
DEADT=.0008333
C DYNAMIC
50 TE=TM-1787
QHEAT=14000-600*TE
TBDOT=A1*(TIN-537)-A2*(TB-537)+QHEAT-A3*TB+A4*TW1
WRITE(6,*) QHEAT,TBDOT
TB=INTGRL(TB,TBDOT,DT)
TW1DOT=A8*TB-A9*TW1+A10*TW2+A11*TEX
TW1=INTGRL(TW1,TW1DOT,DT)
TAU=A12
F=A13*TW1+A14*TW2+A15*TAIR
TEX=TEZ(TEX,TAU,F,DT)
TW2DOT=A16*TEX+A17*TW1-A18*TW2+A19*TINS
TW2=INTGRL(TW2,TW2DOT,DT)
TINSDOT=A20*TW2-A21*TINS+A22*TW3
TINS=INTGRL(TINS,TINSDOT,DT)
TW3DOT=-A23*TW3+A24*TINS+A25*TSURR
TW3=INTGRL(TW3,TW3DOT,DT)
TMDOT=(TB-TM)/TAU2
TM=INTGRL(TM,TMDOT,DT)
DT=DELT
IF(TIME.GE.TPRINT) GO TO 65
GO TO 70
65 WRITE(6,*) TIME,TB,TW1,TEX,TW2,TINS,TW3,TM
WRITE(9,250)TIME,TB,TW1,TEX,TW2,TINS,TW3
250 FORMAT(F10.4,8F10.2)
TPRINT=TPRINT +OUTDEL
70 TIME= TIME +DT
IF(TIME.LE.FINTIM) GO TO 50
STOP
END
FUNCTION INTGRL (Y,DY,DT)
INTGRL=Y+DY*DT
RETURN
END
FUNCTION TEZ(Y,TAU,F,DT)
TAUD=TAU*DT
IF(TAUD=160.)2,2,1
1 TEZ=F/TAU
RETURN
2 TEZ=F/TAU+(Y-F/TAU)*EXP(-TAU*DT)
Appendix F

COMPUTER PROGRAMS FOR DATA ANALYSIS
$CONTROL USLINIT,NOSOURCE
C PROGRAM TO CALCULATE BURNER PERFORMANCE
CHARACTER*8 DATE
DIMENSION S(2),D(2),DP(2),TP(2),QA(2),ZW(8),ZWR(7),ZV(7)
DIMENSION XV(8),XMW(8),XWF(9,3),XMWF(8),A(5,7),P(5),H(7)
EQUIVALENCE (ZW(1),DYN),(ZW(2),DYAO),(ZW(3),DYAH),
#(ZW(4),DYAC2),(ZW(5),DYAA),(ZW(6),DYAS),(ZW(7),DYACL),(ZW(8),ASH)
DATA S/0.2578,0.2578/,D/1.609,3.065/,TK/459.6/,tref/60./
DATA XMW/28.016,32.0,18.016,44.011,39.944,64.066,28.011,1./
DATA XMWF/28.016,32.0,2.016,12.011,39.944,32.066,12.011,1./
DATA XWF/.00381,.4324,.0585,.4706,0.,.0006,0.,.0341,7926.,
#.01270,.0759,.0520,.7603,0.,.0157,0.,.0635,13000.,
#.0158,.4311,.0722,.4633,0.0,.0012,0.0,.0164,8072./
DATA XV/.7552,.2315,0.,.0004,.0128,0.,0.,0./
DATA A/9.470,0.,0.,-3470.,1160000.,11.515,0.,-172.,1530.,0.,
#19.86,0.,-597.,7500.,0.,16.20,0.,0.,-6530.,1410000.,4.953,0.,0.,
#0.,0.,9.46,0.,0.,-3290.,1070000.,0.,0.,0.,0.,0./
DISPLAY "PROGRAM SENGYBAL"
WRITE(6,100)
display "This program is called SENGYBAL. It is
designed to accept
# laboratory data and"
display "do an energy and mass balance. It begins with a series
# of questions which"
display "ask for fuel type,fuel rates,air flow rates and specific
# temperatures."
display ""
display "-----------------------------TO START HIT RETURN------------------"
#-----------------------------
read(5,*) start
display ""
C Orifice plate diameters are d1=1.000,d2=1.905,where l is combustion.
10 DISPLAY "INPUT ---↑DATE(Month-Day-Year), Time(Hour xx.xx)"
display ""
accept DATE,TIME
DISPLAY "INPUT(FUEL)---↑NUMBER(cobs=1,coal=2,shelled corn=3,other
#≠4)"
display ""
ACCEPT NFUEL
DISPLAY "INPUT(FUEL)---↑RATE(LBS/HR),MOISTURE(WB),DENSITY(LB/FT3)"
display ""
ACCEPT QC,XMC,DENC
DISPLAY "INPUT(AMBIENT AIR)---↑DRY BULB,DEW POINT(C),PRESS(MM HG)"
display ""
ACCEPT TAIRO,TDP0,PAIRO
DISPLAY "INPUT(COOLING AIR)---↑DP2(IN H2O),TP2(CM HG),Tin(C),Tout"
display ""
ACCEPT DP(2),TP(2),TCOOLIN,TCOOLOUT
DISPLAY "INPUT(COMBUSTION AIR) ----> DP1(MM H2O),Tin(C),Tout(C)"
display ""
ACCEPT DP1,TCOMBIN,TCOMBOUT
TP(1)=0.0
display "INPUT CO LEVEL(decimal)"
display ""
ACCEPT PCO
XV(3)=0.
XMC2=XMC/(1.-XMC)
tair=1.8*tair0+32.
TDP=1.8*TDP0+32.
pair=PAIR0/25.4
tbl=1.8*TCOMBIN+32.
tb2=1.8*TCOMBOUT+32.
tcl=1.8*TCOOLIN+32.
tc2=1.8*TCOOLOUT+32.
ECOMB=XWF(9,NFUEL)
DP(1)=DP1/25.4
AMW=0.
DO 14 I=1,5
14 AMW=AMW+XV(I)*XMW(I)
PF=.4912*PAIR
TF=TAIR+TK
TR=TF/238.55
PR=PF/546.3
X=TDP+TK
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
IF(J.EQ.2) TF=TCL+TK
HW=DP(J)
PF=PF+TP(J)*.1934
Y=1-(FM*FM*HW/PF)*.0125
GF=(AMW*(PF-PD)+XMW(3)*PD)/(10.73*TF*Z)
QA(j)=359*S(J)*D(J)*FA*FM*FC*Y*(GF*HW)**0.5
18 CONTINUE
QA(2)=1.00*QA(2)
WRITE(6,106) DATE,TIMED,TAIR,TDP,PAIR,XMC,XMC2,DENC
DYC=(1.-XMC)*QC
DYCM=XMC*QC
XMA=XV(3)/(1+XV(3))
DYA=(1-XMA)*QA(1)
DYAM=XMA*QA(1)
DYA2=(1-XMA)*QA(2)
22  ZW(I)=XV(I)*DYA+XMW(I)*XWF(I,NFUEL)*DYC/XMWF(I)
DO 23 J=3,7
23  ZW(2)=ZW(2)-(XMW(J)-XMWF(J))*XWF(J,NFUEL)*DYC/XMWF(J)
   ZW(3)=ZW(3)+DYCM
   EXC=DYAO/(XV(2)*DYA-DYAO)
   WRITE(6,108) DYC,DYCM,TAIR,DYA,DYAM,TB1,TB2,DYA2,DYAM2,TC1,TC2,EXC,
   TOTWT=0.
   TOTMOL=0.
   DO 25 I=1,7
   TOTWT=TOTWT+ZW(I)
   ZV(I)=ZW(I)/XMW(I)
   TOTMOL=TOTMOL+ZV(I)
   ZV(7)=PCO*TOTMOL/(1.-.5*PCO)
   TOTMOL=TOTMOL+.5*ZV(7)
   ZV(2)=ZV(2)+0.5*ZV(7)
   ZV(4)=ZV(4)-ZV(7)
   DMH20=ZV(3)
   DO 27 I=1,7
   ZWR(I)=ZW(I)/TOTWT
27  ZV(I)=ZV(I)/TCMOL
   VPA=DMH20*PAIR/TOTMOL
   WRITE(6,109) (ZWI),ZWR(I),ZV(I), I=1,7),ASH
   SIN=(ECOMB+.36*(TAIN-TREF))*DYC+1.0*(TB1-TREF)*DYCM+.24*(TB1-TREF)
   #*DYA+.45*(TB1-TREF)*DYAM+.24*(TC1-TREF)*DYA2+.45*(TC1-TREF)*DYAM2-
   #1059.3*(DYAH-DYAM)
   TC=TB2+TK
   TR=TREF+TK
   P(1)=TC-TR
   P(2)=(TC*TC-TR*TR)/2.
   P(3)=2.*(TC**0.5-TR**0.5)
   P(4)=LOG(TC/TR)
   P(5)=-(1./TC-1./TR)
   DO 29 I=1,7
   H(I)=0.
   DO 29 J=1,5
29  H(I)=H(I)+A(J,I)*P(J)
   EO=ECOMB=0.0
   DO 30 I=1,5
30  EO=EO+EO+H(I)*ZW(I)/XMW(I)
   TC=TC2+TK
   TR=TREF+TK
   P(1)=TC-TR
   P(2)=(TC*TC-TR*TR)/2.
   P(3)=2.*(TC**0.5-TR**0.5)
   P(4)=LOG(TC/TR)
   P(5)=-(1./TC-1./TR)
   DO 31 I=1,7
   H(I)=0.
DO 31 J=1,5
31 H(I)=H(I)+A(J,I)*P(J)
EOCOOL=0.
DO 32 I=1,5
32 EOCOOL=EOCOOL+H(I)*XV(I)*DYA2/XMW(I)
OTHER=EIN-EOCOMB-EOCOOL
WRITE(6,107) TREF,-EIN,1.0,EOCOMB,EOCOMB/EIN,EOCOOL,EOCOOL/EIN,
OTHER,OTHER/EIN
ACCEPT ZZZZZZ
GO TO 10
99 format(2x"INPUT-DATE,TEST NO."),
100 FORMAT(1X,79"-")
101 FORMAT(2X,"INPUT-COB RATE(LBS/HR),MOISTURE(WB),DENSITY(LB/FT3)"/
#2X,"COMBUSTION AIR IN,OUT(C),COOL AIR IN,OUT(C)"/
# 12X,"AMBIENT AIR CONDITIONS"/20X,"DRY BULB,F,Dew Pt.,F
# PRESSURE,IN HG"/20X,42"-/"/20X,2F10.2,F15.2//12X,"COBS","/20X,
#"MOISTURE,WB MOISTURE,DB DENSITY,LB/FT3"/20X,42"-/"/20X,2F10.4
#F15.4//)
108 FORMAT(12X,"MEASURED VARIABLES DURING TEST"/30X," FLOW,LBS/HR
#TEMPERATURES,F"/30X," DRY WATER IN OUT"/30X,32"-/"/14X
#3X,4F 8.1/20X,"(EXCESS AIR FOR COMBUSTION IS ",F5.2,")/")
109 FORMAT(12X,"PRODUCTS FROM COMBUSTION"/32X,"LBS/HR WEIGHT VOLUME"
#/40X,"RATIO RATIO"/30X,32"-/"/14X,"NITROGEN",7X,3F8.3/
#13X,3F8.3/14X,"ASH",12X,F8.3//)
105 FORMAT(4X,2F8.0,15,3F10.3,F10.3)
107 FORMAT(12X,"ENERGY BALANCE: TREF="F7.1/35X," BTU/HR RATIO"/30X
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
To use the energy balance program, SENGY, the following are required inputs from the time-average program.

The temperature of the combustion air at the inlet and outlet of the burner, the temperature of the cooling air at the inlet and outlet of the heat exchanger, the ambient dry bulb temperature, and the combustion air pressure differential.

Other inputs are the date and time of test, the fuel rate, and moisture content of the fuel, the ambient air dew point, and the barometric pressure, the cooling air manometer reading, and the CO level.

The corncob ultimate analysis found in Appendix C was modified as follows to account for the added lime. Two percent lime by weight was added to the fuel and this lime was treated as additional ash. The weight percents change as follows.

<table>
<thead>
<tr>
<th>element</th>
<th>unaltered wt%</th>
<th>altered wt%</th>
</tr>
</thead>
<tbody>
<tr>
<td>N2</td>
<td>.00389</td>
<td>.00381</td>
</tr>
<tr>
<td>O</td>
<td>.44105</td>
<td>.4324</td>
</tr>
<tr>
<td>2H2</td>
<td>.05962</td>
<td>.0585</td>
</tr>
<tr>
<td>C</td>
<td>.4800</td>
<td>.4706</td>
</tr>
<tr>
<td>S</td>
<td>.00061</td>
<td>.0006</td>
</tr>
<tr>
<td>Ash</td>
<td>.01483</td>
<td>.0341</td>
</tr>
</tbody>
</table>
To get the CO reading, the analyzer was calibrated at 2% full scale, that is, a reading of .5 = 1%.
Appendix G

SIMULATION RESULTS
The results from each on/off controlled simulation run are listed in Table Gl. The values were taken from the end of each run after steady-state had been reached. The temperature profile, TB, ... TW3, in the burner is given at the peak of a cycle, that is, as the cobs are turned off, and at the low point of a cycle, right before the cobs turn back on.

Table Gl. Simulation Data.

<table>
<thead>
<tr>
<th>run</th>
<th>cycle, hr</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>12</th>
</tr>
</thead>
<tbody>
<tr>
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<td>off/on</td>
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<td>.026</td>
<td>.023</td>
<td>.020</td>
<td>.030</td>
<td>.022</td>
<td>.024</td>
<td>.034</td>
<td>.054</td>
<td>.024</td>
<td>.021</td>
</tr>
<tr>
<td></td>
<td>on/off</td>
<td>.053</td>
<td>.046</td>
<td>.053</td>
<td>.071</td>
<td>.039</td>
<td>.070</td>
<td>.042</td>
<td>.015</td>
<td>.133</td>
<td>.055</td>
<td>.078</td>
</tr>
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<td></td>
<td>on/onf</td>
<td>.076</td>
<td>.072</td>
<td>.076</td>
<td>.091</td>
<td>.069</td>
<td>.092</td>
<td>.066</td>
<td>.049</td>
<td>.187</td>
<td>.079</td>
<td>.099</td>
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<td>Peak, R</td>
<td></td>
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<td></td>
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<td></td>
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<td></td>
<td></td>
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</tr>
<tr>
<td>TB</td>
<td></td>
<td>1892</td>
<td>1910</td>
<td>1893</td>
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<td>1876</td>
<td>1911</td>
<td>1870</td>
<td>2002</td>
<td>1894</td>
<td>1877</td>
</tr>
<tr>
<td>TW1</td>
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<td>1899</td>
<td>1881</td>
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<td>1903</td>
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<td>1898</td>
<td>1858</td>
<td>1990</td>
<td>1882</td>
<td>1864</td>
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<tr>
<td>TEX</td>
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<td>1503</td>
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<td>1845</td>
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<td>1294</td>
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<td>1709</td>
<td>1714</td>
<td>1694</td>
<td>1819</td>
<td>1711</td>
<td>1846</td>
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<td>TINS</td>
<td></td>
<td>958</td>
<td>828</td>
<td>974</td>
<td>942</td>
<td>979</td>
<td>959</td>
<td>960</td>
<td>967</td>
<td>979</td>
<td>957</td>
<td>983</td>
</tr>
<tr>
<td>TW3</td>
<td></td>
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<td>571</td>
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<td>575</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>TB</td>
<td></td>
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<td>1702</td>
<td>1714</td>
<td>1694</td>
<td>1715</td>
<td>1715</td>
<td>1701</td>
<td>1740</td>
<td>1614</td>
<td>1700</td>
<td>1708</td>
</tr>
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<td>TW1</td>
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<td>1704</td>
<td>1701</td>
<td>1714</td>
<td>1693</td>
<td>1716</td>
<td>1714</td>
<td>1701</td>
<td>1739</td>
<td>1614</td>
<td>1700</td>
<td>1707</td>
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<td>TEX</td>
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<td>1312</td>
<td>1560</td>
<td>1424</td>
<td>1419</td>
<td>1438</td>
<td>1361</td>
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<td>1698</td>
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<td>1653</td>
<td>1694</td>
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<td>TINS</td>
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<td>974</td>
<td>942</td>
<td>979</td>
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<td>576</td>
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</tr>
</tbody>
</table>
Appendix H

EXPERIMENTAL RESULTS
The following are the results of each experimental test run.

Group 4 Conditions: Cooling Air On
 Fluidizing Air On

Testing:
 Conduction Losses
 1,2,3, Cooling Air Induced Convection Losses
 18,19 Fluidizing Air Induced Losses

Test 1 Conditions: air/fuel ratio = 1.13
 fuel rate = 15.1 lb/hr
 cooling air = 1.5" water

Table H1. Test 1 Data - Dynamic Response.

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/On (min)</th>
<th>Off/On (min)</th>
<th>On/Off (min)</th>
<th>Undershoot (°C)</th>
<th>Overshoot (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.23</td>
<td>.53</td>
<td>--</td>
<td>17.3</td>
<td>.1</td>
</tr>
<tr>
<td>2</td>
<td>4.0</td>
<td>.46</td>
<td>3.54</td>
<td>17.4</td>
<td>.4</td>
</tr>
<tr>
<td>3</td>
<td>3.77</td>
<td>.53</td>
<td>3.24</td>
<td>13.3</td>
<td>4.1</td>
</tr>
<tr>
<td>4</td>
<td>4.23</td>
<td>.46</td>
<td>3.77</td>
<td>17.9</td>
<td>.7</td>
</tr>
<tr>
<td>5</td>
<td>4.7</td>
<td>.54</td>
<td>4.16</td>
<td>16.9</td>
<td>1.1</td>
</tr>
<tr>
<td>6</td>
<td>4.04</td>
<td>.47</td>
<td>3.57</td>
<td>14.7</td>
<td>.4</td>
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<tr>
<td>7</td>
<td>4.88</td>
<td>.5</td>
<td>4.38</td>
<td>16.4</td>
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<td>8</td>
<td>3.92</td>
<td>.46</td>
<td>3.46</td>
<td>9.6</td>
<td>.2</td>
</tr>
</tbody>
</table>

| AVG.  | 4.22        | .49          | 3.73         | 15.4            | 1.0            |
| ST.DEV.| .39         | .04          | .40          | 2.8             | 1.3            |

- continued
### TABLE H1 (continued)

#### ENERGY BALANCE

**DATE** 10-19-84  **TIME** 10:00

<table>
<thead>
<tr>
<th>AMBIENT AIR CONDITIONS</th>
<th>PRESSURE, IN. HG</th>
</tr>
</thead>
<tbody>
<tr>
<td>DRY BULB, F</td>
<td>78.85</td>
</tr>
<tr>
<td>DEW PT., F</td>
<td>60.80</td>
</tr>
<tr>
<td></td>
<td>28.86</td>
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</tbody>
</table>

<table>
<thead>
<tr>
<th>COBS</th>
<th>MOISTURE, WB</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT3</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.0900</td>
<td>0.0989</td>
<td>0.0000</td>
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#### MEASURED VARIABLES DURING TEST

<table>
<thead>
<tr>
<th>FUEL</th>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
<th>DRY</th>
<th>WATER</th>
<th>IN</th>
<th>OUT</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.1</td>
<td>1.5</td>
<td>78.9</td>
<td></td>
<td></td>
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<tr>
<td>95.3</td>
<td>1.1</td>
<td>107.8</td>
<td>713.4</td>
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<tr>
<td>279.7</td>
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<td>117.0</td>
<td>1014.4</td>
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<td></td>
</tr>
</tbody>
</table>

**EXCESS AIR FOR COMBUSTION 15%  .13**

#### PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th></th>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>NITROGEN</td>
<td>71.797</td>
<td>.640</td>
<td>.653</td>
</tr>
<tr>
<td>OXYGEN</td>
<td>2.622</td>
<td>.023</td>
<td>.026</td>
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<tr>
<td>WATER</td>
<td>10.504</td>
<td>.093</td>
<td>.150</td>
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<tr>
<td>CO2</td>
<td>26.087</td>
<td>.232</td>
<td>.143</td>
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<td>ARGON</td>
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<td>.008</td>
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<td>SO2</td>
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<td>.000</td>
</tr>
<tr>
<td>CO</td>
<td>.000</td>
<td>.000</td>
<td>.009</td>
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<td>ASH</td>
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#### ENERGY BALANCE: TREF = 60.0

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<tr>
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<th>RATIO</th>
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<tr>
<td>FUEL</td>
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<tr>
<td>COMBUSTION AIR</td>
<td>19501.4</td>
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<td>COOLING AIR</td>
<td>68294.3</td>
<td>.594</td>
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<tr>
<td>OTHER</td>
<td>27242.2</td>
<td>.237</td>
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</table>
Test 2 Conditions: air/fuel ratio = 1.18  
fuel rate = 14.2 lb/hr  
cooling air = 1.5 " water

Table H2. Test 2 Data - Dynamic Response.

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Off/On (min)</th>
<th>On/Off (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (C)</th>
<th>Overshoot (C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>.55</td>
<td>4.54</td>
<td>5.04</td>
<td>5.09</td>
<td>15.7</td>
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<tr>
<td>2</td>
<td>.50</td>
<td>5.50</td>
<td>6.00</td>
<td>6.00</td>
<td>18.5</td>
<td>2.2</td>
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<tr>
<td>3</td>
<td>.50</td>
<td>7.66</td>
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<td>8.16</td>
<td>16.7</td>
<td>2.0</td>
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<tr>
<td>4</td>
<td>.50</td>
<td>4.92</td>
<td>5.34</td>
<td>5.42</td>
<td>12.9</td>
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<td>4.66</td>
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<td>6.75</td>
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<tr>
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<td></td>
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<td>15.8</td>
<td>.2</td>
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<tr>
<td>AVG.</td>
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<td>3.84</td>
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- continued
TABLE H2 (continued)

ENERGY BALANCE

DATE 10-31-04  TIME 10:32

AMBIENT AIR CONDITIONS

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN. HG</th>
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</thead>
<tbody>
<tr>
<td>74.79</td>
<td>50.00</td>
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</table>

COBS

<table>
<thead>
<tr>
<th>MOISTURE,WB</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT3</th>
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</thead>
<tbody>
<tr>
<td>.0877</td>
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</table>

MEASUREMENT VARIABLES DURING TEST

<table>
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<tr>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DRY WATER IN/OUT</td>
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<tr>
<td>FUEL</td>
<td>14.2 1.4 74.3</td>
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<tr>
<td>COMBUSTION AIR</td>
<td>101.2 .8 104.9</td>
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<tr>
<td>COOLING AIR</td>
<td>285.9 2.2 116.1</td>
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<td>(EXCESS AIR FOR COMBUSTION IS .28)</td>
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PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
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</thead>
<tbody>
<tr>
<td>Nitrogen</td>
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<tr>
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ENERGY BALANCE: TREF = 60.0

<table>
<thead>
<tr>
<th>BTU/HR</th>
<th>RATIO</th>
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<tbody>
<tr>
<td>Fuel</td>
<td>-108620.1</td>
</tr>
<tr>
<td>Combustion Air</td>
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</tr>
<tr>
<td>Cooling Air</td>
<td>69139.0</td>
</tr>
<tr>
<td>Other</td>
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</tbody>
</table>
FIGURE H2. PLOT OF TEMPERATURES INSIDE THE FBC

Date 10/31/84  Time 10:50-11:00  TEST 2
Control Temp  Cooling Exit  Burner Exit  Burner Inlet  Cooling Inlet
T1           T2             T3           T4            T5

TEMPERATURE, DEG C

TIME, HOURS
Test 3 Conditions: air/fuel ratio = 1.18
fuel rate = 16.8 lb/hr
cooling air = 1.5 " water

Table H3. Test 3 Data - Dynamic Response.

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (°C)</th>
<th>Overshoot (°C)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>.4</td>
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<td>2.50</td>
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<td>2.33</td>
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<td>.50</td>
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<td>12.7</td>
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</tbody>
</table>

AVG. 1.94 .50 2.43 2.45 13.2 2.3
ST. DEV. .16 .05 .17 .18 1.7 1.2

- continued
TABLE H3 (continued)

ENERGY BALANCE

DATE 11-2-84.

AMBIENT AIR CONDITIONS

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN.HG</th>
</tr>
</thead>
<tbody>
<tr>
<td>74.12</td>
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</table>

COBS

<table>
<thead>
<tr>
<th>MOISTURE, W.B</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT3</th>
</tr>
</thead>
<tbody>
<tr>
<td>.0877</td>
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MEASURED VARIABLES DURING TEST

<table>
<thead>
<tr>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DRY WATER IN OUT</td>
</tr>
<tr>
<td>FUEL</td>
<td>16.8 1.6 74.1</td>
</tr>
<tr>
<td>COMBUSTION AIR</td>
<td>109.9 0.5 101.3 730.4</td>
</tr>
<tr>
<td>COOLING AIR</td>
<td>287.1 1.2 115.9 1008.8</td>
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</table>

(EXCESS AIR FOR COMBUSTION IS .10)

PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
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</thead>
<tbody>
<tr>
<td>NITROGEN</td>
<td>93.078 .648</td>
<td>.674</td>
</tr>
<tr>
<td>OXYGEN</td>
<td>3.855 .030</td>
<td>.032</td>
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<tr>
<td>WATER</td>
<td>10.840 .005</td>
<td>.137</td>
</tr>
<tr>
<td>CO2</td>
<td>26.990 .226</td>
<td>.140</td>
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<tr>
<td>ARGON</td>
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<td>.008</td>
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<tr>
<td>SO2</td>
<td>.020 .000</td>
<td>.000</td>
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<tr>
<td>CO</td>
<td>.000 .000</td>
<td>.009</td>
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<tr>
<td>ASH</td>
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ENERGY BALANCE: TREF= 60.0

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<tr>
<th>BTU/HR</th>
<th>RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>FULL</td>
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<tr>
<td>COMBUSTION AIR</td>
<td>22972.3</td>
</tr>
<tr>
<td>COOLING AIR</td>
<td>60681.5</td>
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<tr>
<td>OTHER</td>
<td>35518.6</td>
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</tbody>
</table>
Test 18  Conditions:  
air/fuel ratio  = 1.21  
fuel rate  = 16.9 lb/hr  
cooling air  = .72 " water

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (C)</th>
<th>Overshoot (C)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>2.00</td>
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<td>3.9</td>
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<td>.57</td>
<td>2.00</td>
<td>1.86</td>
<td>10.7</td>
<td>2.7</td>
</tr>
<tr>
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<td>1.29</td>
<td>.57</td>
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<td>2.14</td>
<td>8.7</td>
<td>4.0</td>
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<tr>
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<td>.57</td>
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<td>2.00</td>
<td>12.7</td>
<td>3.2</td>
</tr>
<tr>
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<td>15.1</td>
<td>1.9</td>
</tr>
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<td></td>
<td></td>
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<td></td>
<td>7.5</td>
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<td>1.94</td>
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<td>3.4</td>
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<td>ST.DEV.</td>
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<td>.03</td>
<td>.15</td>
<td>.12</td>
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- continued
TABLE H4 (continued)

ENERGY BALANCE

DATE 11-20-64  TIME 17:25

AMBIENT AIR CONDITIONS

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN HG</th>
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<tbody>
<tr>
<td>72.95</td>
<td>33.00</td>
<td>28.90</td>
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</tbody>
</table>

COGS

<table>
<thead>
<tr>
<th>MOISTURE, WD</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT3</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0839</td>
<td>0.0915</td>
<td>0.0000</td>
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</tbody>
</table>

MEASURED VARIABLES DURING TEST

<table>
<thead>
<tr>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DRY WATER IN OUT</td>
</tr>
</tbody>
</table>

| FUEL | 16.9 | 1.6 | 72.9 |
| COBUSTION AIR | 113.5 | 0.5 | 99.5 | 855.2 |
| COOLING AIR | 172.0 | 0.8 | 98.5 | 1122.3 |

(EXCESS AIR FOR COMBUSTION 10.21)

PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>NITROGEN</td>
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<tr>
<td>OXYGEN</td>
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<tr>
<td>WATER</td>
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<td>.003</td>
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<td>CO2</td>
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<td>.222</td>
</tr>
<tr>
<td>ARGON</td>
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<td>.011</td>
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<td>SO2</td>
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<td>.000</td>
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<tr>
<td>CO</td>
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<tr>
<td>ASH</td>
<td>.578</td>
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ENERGY BALANCE: TREF = 60.0

<table>
<thead>
<tr>
<th>BTU/HR</th>
<th>RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>FUEL</td>
<td>-126363.8</td>
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<tr>
<td>COBUSTION AIR</td>
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</tr>
<tr>
<td>COOLING AIR</td>
<td>52652.5</td>
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<tr>
<td>OTHER</td>
<td>45737.6</td>
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</table>
FIGURE H4. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/28/84          Time 17.00-17.30          TEST 18

Control Temp          Cooling Exit          Burner Exit
T1                   T2                       T3

Burner Inlet          Cooling Inlet
T4                   T5

TEMPERATURE, DEG C

750
700
650
600
550
500
450
400
350
300
250
200
150
100
50

0.0  0.1  0.2  0.3  0.4  0.5
TIME, HOURS
Test 19  Conditions:  
- air/fuel ratio = 1.21
- fuel rate = 16.9 lb/hr
- cooling air = .72 " water

Table H5. Test 19 Data - Dynamic Response.

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (C)</th>
<th>Overshoot (C)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
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<td>1.69</td>
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<td>10.5</td>
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<tr>
<td>4</td>
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<td>1.68</td>
<td>8.3</td>
<td>3.9</td>
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<tr>
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<td>.75</td>
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<td>1.75</td>
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<td>1.75</td>
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<td>1.69</td>
<td>7.5</td>
<td>6.3</td>
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<td>.19</td>
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- continued
### Table H5 (continued)

**Energy Balance**

**Date:** 11-20-04  
**Time:** 22:24

**Ambient Air Conditions**

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN Hg</th>
</tr>
</thead>
<tbody>
<tr>
<td>71.06</td>
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<td>29.94</td>
</tr>
</tbody>
</table>

**COBS**

<table>
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<th>MOISTURE, WB</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT³</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0838</td>
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</table>

**Measured Variables During Test**

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<tr>
<th>Flow, LBS/HR</th>
<th>Temperatures, F</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>DRY</td>
</tr>
<tr>
<td>FUEL</td>
<td>16.7</td>
</tr>
<tr>
<td>Combustion Air</td>
<td>113.4</td>
</tr>
<tr>
<td>Cooling Air</td>
<td>96.0</td>
</tr>
</tbody>
</table>

*(Excess Air for Combustion is 0.20)*

**Products from Combustion**

<table>
<thead>
<tr>
<th>LBS/HR</th>
<th>Weight Ratio</th>
<th>Volume Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nitrogen</td>
<td>85.704</td>
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<tr>
<td>Oxygen</td>
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<td>0.034</td>
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<tr>
<td>Water</td>
<td>10.818</td>
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</tr>
<tr>
<td>CO₂</td>
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</tr>
<tr>
<td>Argon</td>
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<td>0.011</td>
</tr>
<tr>
<td>SO₂</td>
<td>0.020</td>
<td>0.000</td>
</tr>
<tr>
<td>CO</td>
<td>0.000</td>
<td>0.000</td>
</tr>
<tr>
<td>Ash</td>
<td>578</td>
<td>0.000</td>
</tr>
</tbody>
</table>

**Energy Balance: TREF = 60.0**

<table>
<thead>
<tr>
<th></th>
<th>BTU/HR</th>
<th>Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
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<tr>
<td>Other</td>
<td>63,325.7</td>
<td>0.547</td>
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</table>
FIGURE H5. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/28/84  Time 22.00-22.40  TEST 19

<table>
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<th>Control Temp</th>
<th>Cooling Exit</th>
<th>Burner Exit</th>
<th>Burner Inlet</th>
<th>Cooling Inlet</th>
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<tr>
<td>Ti</td>
<td>T2</td>
<td>T3</td>
<td>T4</td>
<td>T5</td>
</tr>
</tbody>
</table>

TEMPERATURE, DEG C

TIME, HOURS
Group 2 Conditions: Cooling Air Off  
Fluidizing Air On  

Testing:  
6,8,14 Conduction Losses  
Fluidizing Air Induced Losses  

Test 6 Conditions:  
air/fuel ratio $= 1.44$  
fuel rate $= 13.9$ lb/hr  

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (C)</th>
<th>Overshoot (C)</th>
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<td>.63</td>
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<td>2.00</td>
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<td>15.9</td>
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<td>1.17</td>
<td>.63</td>
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<td>1.80</td>
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<tr>
<td>5</td>
<td>1.08</td>
<td>.72</td>
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<td>1.80</td>
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<td>.77</td>
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<td>.86</td>
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<td>1.89</td>
<td>8.1</td>
<td>3.7</td>
</tr>
</tbody>
</table>

| AVG.  | 1.10        | .72         | 1.83        | 1.81         | 7.1            | 4.3           |
| ST.DEV.| .20         | .10         | .15         | .13          | 2.1            | 3.6           |

- continued
TABLE H6 (continued)

ENERGY BALANCE

DATE 11-23-84
TIME 14:53

AMBIENT AIR CONDITIONS
DRY BULB, F DEW PT., F PRESSURE, IN.HG
72.01 28.40 29.33

CCBS
MOISTURE, WB MOISTURE, DB DENSITY, LB/FT³
-118 1259 .0000

MEASURED VARIABLES DURING TEST
FLOW, LBS/HR TEMPERATURES, F
FLOW, LBS/HR TEMPERATURES, F

FUEL 13.9 1.8 72.0
COMBUSTION AIR 111.7 .4 99.5 967.7
COOLING AIR .0 .0 105.8 796.8
(EXCESS AIR FOR COMBUSTION IS .44)

PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th></th>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
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</thead>
<tbody>
<tr>
<td>NITROGEN</td>
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<td>.888</td>
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<td>OXYGEN</td>
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<td>.119</td>
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<td>ARGON</td>
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<td>CO</td>
<td>.000</td>
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ENERGY BALANCE: TREF = 60.0
BTU/HR RATIO

<table>
<thead>
<tr>
<th></th>
<th>BTU/HR</th>
<th>RATIO</th>
</tr>
</thead>
<tbody>
<tr>
<td>FUEL</td>
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<tr>
<td>COMBUSTION AIR</td>
<td>30887.6</td>
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<td>COOLING AIR</td>
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<td>.000</td>
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<tr>
<td>OTHER</td>
<td>71252.1</td>
<td>.698</td>
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</table>
Test 8  Conditions:  air/fuel ratio = 1.32
fuel rate = 15.5 lb/hr

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (C)</th>
<th>Overshoot (C)</th>
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</thead>
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<td>1.43</td>
<td>1.62</td>
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<tr>
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<tr>
<td>8</td>
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<td>1.64</td>
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<td>8.2</td>
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<td>.86</td>
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<td>.64</td>
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<td>1.43</td>
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<td>5.6</td>
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<tr>
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<td>.71</td>
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<td>1.65</td>
<td>1.57</td>
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<td>ST.DEV.</td>
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<td>.30</td>
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</table>

- continued
TABLE H7 (continued)

ENERGY BALANCE

DATE 11-24-04  TIME 18.71

AMBIENT AIR CONDITIONS

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN. HG</th>
</tr>
</thead>
<tbody>
<tr>
<td>72.01</td>
<td>33.00</td>
<td>29.21</td>
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</table>

COBS

<table>
<thead>
<tr>
<th>MOISTURE, WB</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT³</th>
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</thead>
<tbody>
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<td>.1118</td>
<td>.1257</td>
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</table>

MEASURED VARIABLES DURING TEST

<table>
<thead>
<tr>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
</tr>
</thead>
<tbody>
<tr>
<td>DRY WATER</td>
<td>IN</td>
</tr>
<tr>
<td></td>
<td>OUT</td>
</tr>
</tbody>
</table>

| FUEL | 15.5 | 1.9 | 72.0 |
| COMBUSTION AIR | 112.9 | .5 | 99.1 | 989.6 |
| COOLING AIR | 0.0 | 0.0 | 112.9 | 826.8 |

(EXCESS AIR FOR COMBUSTION IS .32)

PRODUCTS FROM COMBUSTION

<table>
<thead>
<tr>
<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
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</thead>
<tbody>
<tr>
<td>NITROGEN</td>
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<tr>
<td>OXYGEN</td>
<td>6.266</td>
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<td>WATER</td>
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<td>.081</td>
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<td>ASH</td>
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ENERGY BALANCE: TREF = 60.0

<table>
<thead>
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<th>BTU/HR</th>
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<tbody>
<tr>
<td>FUEL</td>
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<tr>
<td>COMBUSTION AIR</td>
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<td>.0</td>
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<tr>
<td>OTHER</td>
<td>80467.7</td>
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</table>
Test 14  Conditions:  
air/fuel ratio = 1.03  
fuel rate = 19.6 lb/hr  

Table H8. Test 14 Data - Dynamic Response.  

<table>
<thead>
<tr>
<th>Cycle</th>
<th>On/Off (min)</th>
<th>Off/On (min)</th>
<th>On/On (min)</th>
<th>Off/Off (min)</th>
<th>Undershoot (°C)</th>
<th>Overshoot (°C)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
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<td>9.0</td>
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<td>1.43</td>
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<tr>
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<td>.72</td>
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<td>1.57</td>
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<td>5.3</td>
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<td>.67</td>
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<td>1.43</td>
<td>8.0</td>
<td>4.3</td>
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<td>.76</td>
<td>.72</td>
<td>1.48</td>
<td>1.72</td>
<td>10.1</td>
<td>6.8</td>
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<tr>
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<td>1.00</td>
<td>.71</td>
<td>1.71</td>
<td>1.52</td>
<td>8.4</td>
<td>4.2</td>
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<tr>
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<td>.76</td>
<td>1.57</td>
<td>1.48</td>
<td>5.3</td>
<td>6.6</td>
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<td>.85</td>
<td>1.57</td>
<td>1.52</td>
<td>8.0</td>
<td>10.3</td>
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<tr>
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<td>.76</td>
<td>1.43</td>
<td>1.76</td>
<td>11.2</td>
<td>8.7</td>
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<td>1.86</td>
<td>1.61</td>
<td>9.5</td>
<td>6.3</td>
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<td>.72</td>
<td>1.60</td>
<td>1.57</td>
<td>8.7</td>
<td>6.4</td>
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<td>ST.DEV.</td>
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<td>.08</td>
<td>.13</td>
<td>.11</td>
<td>1.4</td>
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- continued
TABLE H8 (continued)

ENERGY BALANCE

**DATE 11-23-84**

**TIME 10:42**

**AMBIENT AIR CONDITIONS**

<table>
<thead>
<tr>
<th>DRY BULB, F</th>
<th>DEW PT., F</th>
<th>PRESSURE, IN.HG</th>
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</thead>
<tbody>
<tr>
<td>74.40</td>
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**COBS**

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<th>MOISTURE, WD</th>
<th>MOISTURE, DB</th>
<th>DENSITY, LB/FT³</th>
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<td>0.0038</td>
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**MEASURED VARIABLES DURING TEST**

<table>
<thead>
<tr>
<th>FLOW, LBS/HR</th>
<th>TEMPERATURES, F</th>
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</thead>
<tbody>
<tr>
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</table>

(Excess air for combustion is 0.03)

**PRODUCTS FROM COMBUSTION**

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<th>LBS/HR</th>
<th>WEIGHT RATIO</th>
<th>VOLUME RATIO</th>
</tr>
</thead>
<tbody>
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<td>NITROGEN</td>
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**ENERGY BALANCE: TREF= 60.0**

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<th>RATIO</th>
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**Conditions:** Cooling Air On
Fluidizing Air Off

**Testing:**

<table>
<thead>
<tr>
<th>Test</th>
<th>Conduction Losses</th>
<th>Cooling Air Induced Losses</th>
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</thead>
<tbody>
<tr>
<td>9, 10, 11</td>
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<td></td>
</tr>
<tr>
<td>16, 17</td>
<td>Conditions: Cooling Air Flow Rate = 197 lb/hr</td>
<td></td>
</tr>
<tr>
<td>20, 22</td>
<td>Conditions: Cooling Air Flow Rate = 87 lb/hr</td>
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</table>
FIGURE H7. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/26/84      Time 22.95-23.10      Test 10

TEMPERATURE, DEG C

Loc 5
T6

Loc 4
T5

Loc 3
T4

Loc 2
T3

Loc 1
T2

Cool Air Exit
T1

TIME, HOURS

0.00   0.03   0.06   0.09   0.12   0.15
FIGURE H8. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/26/84  Time 23.85-24.00  TEST 11

TEMPERATURE, DEG C

Burner Exit
T6
Loc 5
T5
Loc 4
T4
Loc 3
T3
Loc 2
T2
Loc 1
T1
FIGURE H9. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/26/84    Time 23.05-24.00    Test 11
FIGURE H10. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/28/84      Time 12.85-13.05      TEST 16

TEMPERATURE, DEG C

Burner Exit
T6

Loc 5
T5

Loc 4
T4

Loc 3
T3

Loc 2
T2

Loc 1
T1

TIME, HOURS
FIGURE H11. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/28/84          Time 12:35-13:05          Test 16

TEMPERATURE, DEG C

Loc 5
T6

Loc 4
T5

Loc 3
T4

Loc 2
T3

Loc 1
T2

Cool Air Exit
T1

TIME, HOURS

0.00 0.05 0.1 0.15 0.2
FIGURE H12. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/28/84  Time 14:15-14:30  TEST 17

TEMPERATURE, DEG C

Burner Exit T6
Loc 5 T5
Loc 4 T4
Loc 3 T3
Loc 2 T2
Loc 1 T1

TIME, HOURS

0.00  0.05  0.1  0.15  0.2
FIGURE H13. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/28/84   Time 14.15-14.30   Test 17
FIGURE H14. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/28/84  Time 22.75-23.20  Test 20

Temperature, Deg C

Burner Exit
T6

Loc 5
T5

Loc 4
T4

Loc 3
T3

Loc 2
T2

Loc 1
T1

Time, Hours
FIGURE H.15. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/28/84
Test 20

Temperature, deg C

Time, Hours

Loc 5
Loc 4
Loc 3
Loc 2
Loc 1
Cool Air Exit
T1
T2
T3
T4
T5
T6
FIGURE H16. PLOT OF TEMPERATURES INSIDE THE FBC

Date 11/29/84        Time 3.40-3.75    Test 22
FIGURE H17. PLOT OF TEMPERATURES INSIDE THE EXCHANGER

Date 11/29/84       Time 3.40-3.75       TEST 22

[Graph showing temperature vs. time for different locations labeled Loc 1 to Loc 5, Cool Air Exit T1, and T2 to T5.]