Design and Operational Characteristics of a

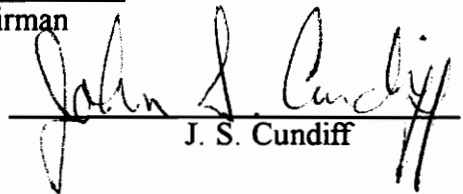**Gasification-Combustion Process:**

Flammability Model

by

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Thesis submitted to the Faculty of the

Virginia Polytechnic Institute and State University

in partial fulfillment of the requirement for the degree of

Masters of Science

in

Agricultural Engineering

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April, 1995

Blacksburg, Virginia

Key words: Gasification, Combustion, Producer gas, Flammability limit, Wood

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GASIFICATION-COMBUSTION PROCESS:
FLAMMABILITY MODEL

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(ABSTRACT)

The research reported here explored the flammable range of gasification product "producer gas" in a combustion chamber to ensure complete combustion. Rising fuel prices has led to increased research in renewable energy sources. Biomass is a renewable resource whose use does not result in a net increase of CO_2 in the atmosphere. Wood was selected as the biomass for this research. Applications for wood as a fuel source includes crop drying, space heating, and power generation. Flammability limit and chemical equilibrium theory were used to model the flammable range of the gasification product in a combustion chamber.

The model predicted an adiabatic flammable zone within an equivalence ratio of 0.56 to 1.67 for oak with 20 percent moisture content (w.b.), and a maximum adiabatic flame temperature of 2025°C for dry oak. Chemical equilibrium theory was used to predict gasification-combustion product concentration.

Based on the analysis of the data, the following conclusions were made: (1) Flammability of gas-air mixture is largely determined by the amount of heat loss prior to combustion, (2) At equivalence ratios greater than 1.25, CO appears in the combustion products, (3) Adiabatic Flame Temperatures are largely influenced by moisture and excess air, (4) Combustion temperature is a critical parameter that influences composition distribution of the gasification-combustion product. (Product compositions are important to the

designer, for both energy and environmental impact), and (5) Maximum benefit for a gasifier-combustor system could be obtained if heat loss, excess air, moisture content, mixing effectiveness, and residence time are optimized.

Acknowledgments

I would like to express my appreciation to my committee members, Dr. L. Marsh, for serving as the committee chairman and for her support and encouragement during the past two years and Dr. J. S. Cundiff for serving on my committee and for his support and technical assistance during the preliminary experiments. To Dr. Dennis Jaasma, much gratitude is due, for serving on my committee, and for his knowledge and suggestions that I have valued throughout this research.

A number of people are acknowledged for their work and assistance during the preliminary experiments, Leon Alley, Clyde Adkins, and the Solid Fuel Combustion laboratory personnel for their support.

Also many thanks to my friends for their friendship and encouragement. Most of all I thank God for blessing me with life, health and strength throughout the last two years.

Finally, I wish to thank my family for their support and encouragement throughout my life, especially my wife Charity and our two children for their love, patience, and understanding. It is to them I dedicate this work.

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1.0   INTRODUCTION

Throughout history, wood fuel has played a fundamental role as an energy supply. Until 1850, wood was the primary residential and industrial energy source. However, the discovery and development of fossil fuel has significantly reduced the importance of wood fuel.

Today, wood fuel does not play a major role in developed nations. Nevertheless, in oil-importing developing nations wood accounts for approximately 38 percent of consumed energy, and 90 percent of the total energy required in most African nations. For example, Ethiopia and Nepal are almost totally dependent on biofuels while Kenya is 75 percent dependent on biomass for energy (Tillman, 1991).

World-wide production of dry biomass is estimated at 170 billion tons per year, and current consumption is estimated at 1.3 billion tons per year (International Energy Workshop, 1991). There is need to pursue the development of biomass energy systems in both developed and developing nations. Researchers have linked global warming to an increase of CO_2 in the atmosphere. Therefore, developed nations need to utilize more biomass fuel to reduce the effect of CO_2 release from combustion of fossil fuels. Developing nations need to utilize more biomass because of the potential of stimulating economic growth.

Applications of wood as a fuel source range from small to large scale. Examples for small-scale applications include on-farm operations that require relatively low temperatures (such as drying and space heating). Large-scale applications include district heating and power generation. Increases in the cost of fuel for operations such as crop drying, tobacco curing, poultry brooding, and peanut drying have stimulated increased research on the use of wood as an alternative fuel source. With rising fuel prices, the

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gasification-combustion process is becoming economically feasible. It may prove appropriate for developing countries that are totally dependent on fossil fuel import, such as Kenya.

To be economical, biomass must be used locally; consequently, biomass fuel is most competitive in rural areas. This is not a major limitation for the use of biomass, since there are many applications in rural areas, specifically crop drying and space heating. Lack of clean-burning, inexpensive combustors is a limiting factor in many potential applications. If not completely burned, biomass can foul heat exchangers and cause air pollution.

There are two major wood conversion processes: direct combustion and gasification. In direct combustion the fuel is burned at high temperatures to achieve clean combustion products. Direct combustion is a suitable method for a process where power output control is not necessary. However, many processes, crop drying for example, require power output control.

Gasification is the heating of fuels without oxygen to drive off gaseous fuel (producer gas), which is then burned to produce energy. Gasification processes were popular at the turn of the century, but became uneconomical with the availability of inexpensive fossil fuels. Gasification can provide gas for heat and power for domestic or commercial use, and is the basis for synthesis of liquid fuels and ammonia.

A third conversion process combines gasification and combustion, in a two-stage process. Initially, wood is gasified in an oxygen-starved environment (primary chamber). Volatiles are driven off and char is oxidized to produce gas. This gas exits the primary chamber into an adjacent secondary chamber where secondary air is injected to facilitate complete

combustion. Depending on the concentration of the combustibles, the mixture may, or may not, propagate a flame.

The research reported here is intended to provide solutions to various technical problems experienced by the author while investigating the fuel-feedrate of an updraft gasifier. The gasifier-combustor unit is installed in the Bioprocessing laboratory of the Biological Systems Engineering Department. The unit heats water at atmospheric pressure to provide space heating for the laboratory. Equipment is in place to simulate energy demand on the unit. Ultimately, the equipment will be used for research on the combustion of wood to meet a typical time-varying heat energy demand for crop drying.

Moscahlaidis et al. (1989) burned chunkwood in the gasifier-combustor unit (hereafter referred to as a "chunkwood combustor"). The unit had a 500 MJ/hr design capacity, expected secondary combustion zone temperatures of $1200^{\circ}C$-$1800^{\circ}C$, and an expected efficiency of 80-90 percent at design temperatures. Moscahlaidis found that for one-hour tests, using a 45 kg charge of mill ends, the combustion efficiency ranged from 24.1 to 52 percent. For a continuous test, efficiencies ranged from 42.2 to 52.3 percent. The maximum secondary chamber temperature obtained was $1060^{\circ}C$, with a kerosene flame ($560^{\circ}C$) used as an ignition source. In all cases, he assumed complete combustion of the fuel charge at the end of each gasification-combustion event.

1.1   Summary of the Preliminary Investigation

Several trials were conducted using the existing unit burning chunkwood pieces. A permanent kerosene flame (500 to 630°C) was used as an ignition source in the secondary combustion chamber (SCC). With gasification in progress and the kerosene burner operating, SCC temperatures ranged 800 to 900°C, with heat exchanger flue gas temperatures as high as 43°C. The preliminary investigation has shown that the flue gas

temperature (taken at 12 cm above the base of heat exchanger) was too low to be considered as a heat source to preheat the secondary air. The flue gas CO reading saturated at 1.6 percent five minutes after each test. The SCC temperature, the exhaust gas analysis, and visual examination of the combustor behavior (smoky) implied a major operational problem. Three conclusions were made from this result. (1) It was evident that the secondary air was cooling the producer gas in the SCC, (2) The secondary air inlets were designed to inject air at an angle of 45° towards the exit of the SCC; however, some of this air forced its way back to the primary chamber, causing the primary chamber to leak; and (3) The rest of the secondary air left the SCC without complete combustion as indicated by a low flue gas CO_2 reading. Low CO_2 readings implied that the O_2 level was high in the flue gas--an unfavorable condition for the combustion process (Jaasma, 1994). Appendix A1 contains the details of the procedure and results of the preliminary experiments.

Design changes for the existing chunkwood combustor that could be considered are: (1) Investigate the effect of secondary air on combustion zone temperature by monitoring the secondary air entry temperature and mixing effectiveness, (2) Alternatives for preheating the secondary air besides using the flue gas; and (3) Investigate the effect of changes in SCC geometry. These design modifications would affect one of the most important parameters that control the combustion process, that is, temperature; however, these changes may demand additional equipment and cost, defeating the original idea of 'simplicity' at the least cost.

To understand the cited technical problems, it was found necessary to model the gasification-combustion process. It was evident that the flammability limit theory and chemical equilibrium model could provide important technical information for the design and operation of a gasifier combustor system.

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1.2   Goal for this Study

In developing countries, like Kenya, emphasis is now being placed on making maximum use of indigenous energy resources in order to minimize the outflow of foreign exchange and to ensure energy security. Kenya has abundant waste biomass, 2.7 million tons per year (World Bank Report, 1987). Refuse remaining after logging operations is a widely available material that could be used to produce chunkwood. It is my hope that this study will provide technical information necessary for designing an economical and environmentally safe chunkwood combustor for use in rural Kenya.

Chapter 2

REVIEW OF LITERATURE

2.0 Characteristics of Biomass

Biomass is any material that was recently a living plant or animal. Compared to coal, the characteristics of biomass that are relevant to thermal processing, especially where the objective is maximum gasification, are: (1) it contains water and oxygen necessary for converting carbon to gaseous fuel; (2) it has a high content of volatiles; (3) generally it contains more hydrogen than carbon; and (4) it has a lower sulfur content (White and Plaskett, 1981).

Biomass has a lower energy content than fossil fuels, and often has a high moisture content. A high moisture content consumes energy during combustion and also causes storage problems. Biomass has a low bulk density, a major factor in design of handling equipment and storage facilities. It is rarely homogenous and free flowing, thus automatic feeding of the combustor is a challenge. Biomass, however, is a renewable resource and its use does not produce a net increase of CO_2 in the atmosphere. Carbon stored in plants is released when the plants are burned, but the resulting CO_2 is captured by the next generation of plants.

The behavior of wood as a fuel is influenced by anatomical, physical and chemical characteristics. Anatomical characteristics, which include wood fibers and pathways of moisture movement, determine the wood-fuel-moisture relationships. Physical

characteristics include moisture content, specific gravity, void volume, and thermal properties. The relationship between moisture content, specific gravity, void volume, and thermal properties is critical because moisture content affects the heat produced by wood. Chemical characteristics include proximate analysis, ultimate analysis, and the heating value. Proximate analysis defines the fuels as four components--volatile matter, fixed carbon, moisture and ash. It is used to measure the volatile matter of a given fuel. Ultimate analysis lists fuel composition by chemical elements (Tillman et al. 1981).

2.1   Conversion of wood to energy

There are three methods of converting wood to energy: pyrolysis, gasification, and direct combustion. Direct combustion is the most widely used of these methods.

2.2.1   Direct wood combustion

Direct combustion is the simplest approach of converting biomass to energy. The combustion process is a complex series of time and temperature dependent chemical reactions. Physically, biomass does not burn directly, rather a two-stage process takes place. In the first stage, the fuel is heated to form and drive off combustible gas, a process known as pyrolysis. The second stage oxidizes the gas produced by pyrolysis and energy is released. The heat released during this stage fuels the pyrolysis process and the cycle continues. There are four basic chemical processes involved in wood combustion: (1) heating and drying, (2) solid-phase pyrolysis, (3) gasification and (4) char oxidation (Tillman, 1991).

2.2.1.1 Heating and drying

Upon exposure to heat, biomass chunkwood pieces are heated via an endothermic process. The limiting factor in drying and heating is the initial wood moisture content. Energy needed to heat chunkwood to pyrolysis temperature increases with moisture content. During this stage, heat is driven into the chunkwood particle from the gasifier environment. The moisture in the pores of the particle is vaporized from the surface to the center and steam escapes from the particle. Heating of the chunkwood core is achieved when all the moisture is vaporized. Also, thermal conductivity increases with an increase in moisture content and this increases the heat transfer rate from the chunkwood surface to the center. The energy required in this phase is a function of the chunkwood moisture content, ash content, dimensions, heat capacity, diffusivity, and thermal conductivity. Consequently, the heating and drying phase is limited by the moisture level. This phase continues until temperature (350°C) is favorable for pyrolysis to take place (Tillman, 1981, 1991).

2.2.1.2 Pyrolysis

Pyrolysis is the breaking down of a material by heat in the absence of oxygen. Pyrolysis produces multiple products, some of which are liquid and can be transported remotely. Pyrolysis takes place when the fuel is heated to about 350°C, producing volatiles, tar, and charcoal. The volatiles remain in gas phase (CO, CO_2, H_2, H_2O, CH_4). Tar is a heavy hydrocarbon and can be treated as product of pyrolysis by itself. Char is a carbon-rich solid with very little hydrogen and oxygen. Pyrolysis depends on the specific combustor, combustor temperature, fuel size, and the way fuel is distributed inside the combustor (pile

or continuous flow). Pyrolysis is an endothermic process, which depends on the thermodynamic properties of the fuel.

2.2.2 Gasification

Gasification, a modification of pyrolysis, is the partial combustion of wood, in an oxygen or air- starved (less than stoichiometric) combustor, to produce a mixture of combustible gases (Fung, 1982). The result of gasification is partial combustion of biomass fuel, to either low-or medium-grade fuel depending on whether air or oxygen is used to activate combustion. Unlike pyrolysis, gasification reactions take place at higher temperatures. The higher temperatures restrict the production of tars and oil, which can condense and cause sticking and other operating problems.

The product of gasification, commonly known as 'producer gas', is a mixture of gases (CO, CO_2, N_2, H_2, CH_4, H_2O) and higher hydrocarbons. Unlike pyrolysis, gasification yields a single product (gas), which is difficult to transport. It is best used for a close-coupled system, where the producer gas is burned as it is generated. The gas quality produced by a gasifier depends on the nature of fuel used and the gasifier operating conditions (Tillman, 1978).

Leppa and Saarni (1982) provided a simplified explanation of the gasification process. They explained that the process begins with addition of air to the glowing layers of coals in a combustor. This initiates a process characterized by the following reactions:

$$C + O_2 => CO_2 \qquad +32.8 \text{ MJ/kg of carbon} \qquad [2.1]$$

$$CO_2 + C => 2CO \qquad -12.9 \text{ MJ/kg of carbon} \qquad [2.2]$$

$$H_2O + C => CO + H_2 \qquad -9.6 \text{ MJ/ kg of carbon} \qquad [2.3]$$

$$3C \ + O_2 \ + \ H_2O \ => \ 3CO + H_2 \qquad + 3.4 \ MJ/ \ kg \ of \ carbon. \qquad [2.4]$$

$$3CO \ + \ H_2 \ + \ 2O_2 \ => \ 3CO_2 \ + H_2O \qquad + 29.4 \ MJ/kg \ of \ carbon. \qquad [2.5]$$
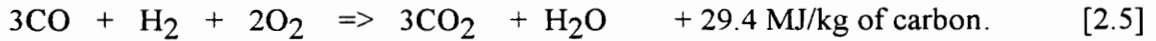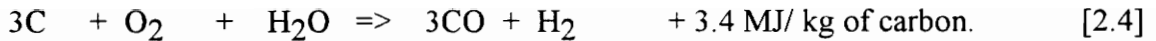
In reaction 2.1 above, carbon is completely oxidized to CO_2 and heat is released, simultaneously, CO_2 is reduced (reaction 2.2), and some heat is absorbed as CO_2 flows through the charred wood. Steam from the drying wood flows upward, reacting with the glowing coals to absorb heat (reaction 2.3). Gasification occurs in the glowing coals and heat is released (reaction 2.4). If the products of reaction 2.4 are mixed with air in an adjacent combustion chamber, they burn easily producing heat (reaction 2.5).

There is a wide range given in the literature for the gasification temperature. Desrosiers (1981) reported a temperature range of 726-1126°C. Reed and Das (1988) reported a gasification temperature range of 700-1000°C. They found that the compositions of importance at these temperatures are: C, CO, CO_2, CH_4, H_2, and H_2O.

There are variations in gasification configurations and systems. Different combustor designs, different air flow systems (induction or forced), steam injection, and pressurized gasifiers are some of the options that have been investigated. These parameters can be controlled to produce a desired product and to optimize the process.

Several configurations for two-stage combustors for converting biomass into thermal energy for crop processing have been investigated (Payne, 1980; Richey and Kutz 1981; McCoy, et al. 1981; Morey and Timsen 1981, and others). If properly controlled, two-stage combustors can provide both clean gases and output modulation.

There are three types of fixed-bed gasifiers: updraft, downdraft, and crossdraft. The choice of a gasifier type depends on the use of the producer gas. Updraft gasifiers yield a fuel gas appropriate for combustion systems, for example boilers (Reed, 1981). Figure 1 shows the basic configuration of a two-stage updraft gasifier. Air is introduced at the bed of the gasfier, striking the hot charcoal and causing gasification. The hot gases then rise up the bed, heating the wood, and causing carbonization. As the gases rise further up, they dry the wood before exiting the combustor at the top. Wood is fed at the top of the updraft gasifier. The combustible gases contain a high proportion of volatile matter, and they are best burned before condensation can occur. Downdraft gasifiers are preferred for conversion of biomass to methanol. In a down-draft gasifier, air and wood are both introduced at the top and move downward. In a crossdraft gasifier, air is introduced perpendicular to the direction of wood flow.

Payne (1980) investigated gasification-combustion process for drying grain. A two-stage updraft gasifier was used with controlled combustion of the producer gas. The products of combustion were mixed with ambient air and passed directly through a grain dryer. Experimental results indicated that gasification-combustion of corncobs was a feasible alternative for drying corn.
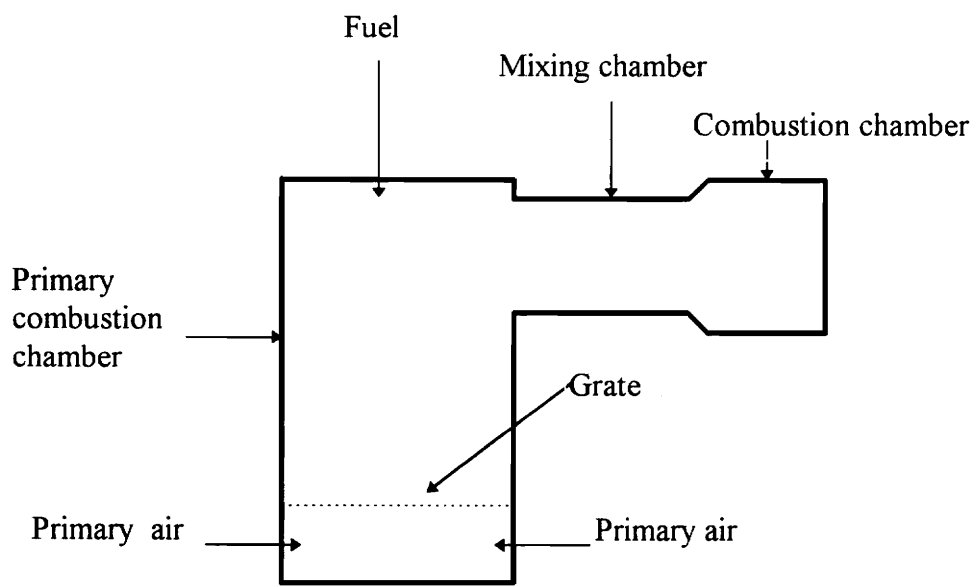
Figure 1.   Basic configuration for a two-stage gasifier

2.3   Chemical equilibrium

Chemical equilibrium exists when there are no changes in the chemical composition of a system. At equilibrium the reactants interact to form the products; simultaneously, the products interact to form the original reactants. The rates of forward reaction and backward reactions are the same. Kuo (1986) stated that for a chemical equilibrium to occur the sum of chemical potential of the products must be equal to the sum of chemical potential of the reactants, that is, mass is conserved. The criteria for a chemical process to be at an equilibrium depends on the condition at which certain thermodynamic properties (property) are kept: for example, temperature and pressure could be kept constant. The state for chemical equilibrium could be determined by several methods such as: minimization of either the Gibbs free energy or the Helmholtz free energy, or the maximization of the entropy of the universe.

For an adiabatic system, chemical equilibrium is reached when the entropy of the reacting system reaches a maximum. For a specified temperature and pressure, chemical reactions and heat transfer will proceed in the direction of decreasing Gibbs function. The reaction attains chemical equilibrium when Gibbs function attains a minimum value.   The composition of a chemical equilibrium mixture depends upon temperature and pressure (Boles, 1989).

Reynolds (1983) developed a computer program 'STANJAN' that applies the chemical equilibrium theory to solve for the final equilibrium product composition and the adiabatic flame temperature of a combustion system. STANJAN is an interactive program for chemical equilibrium analysis by the method of element potentials. It considers a large number of possible reaction products, under the constraint of conservation of atomic species at an assigned temperature and pressure or total enthalpy, and selects the mix of products that minimizes the free energy of the system phases. Each mixture is considered

as an ideal gas or ideal solution. The program uses thermochemical data for chemical species from the JANNAF tables (Stull and Prophet, 1971).

2.4 Adiabatic Flame Temperature

Adiabatic flame temperature is the maximum temperature reached in a combustion process taking place adiabatically with no work or changes in kinetic or potential energy. The products are assumed to be at a chemical equilibrium. For such a process, all the heat evolved during combustion is used up internally to raise the temperature of the combustion products. Adiabatic flame temperature depends on the pressure, initial temperature, and composition of the reactants.

Components involved in a combustion process have different chemical composition. For example, wood is composed of carbon, hydrogen, and oxygen. The enthalpy of a substance at a specified state due to its chemical components is called enthalpy of formation. It is the energy released or absorbed during the formation of that compound from its stable elements.

Kuo (1986) provided a detailed methodology for calculation of the adiabatic flame temperature. The method is based on knowledge of enthalpy of formation and the changes in enthalpies of reactants and products of a combustion process. Adiabatic theory is an important engineering tool that can be used to predict real combustion conditions. Adiabatic flame temperatures are governed by the first law of thermodynamics:

$$H_P = H_R \qquad\qquad [2.6]$$

$$H_R = \sum_{j=R} n_j h_{fj} \qquad\qquad [2.7]$$

$$H_P = \sum_{i=p} n_i h_{fi} \qquad\qquad [2.8]$$

where,

H_R = changes in the enthalpy of combustion of the reactants (kJ/kg),

H_p = changes in the enthalpy of combustion of the products (kJ/kg),

n_j = the moles of each of the reactants component (kmol),

n_i = the moles of each of the products component (kmol),

h_{fj} = heat of formation of each of the reactants component (kJ/kmol), and

h_{fi} = heat of formation of each of the products component (kJ/kmol).

Most actual engineering processes are non-adiabatic and the maximum temperature is less than adiabatic flame temperature. This is due to incomplete combustion, heat loss to the surroundings, and disassociation of some of the products at high temperatures. These events tend to cool the combustion gases. To model a real combustion process, adiabatic flame temperature can be converted to the actual temperature by adjusting for heat loss. Researchers have provided a first approximation factor of 0.8 implemented by the following equation:

$$T_{actual} = T_{AF} \times CF \qquad\qquad [2.9]$$

where,

T_{AF} = adiabatic flame temperature, and

CF = correction factor.

In refractory-lined units correction factors vary from 0.9 to 0.95 depending on the type and thickness of the refractory material (Tillman, 1991).

Proper temperature control in a combustion system allows manipulation of combustion mechanisms, control of some pollutants (No_x), and optimization of efficiency and performance. Temperature can be controlled by manipulating the following variables: (1) air-fuel ratio (by adding more or less combustion air), (2) oxygen enrichment (moles O_2 per mole N_2), (3) preheating the oxidant above ambient conditions, (4) calorific value of the fuel, HHV or LHV (kJ/kmol), and (5) moisture content of the combustion atmosphere governed by either fuel moisture content or by water or steam injection into the system. Each of the above parameters can be varied independently. Flame temperature calculations can show the influence of these variables (Tillman, 1991).

2.5   Air supply

The combustion of solid fuel in air with less or excess air can be described by the general hydrocarbon combustion equation:

$$C_a H_b O_c + kH_2O + m(O_2 + 3.76N_2) \Rightarrow$$
$$aCO_2 + \left[\frac{b}{2} + k\right]H_2O + 3.76 \times \left[a + \frac{b}{4} - \frac{c}{2}\right]N_2 + (m - n)O_2 \qquad [2.10]$$
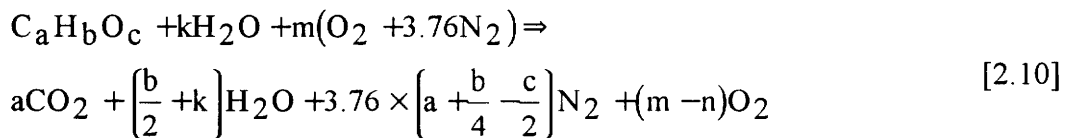
where,

$$m = (1 + \varepsilon) \times \left[a + \frac{b}{4} - \frac{c}{2}\right] \qquad [2.11]$$

a, b, c = the fuel composition coefficient for elemental carbon, hydrogen, and oxygen,

k = the number of moles of water in the fuel,

n = the number of moles of oxygen needed for stoichiometric combustion, and

ε = the amount of excess air used (decimal).

Air-fuel ratio for complete combustion of biomass fuel is then calculated based on the empirical formula (for example, $C_{4.12} H_{6.62} O_{2.3}$ for red oak), and the mass balance of a general hydrocarbon combustion equation 3.0. This is the stoichiometric air requirement,

S°, defined as the mass of dry air required to burn a unit mass of dry fuel. Stoichiometric air requirement varies with the chemical composition of biomass (Tillman, 1991).

In practice, more air is needed to drive the combustion reactions to completion. Air supplied above the stoichiometric requirement is called excess air, and is expressed as a percentage of stoichiometric air. In gas combustion, excess air can be varied to maintain a desired combustion temperature. Payne and Chandra (1985) reported that for a practical gasification-combustion system, a minimum of 10 to 30 percent excess air is required for complete combustion of the gasification products. Typical values for excess air range from 10 to 150 percent.

The biomass moisture content controls the excess-air range a combustor can operate within. The excess air requirement is inversely proportional to the moisture content of the biomass--the greater the biomass moisture content, the more excess air required to facilitate complete combustion. Dry biomass burned in a well-insulated combustor would require less excess air. Payne and Chandra (1985) concluded that, for design purposes, 0-200 percent of excess air is reasonable for a two-stage, steady-state combustor.

The gasification processes, unlike combustion, require less than stoichiometric air. Payne and Chandra (1985) defined the equivalence ratio for gasification ϕ_g as the fraction of stoichiometric air that is used to gasify biomass. Their results indicated that, for an updraft gasification of wood chips, ϕ_g is linearly related to the fraction of the fuel gross heat of combustion converted to sensible and latent energy in the gasifier. They concluded that gasifier insulation and fuel moisture content are the parameters that control the gasification equivalence ratio.

2.6 Gas production

The product of gasification is normally a hot, dirty, anaerobic gas. In a batch-fed gasification process, gas production fluctuates with the rate of gasification. The quantity and composition of the producer gas varies widely. English (1974) reported that the amount of volatiles shortly after a new load can reach 200 percent of the design conditions. Payne and Chandra (1985) reported that for a two-stage steady-state gasifier-combustor, gas production from the gasification chamber is a function of biomass gasification rate, moisture content, ash content, and stoichiometric air requirement. Gas production as a function of the aforementioned parameters was thus defined as:

$$M_g = M_b \times (1 - X_m) \times \left[\frac{1}{1 - X_m} + S^\circ \times \Phi_g - X_a^o \right] \qquad [2.12]$$

where,

M_g = gas production rate (kg/hr)

M_b = biomass gasification rate (kg wet biomass/ hr),

X_m = biomass moisture content (wet basis, decimal),

S° = stoichiometric air requirement (kg air/kg of dry biomass),

Φ_g = equivalent ratio for gasification (fraction of S° used in gasification), and

X_a^o = mass fraction of ash in biomass (kg ash/ kg dry biomass).

The gasification product could go through an adiabatic mixing with air in the SCC. The flammability of the producer gas depends on the extent of mixing in the SCC. For clean burning it is important that the producer gas and air mix well and for the mixture to remain in the SCC long enough for a complete burn. The mixing and duration determine whether or not the gas will support a pre-mixed flame. Pre-mixed flames are those that have the

fuel (producer gas) and the oxidizer (air) mixed before combustion. Alternatively, there could be a possibility where the gases could support a diffusion flame. In diffusion flames the fuel and the oxidizer are initially separated, for example a lighted candle flame (Kuo, 1987).

2.7   Flammability limit

Flammability is defined as the ability of a mixture of combustible gases to support (propagate) a flame freely over a limited range of compositions (Hilado, 1973). Whether a gas-air mixture is flammable or not depends on the changes of the fuel-air ratio (equivalence ratio), Φ, inside the combustion chamber. For a liquid fuel-air combustion process, flow of both fuel and air can easily be regulated; therefore, Φ can be held constant. In a batch-fed solid fuel-air process (for example gasification-combustion), Φ is a variable in the secondary combustion chamber; it depends on the stage of gasification and gasification equivalence, Φ_g. However, for both liquid and solid fuels there exists a lower and an upper mixture limit beyond which a gas or vapor-air mixture will not be flammable. These mixture ratios fall at the lean (more fuel and less oxidizer) and rich (less fuel and more oxidizer) ends of the combustible concentration-range (Glassman, 1987).

The lean flammability limit (LFL) is the lowest concentration of the combustible gas that will give a flammable mixture. The upper flammability limit (UFL) is the highest concentration of the combustible gas that will give a flammable mixture. Concentrations below the lower limit do not have enough combustibles to maintain a flame. Concentrations above the upper limit do not contain enough oxygen to maintain a flame. Glassman (1987) stated that flammability limit of a combustible mixture of gases or vapor is determined by the competition between the heat generated by the reaction, and the

external heat loss by the flame to the surrounding. Consequently, the higher the heat loss in a combustor, the less flammable a mixture of gases would be.

2.7.1 Effects pressure and temperature on flammability limit

The flammability limits discussed above are based on standard atmospheric temperature and pressure. Changes in pressure and temperature have pronounced effects on both limits.

Kuo (1986) explained that flammability limits of premixed laminar flames depend on temperature and pressure. He stated that as pressure is increased UFL increases, but the LFL is not affected appreciably (especially for hydrocarbon-air mixtures). Raising the temperature broadens the flammability range, and experiments have shown that flammability limits vary linearly with temperature.

Zabatakis (1964) obtained flammability limits of several vapor and gas-air mixtures. Using the composition of a flammable and a non-flammable limiting mixture, the limits were calculated as:

$$L_{T,P} = \left(C_{g,n} + C_{1,f}\right) \div 2 \qquad [2.13]$$

$$U_{T,P} = \left(C_{g,f} + C_{1,n}\right) \div 2 \qquad [2.14]$$

where,

$L_{T,P}$ = lower flammability limit, at specific temperature and pressure,

$U_{T,P}$ = upper flammability limit, at specific temperature and pressure,

$C_{g,n}$ = greatest concentration of fuel in oxidant that is non-flammable,

$C_{1,n}$ = least concentration of fuel in oxidant that is non-flammable,

$C_{1,f}$ = least concentration of fuel in oxidant that is flammable, and

$C_{g,f}$ = greatest concentration of fuel in oxidant that is flammable.

Zabatakis (1964) observed that flame propagation through a mixture depends on temperature, pressure, and mixture composition. The propagation rate is minimum at the limits and maximum at or near the stoichiometric mixture.

A technique for calculating the flammability limits as a function of temperature was reported by Bodurtha (1980). He stated that LFLs are decreased 8 percent by a temperature rise of 100°C, and UFLs are increased 8 percent by each temperature rise of 100°C. This relationship is described by:

$$L_t = L_{25°C} - \left(0.8 \times L_{25°C} \times 10^{-3}\right) \times (t - 25) \qquad [2.15]$$

$$U_t = U_{25°C} - \left(0.8 \times U_{25°C} \times 10^{-3}\right) \times (t - 25) \qquad [2.16]$$

where,

L_t = lower flammability limit at temperature t,

U_t = upper flammability limits at temperature t,

$L_{25°C}$ = lower flammability limits at room temperature,

$U_{25°C}$ = upper flammability limits at room temperature, and

t = the specified temperature.

2.7.2 Lower flammability limit and heat of combustion

The heat of combustion is the amount of heat produced by the complete combustion of a unit quantity of a material. Bodurtha (1980) reported that for many hydrocarbon-air mixtures the lower flammability limit (volume percent) of a fuel multiplied by its net heat of combustion (kJ/mole) was approximately a constant (4350 kJ/mole). This observation implies that the flame temperature (T_f) in a combustion system is a constant for all hydrocarbons. Hilado (1973) reported a flame temperature of 1500 K (1227°C); below this temperature, a flame will become extinct for hydrocarbon fuels.

The complete oxidation of CO to CO_2 in actual hydrocarbon combustion systems does not take place if the temperatures are below 1100 K. Thus, the oxidation temperature of CO is 1100 K (826°C) (Glassman, 1987). Therefore, if a flame propagates at 1227°C, ideally all CO should be oxidized. However, poor fuel-air mixing, lack of enough combustion air, and incorrect residence time in the SCC result in the formation of cold spots. The CO in these cold spots is not oxidized, and hence the presence of CO in the combustion products.

2.8 Fuel-Air equivalence ratio

The equivalence ratio, Φ, is defined as the mass ratio of the fuel to oxidant divided by the stoichiometric fuel-to-air ratio, $S°$. Mathematically, Φ is expressed as:

$$\Phi = \frac{\dfrac{M_f}{M_o}}{\left[\dfrac{M_f}{M_o}\right]_s} \qquad [2.17]$$

where,

M_f = mass of the fuel (kg of dry biomass),

M_o = mass of oxidant (kg of air), and

s = mass of fuel to mass of oxidant ratio for a stoichiometric air mixture.

When Φ is less than one, there is excess air and the mixture is said to be fuel lean. When Φ is greater than one, there is excess fuel and the mixture is said to be fuel rich. Under this latter circumstance, combustion can not be complete.

2.9   Producer gas composition

The dirty anaerobic gas resulting from gasification process is mainly composed of N_2, CO_2, O_2, CO, CH_4, and H_2. Table 1 shows gas composition reported by Tillman (1978) and Rajvanshi (1986).

Table 1.   **Gasification product composition**

Constituent	Percentage	
Reference	[1]	[2]
Nitrogen (N_2)	50	55-60
Carbon dioxide (CO_2)	6	1-2
Hydrogen (H_2)	10	5-10
Methane (CH_4)	1	1-2
Carbon monoxide (CO)	30	28-31
Oxygen (O_2)	-	-

[1] Tillman (1978)
[2] Rajvanshi (1986)

The composition of the producer gas reflects the composition of fuel and air that has reacted in the gasifier.

The composition and temperature of biomass flames are two important considerations when the exhaust products are intended for use as an energy source. To be successful as an energy source for applications such as crop drying, it is necessary to understand the characteristics of biomass flames to: (1) optimize system efficiency and performance, (2) to control airborne emissions of the exhaust products, and (3) to facilitate the design of efficient, low-cost gasifier-combustors.

Thimsen and Morey (1981) performed an exhaust analysis of a two-stage downdraft biomass- combustion system. They used an equilibrium model to predict composition for fuel lean, rich, and near-stoichiometric mixtures of corn cobs and air. They used conservation of mass and the first and second laws of thermodynamics to analyze the equilibrium properties of corncob flames. The researchers used a downdraft gasifier to verify their model. They observed that the model was reasonably good for equivalence ratios of one, but underestimated CO, O_2 and hydrocarbons for equivalence ratios greater than one. For equivalence ratios less than one, the model was accurate, predicting little or no CO or hydrocarbons in the combustion products.

Desrosiers (1981) presented equilibrium composition calculations of gasification of wood in an updraft gasifier. He indicated that gasification occurs at an equivalence ratio of 5-2.5. He reported that gasification products have their highest chemical energy within the equivalence ratio of 5 to 2.5, and reported an adiabatic flame temperature of 2300 K (2026.85 °C) for dry wood at 1 atm.

To the author's knowledge no work was found linking the flammability limit theory and the chemical equilibrium model. These theories can be used to provide operational and design characteristics for gasifier-combustor systems.

Chapter 3

OBJECTIVE

The main focus of this study was to investigate the effect of the flammability range on the chemical equilibrium characteristics and their relations to the operational and design features of a gasification-combustion process. The specific objectives are:

1. to use gasification equivalence Φ_g and characterize the flame temperature as a function of fuel-air equivalence ratio, Φ in the secondary combustion chamber;

2. to use flammability limit theory and chemical equilibrium theory to predict the amount of secondary air required to ensure gases in the SCC are always above the lower flammable limit;

3. to study the effect of heat loss on the flammable limit of the producer gas-air mixture; and

4. to conveniently determine flame temperature of wood fuel at any equivalence ratio.

Chapter 4

MODEL DEVELOPMENT

4.0   Concept

The essential parameters for the evaluation of combustion systems are the equilibrium products temperature and composition. As explained earlier, if all the heat liberated in a chemical reaction is used internally to raise the temperature of the products, then this temperature is called adiabatic flame temperature.

The average thermodynamic properties of the reactant mixture (wood and air) were calculated from knowledge of various thermodynamic properties, for example, temperature, pressure, and volume of each of the reactant mixture components. The reactants were allowed to react and attain an equilibrium state. The laws of thermodynamics were then used to calculate the composition and other properties (temperature, internal energy, enthalpy, and entropy) of the final mixture.

The heat of formation and the lean flammability theory require that the final temperature, T_{final}, be greater than or equal to 1200°C for the gas mixture to propagate a flame. Therefore, this relation was used to check the flammability of the mixture in the SCC. Secondary air parameters (mass flow and temperature), and the combustor heat loss are the only variables that could be varied to maintain T_{final} greater than 1200°C. To maintain 'simplicity' at the least cost for the existing gasifier-combustor, the option for preheating

the secondary air was not feasible. Therefore, the secondary air was considered to be at ambient temperature.

The model developed here provides the two essential parameters: temperature and composition. The model also provides other final equilibrium mixture properties as outlined above. The gasification process begins with the reactants (wood and air), which react to form producer gas as the end product. As discussed earlier, gasification-combustion of wood is a complex series of time-and temperature-dependent chemical reactions. Heat transfer takes place by conduction, convection, and radiation. The chemical reactions take place in solid, liquid, and gas phase, and mass transfer by diffusion and convection. A chemical equilibrium code STANJAN was used to model this complex combustion process. The following assumptions are made:

1. The reactants (fuel and air) are both at 25 °C.
2. The molar composition of air is 79 percent N_2 and 21 percent O_2.
3. Empirical formula for red oak is $C_{4.12}H_{6.62}O_{2.73}$.
4. The product species are CO_2, CO, O_2, N_2, H, H_2O, and CH_4.

4.1   Equilibrium calculations

STANJAN, the equilibrium code, requires the atomic composition of the wood species, the air-fuel ratio, and the total enthalpy of the reactants.

4.1.1 Stoichiometry

Table 2 shows the ultimate analysis of red oak, the wood species that was used for the model.

The stoichiometry of wood combustion is described by equation [3.0] and [3.1], based on the assumptions listed above. The gasification-combustion of wet wood with excess air was thus described as:

$$C_aH_bO_c + kH_2O + m(O_2 + 3.76N_2) \Rightarrow$$
$$aCO_2 + \left(\frac{b}{2} + k\right)H_2O + 3.76 \times \left(a + \frac{b}{4} - \frac{c}{2}\right)N_2 + (m - n)O_2 \quad [4.1]$$
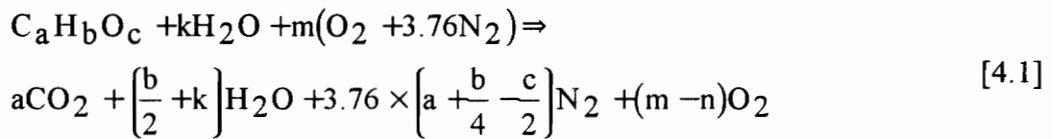
where,

$$m = (1 + \varepsilon) \times \left(a + \frac{b}{4} - \frac{c}{2}\right) \quad [4.2]$$

In the above equations a, b, and c are the fuel composition coefficients for elemental carbon, hydrogen, and oxygen, k is the number of moles of water, n is the number of moles of oxygen needed for stoichiometric combustion, and ε is the amount of excess air used. The above stoichiometry was used to calculate air-fuel ratio for complete combustion of oak, and it was found to be 6.10 kg of air per kg of dry fuel.

Table 2.   Ultimate analysis for red oak

Carbon (C)	Hydrogen (H_2)	Oxygen (O_2)	Ash
49.49	6.62	43.74	0.15

4.1.2 Enthalpy of formation of the reactants

The enthalpy of formation of red oak wood was calculated based on equation [2.6], with the knowledge of the higher heating value of oak (20 MJ/kg). The heating value of a compound is equal to the absolute value of its enthalpy of combustion. Consequently, the enthalpy of formation was calculated from the following relations.

$$h_c = H_P - H_R \qquad [4.3]$$

$$H_P = \sum_P n_p h_f^o \qquad [4.4]$$

$$H_R = \sum_R n_R h_f^o \qquad [4.5]$$

where,

h_c = enthalpy of combustion of wood (J/kg of fuel),

H_P = enthalpy of the products (J/kg),

H_R = enthalpy of the reactants (J/kg).

n_p and n_R = the mole fraction of each of the product and reactant components respectively (mol), and

h_f^o = enthalpy of formation of each of the reactant and product component (J/kmol).

The heat of formation for elements in their natural standard state, for example O_2 and N_2, is zero. The heat of formation of wood was calculated and found to be -5726.65 kJ/kg .

4.1.3   Enthalpy of the reactants mixture

The laws of thermodynamics were used to calculate the enthalpy of the reactant mixture. Since the reactants were at a standard state (25°C), the enthalpy of formation of wood (h_f^o) was equal to the enthalpy of wood (h_{wood}) at 25°C. The enthalpy of elements O_2 and N_2 were zero. Hence, the enthalpy of the reactant mixture was equal to the enthalpy of wood (J/kg of wood) described as:

$$H_R = \sum_R n_R h_f^o \qquad\qquad [4.6]$$

where,

H_R = enthalpy of reactant mixture (J/kg of wood),

n_R = moles of each reactant component (mole), and

h_f^o = enthalpy of formation of each reactant component (J/kmol).

A mass balance of equation [4.1] yielded the total mass of the reactants.   The total enthalpy per kg of mixture was then calculated as:

$$H = H_R \times \frac{M_{wood}}{M_R} \qquad\qquad [4.7]$$

where,

H = total enthalpy of the reactant mixture (J/kg of reactant mixture),

H_R = enthalpy of the reactant mixture (J/kg of wood),

M_R = total mass of the reactant mixture (kg),

M_{wood} = mass of the wood (kg).

As mentioned previously, the total enthalpy of the reactant mixture is equal to the total enthalpy of the final product mixture after the adiabatic steady-flow gasification-combustion process. This parameter was necessary for calculation of adiabatic flame temperature.

4.2 Air flow

A mass balance for equation [4.1] was used to calculate primary and secondary air. Gasification equivalence (ϕ_g) of 0.4, 0.3 and 0.2 were used to yield the amount of primary and secondary air for each case. The relationship was expressed by:

$$M_p = M_b(1-X_m)S°\Phi_g \qquad\qquad [4.8]$$

$$M_s = M_b(1-X_m)S°\left(1+\varepsilon-\Phi_g\right) \qquad\qquad [4.9]$$

where,

M_p = mass of the primary air (kg/min.),

M_b = mass of fuel burned (kg of wet fuel/min.),

M_s = mass of the secondary air (kg/min.),

X_m = fuel moisture content (decimal),

$S°$ = stoichiometric air (kg of air/ kg of dry fuel), and

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ϕ_g = fraction of stoichiometric air used for gasification (decimal).

The program inputs were: (1) atom distribution of reactants, C, H, O, and N (which depend on the equivalence ratio), and (2) total enthalpy of the reactants. The program then minimized the free energy of the system under constrained pressure and total enthalpy.

4.3   Equivalence ratio

The producer gas enters the SCC as a rich product. If the gas is properly mixed with secondary air, it should sustain a flame inside the SCC even if an ignition source is removed.

The amount of secondary air, used for combustion, was the fraction of stoichiometric air that was not injected in the primary chamber, that is, $(1-\phi_g)S°$. Furthermore, as stated earlier, practical combustion systems require excess air for complete combustion. The excess air was expressed as a fraction of the stoichiometric air, that is, $(\varepsilon)S°$. Equation [4.9] was used to calculate the total secondary air required.

The final equivalence ratio, ϕ (after combustion) was calculated from equation[2.10], after adjusting for the secondary air. The final equivalence ratio was calculated for gasification equivalence, ϕ_g of 0.4 and excess air of 0.1 to 2.5 percent of stoichiometry.

4.4   Atom balance

Referring to Figure 2, a mass balance was performed on the control volume, taking into account the mass flow crossing the boundary at points 1, 2, and 3. Atom balances of carbon, hydrogen and oxygen were calculated based on the stoichiometric equation [4.1]. Because of the excess air added in the SCC, atom distribution for the combustion process was different from that of gasification. The atom distributions were found to be:

C: a,

H: b + 2k,

O: c + k + 2m, and

N: 2*3.76m

Where a, b, c, k, and m are as defined in equation [4.1].
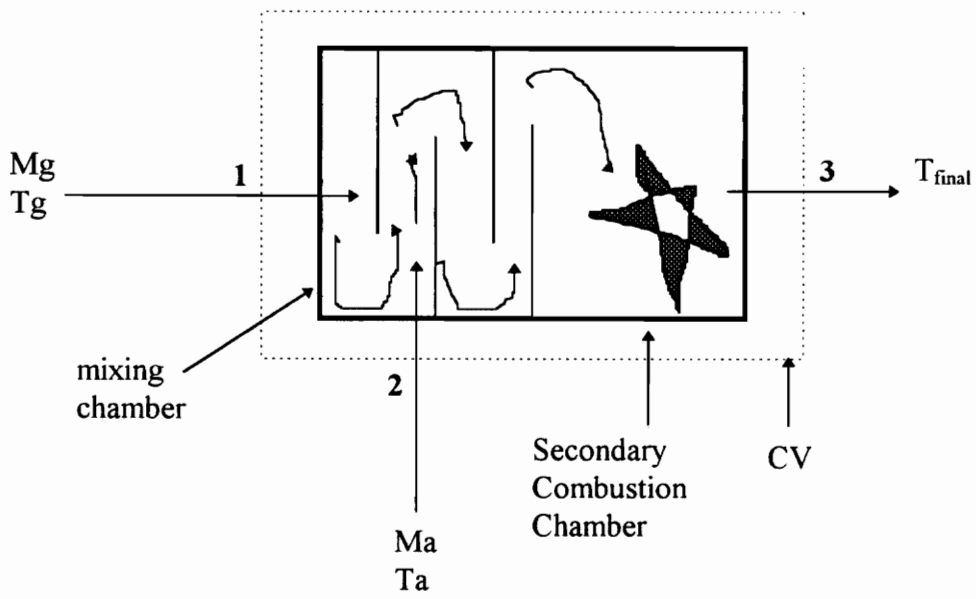
Figure 2.   Control volume for the mixing and secondary combustion chambers

Since the combustion products were at equilibrium, their flame temperature was calculated from equation[2.6]. Mass was conserved during the combustion process, accordingly, the mass and enthalpy of the final product was equal to the enthalpy and mass of the reactants.

The combustor heat loss decreases the energy of the producer gas. Therefore, it was accounted for by estimating 10, 20, and 30 percent combustor heat loss and subtracting it from the total enthalpy of reactants (J/kg reactants).

Adiabatic temperatures were calculated for moisture contents of 0, 25, 35, and 45 percent w.b. Equilibrium compositions were calculated at each of these moisture contents. Flame temperature was calculated for 10, 20 and 30 percent heat loss.

Chapter 5

RESULTS AND DISCUSSION

The results are presented in form of plots. Each plot is discussed separately.

5.1   Adiabatic flame temperature

Twenty-two calculations for the adiabatic system were done. The curve in Figure 3 illustrates the adiabatic flame temperature (AFT) as a function of equivalence ratio and moisture content. The AFTs are maximum at or near the equivalence ratio of one. AFT decreases with either an increase or decrease of air. The adverse effect of moisture content is clearly visible. It influences the entire process of gasification-combustion by lowering the AFT. This is attributed to the high superheated steam that is evolved in the gasification-combustion process that suppress the combustion temperature (AFT). The model predicted a maximum flame temperature of 2025°C. This is very close to values reported in the literature (2026.85°C) (Desrosiers, 1981).

5.2   Gasification region

Referring to Figure 4, as the equivalence ratio decreases from 10 to 3.5, the AFT rises gradually from 450 to 600°C for oak with moisture content of 25 percent (w.b). This is due to the low amount of air injected into the gasifier. As a result, the rate of carbon disappearance is very slow. Pyrolysis (the breaking down of fuel without oxygen) takes place in the region above equivalence ratio of 5. The equivalence ratio of 5 to 10

corresponds to a gasification equivalence (ϕ_g) of 0.2 to 0.1, which the percent of stoichiometric air supplied in the primary chamber. This range of ϕ_g corresponds to extremely low air supply to the gasifier, (1.22 to 0.61 kg of air per kg of dry fuel).

Equivalence ratios (ϕ) of 2.5 to 5 correspond to a gasification equivalence (ϕ_g) of 0.4 to 0.2. Meaning that 40 to 20 percent of primary air was used for gasification. As air is added beyond an equivalence ratio of 3.5, the increase in AFT is more pronounced. The sudden rise in AFT corresponds to the point at which carbon is burned off. The gasification region [equivalence ratios (ϕ) of 2.5 to 5] temperatures ranged from 600 to 900°C for dry wood. As air is added beyond an equivalence ratio of 2.5, AFT increases rapidly to the combustion temperature (2000°C).

The above result was consistent with values reported in literature (700 to 900°C) (Desrosier, 1981; Reed and Das, 1988). This temperature range will obviously decrease when heat loss and fuel moisture content are considered. For example, oak with 25 percent (w.b.) moisture content has a temperature range of 580 to 750°C (fig. 4).
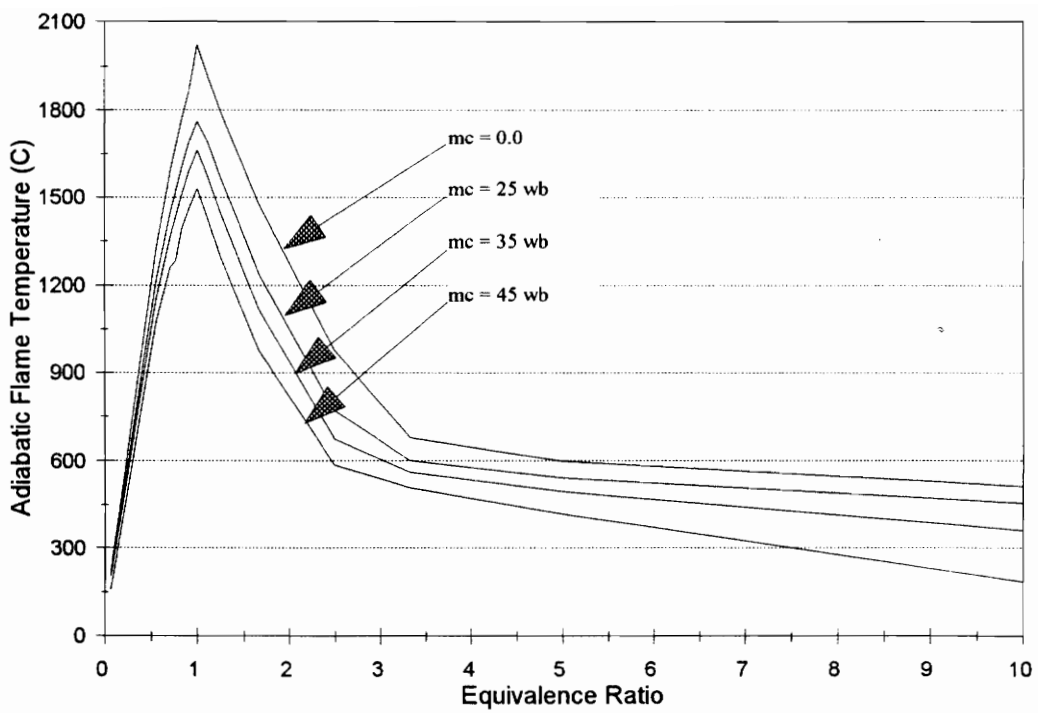
Figure 3.   Adiabatic flame temperatures calculated for oak as a function of equivalence
ratio.

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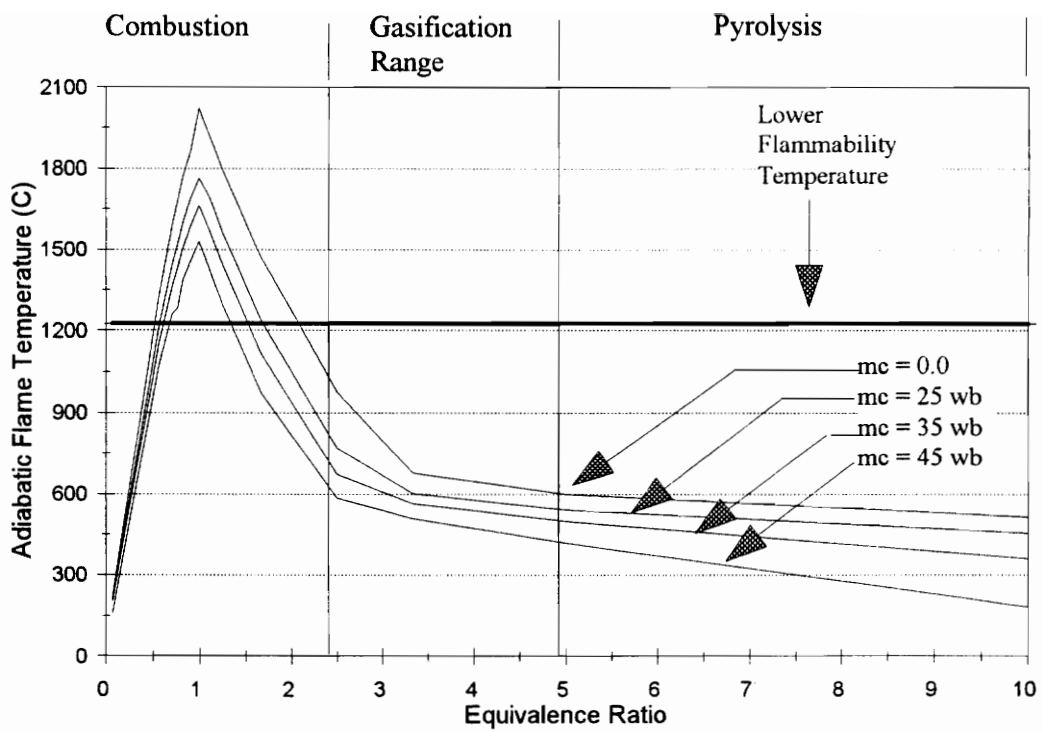
Figure 4.   Pyrolysis, gasification, and combustion of oak

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5.3   Combustion

The results presented here are for the combustion of the producer gas in the secondary combustion chamber. The equivalence ratio (ϕ) of the final mixture in the secondary combustion chamber was calculated based on the secondary air mass flow including the excess air.

Referring to Figure 5, the model predicted a flammable zone for adiabatic combustion of wood as characterized by an equivalence ratio (ϕ) between 0.56 and 1.67. From previous calculations, this range of equivalence ratio corresponds to an excess air (ε) range of 1.2 to 0.0 respectively. That is, an equivalence ratio of 0.56 would require 120 percent excess air to achieve complete combustion. For a gasification equivalence (ϕ_g) of 0.4, and an excess air (ε) range of 1.2 to 0.0, the model predicted SCC total air flow of 13.42 to 6.10 kg per kg of dry wood, assuming no heat loss.

The above result can be used to calculate design air requirement to burn a given charge, for example, burning a 45 kg charge of wet wood(m.c 25% w.b.) in one hour. Multiplying the dry wood weight by the lower air flow limit (1.67) and dividing by the density of wood (1.163 kg/m^3) , yields a total air flow of 2.42 cubic meters per minute. A similar calculation for the upper air flow limit yields 5.32 cubic meters per minute. However, for a batch-fed system, gas production fluctuates due to the non-uniform destruction of wood mass. To maintain the correct air-fuel ratio inside the SCC, it would be necessary to meter the secondary air flow through a variable control mechanism. Temperature could be used as a feedback parameter for a damper-control mechanism to regulate the air flow.
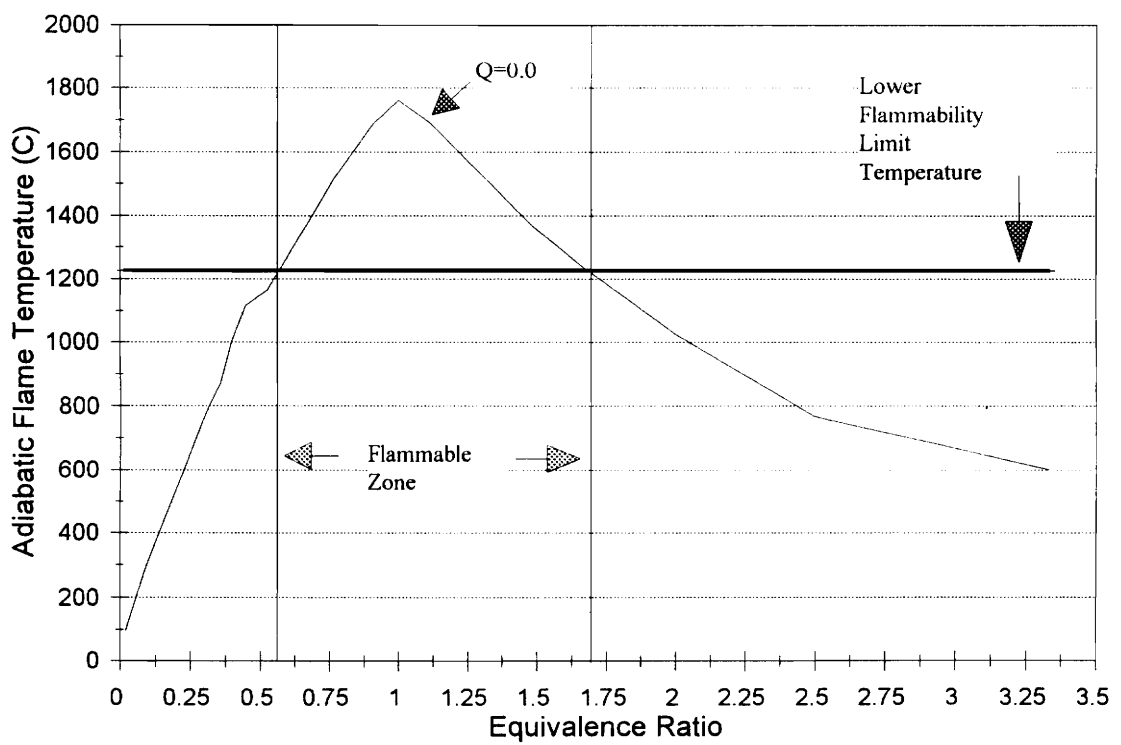
Figure 5.   Lower flammability temperature and the flammable zone for an adiabatic case

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5.4   Effect of heat loss on flammable zone

The most likely place for gasifier heat loss is to the surroundings, or to the cold secondary air. Figure 6 illustrates the flammability model prediction of the adverse effect of gasifier heat loss on the flammable zone . The thirty percent heat loss condition (region D-D) has the narrowest flammable zone corresponding to an equivalence ratio of 0.75 to 1.25. The adiabatic case (region A-A) has the widest flammable zone corresponding to an equivalence ratio of 0.56 to 1.65. Thus, the gasifier heat loss affects the flammability of the gas. For example, if the equivalence ratio is 0.56 for an adiabatic case, the excess air required (ε) was previously calculated as 1.20. If however, 10 percent of the heat produced by the gasifier is lost, the gas temperature drops to 1130°C, that is, about 100°C below the temperature for the gas-air mixture to support a flame. A 20 percent heat loss drops the temperature to 1060°C. Thus, even though the required amount of excess air is supplied, the gas-air mixture can not support a flame.

This result infers that, for a batch-fed system, if a constant volume of secondary air is supplied continuously it absorbs heat from the hot gases, thus cooling it. This is critical at the beginning and towards the end of the process, due to the fluctuation of gas production rate, and hence the equivalence ratio.

Heat loss for a specific gasifier-combustor system can be calculated based on the materials used for construction. To maintain flammability, for a specific heat loss, one remedy would be to decrease the secondary air flow. That is, instead of using the calculated air flow for an adiabatic case, the secondary air should be reduced proportionately to account for the heat loss. For example, for gases in a combustor with a 30 percent heat loss (region D-D in fig. 6) to support a flame, the mixture temperature must be above 1200°C. This temperature corresponds to an equivalence ratio of 0.75 to 1.25. From previous calculations these equivalence ratios corresponds to an excess air (ε) of 0.85 to 0.15 (fig.

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7). These ratios provide actual secondary air required for complete combustion considering a 30 percent heat loss.

The second alternative would be to pre-heat the secondary air to compensate for the heat loss. However, the best remedy would be to insulate the combustor.

5.5   Effect of excess air

Figure 7 illustrates the effect of excess air on adiabatic flame temperature. For a gasification equivalence, ϕ_g, of 0.4, the model predicts an excess air (ϵ) of 0.15 to 0.85 within which the gas-air- mixture temperature would be above 1200°C (the lower flammability limit temperature). The maximum combustion temperature was predicted as 1600°C with an excess air of 0.38.
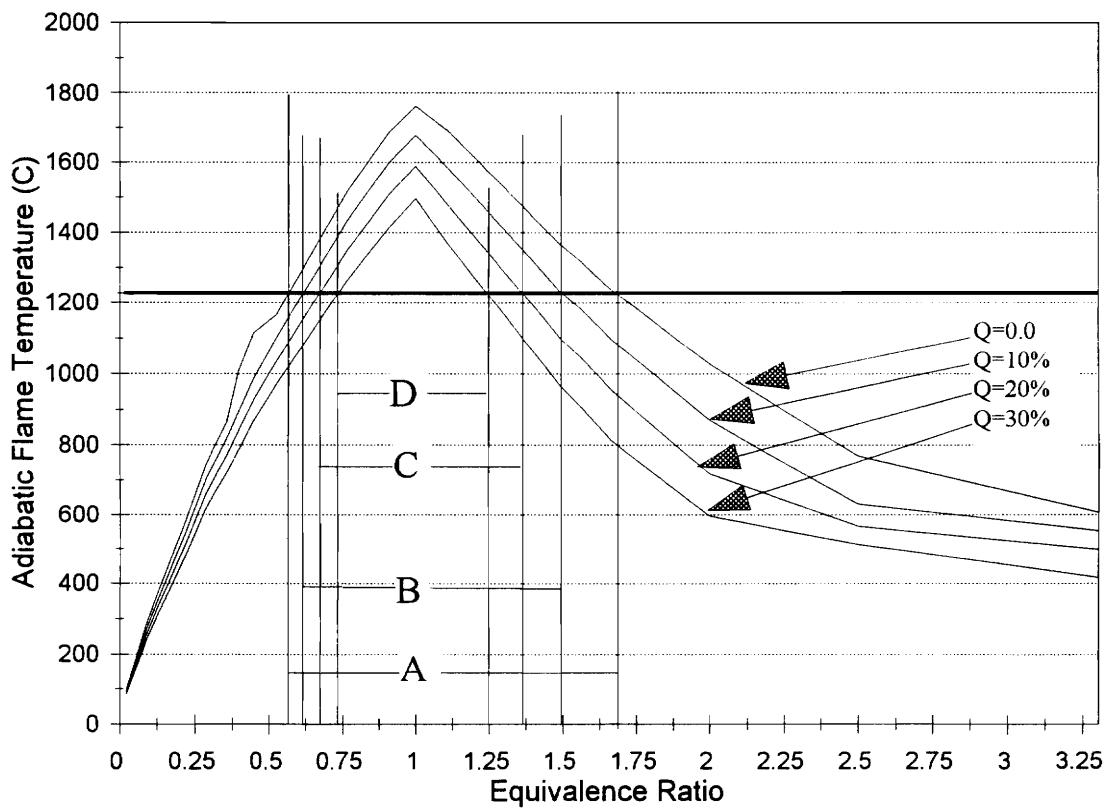
Figure 6.   Effect of heat loss on the flammable zone

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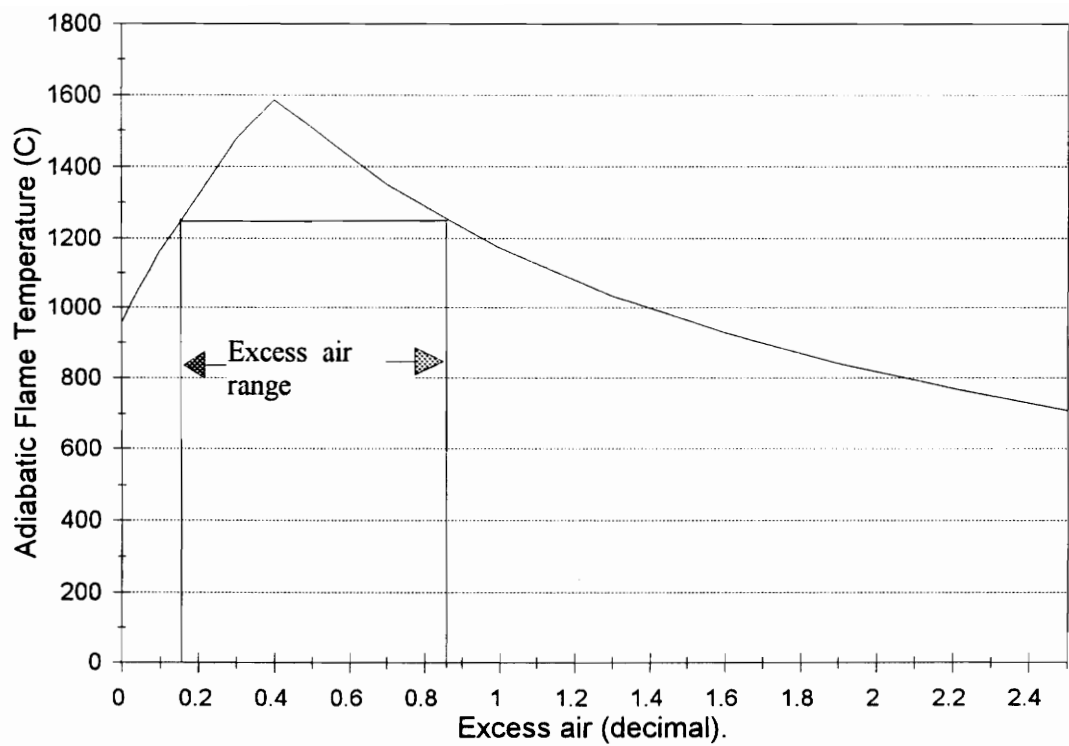
Figure 7.   Excess air range for gasification equivalence of 0.4, 20 % heat loss, and 25 %
wb moisture content

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5.6   Equilibrium Composition

Figures 8, 9, and 10 illustrate the composition of the combustion products at equivalence ratios of 0.25, 0.5, and 1. The composition consists of CO_2, H_2O, and N_2. These results are as predicted by the chemical equation used to calculate the stoichiometric air-fuel ratio: the products are water vapor, carbon dioxide and nitrogen. There are very small amounts of CO, CH_4, and H_2 present, but they are less than 0.0007 percent.

Figures 11 to 14 represent the equilibrium composition of the products at equivalence ratios (ϕ) of 1.25, 2.5, 3.3, and 5 respectively. As equivalence ratio increases, equilibrium concentrations for hydrogen and carbon monoxide increase. At an equivalence ratio of 1.25 and a temperature of 900°C, about 5% carbon monoxide starts to appear in the combustion products. Presence of carbon monoxide in combustion product indicates that, either enough air is not supplied for combustion, or that air is not completely mixed with the fuel. Also, AFT drops to about 1300°C because of lack of air, which results in incomplete combustion. Figures 12, 13, and 14 illustrate the effect of insufficient air on the AFT and the high concentrations of CO and other gases in the combustion products.

The advantages of equivalence ratios greater than 1 are realized during the gasification process. The concentrations illustrated in Figures 11 to 14 are the desirable products for gasification process. As equivalence ratio increases, equilibrium concentrations of CO and other combustible increase within the gasification temperature (600 to 900°C) for wood. For example, when ϕ is 2.5 (fig. 12) the maximum concentration of CO is 20 percent; concentrations of CO_2, H_2, and CH_4 are less than 20 percent. When ϕ is equal to 5 (Fig. 14), CO maximum concentration is 25 percent, H_2 is 33 percent, and CO_2 is 10 percent. High concentrations within the gasification temperature imply that the gases have more chemical energy. It is worth noting that ϕ equal to 5 corresponds to a gasification equivalence ϕ_g of 0.4

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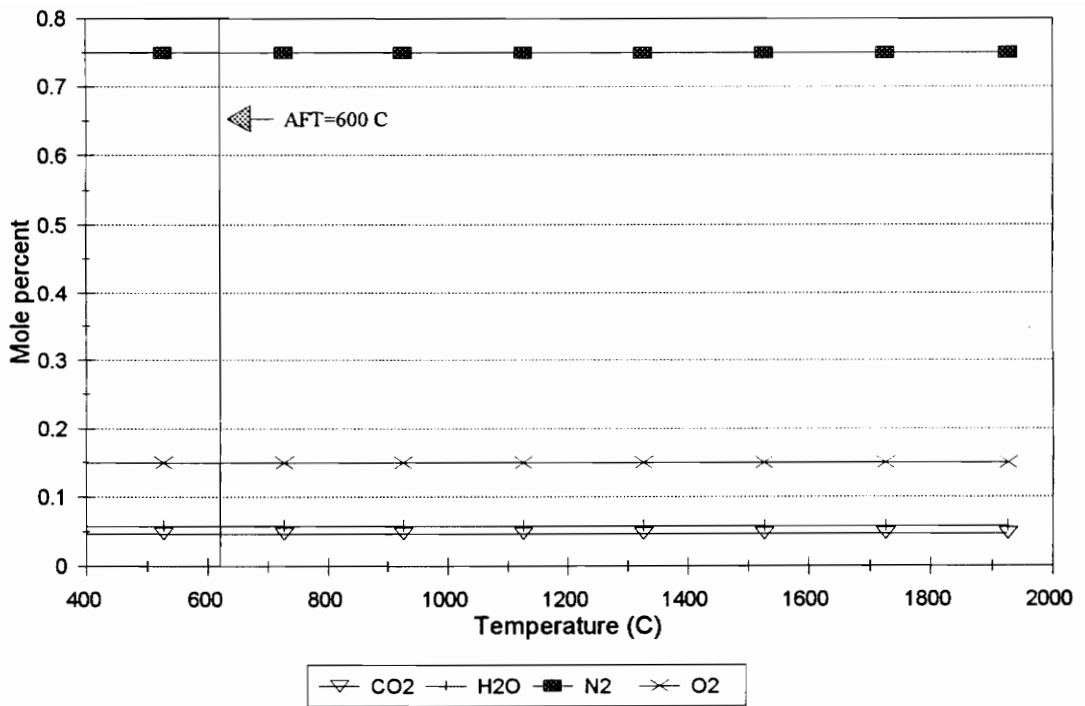
Figure 8.   Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4,

ϕ = 0.25.

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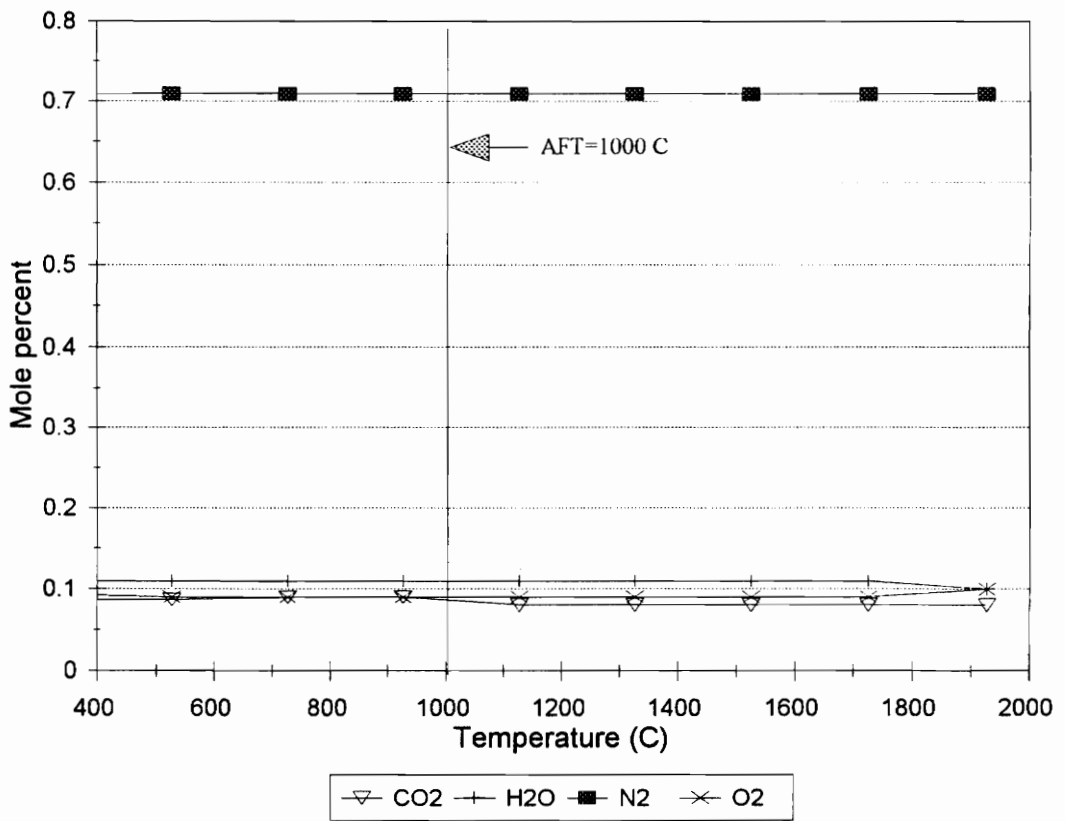
Figure 9.   Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4, ϕ =0.5

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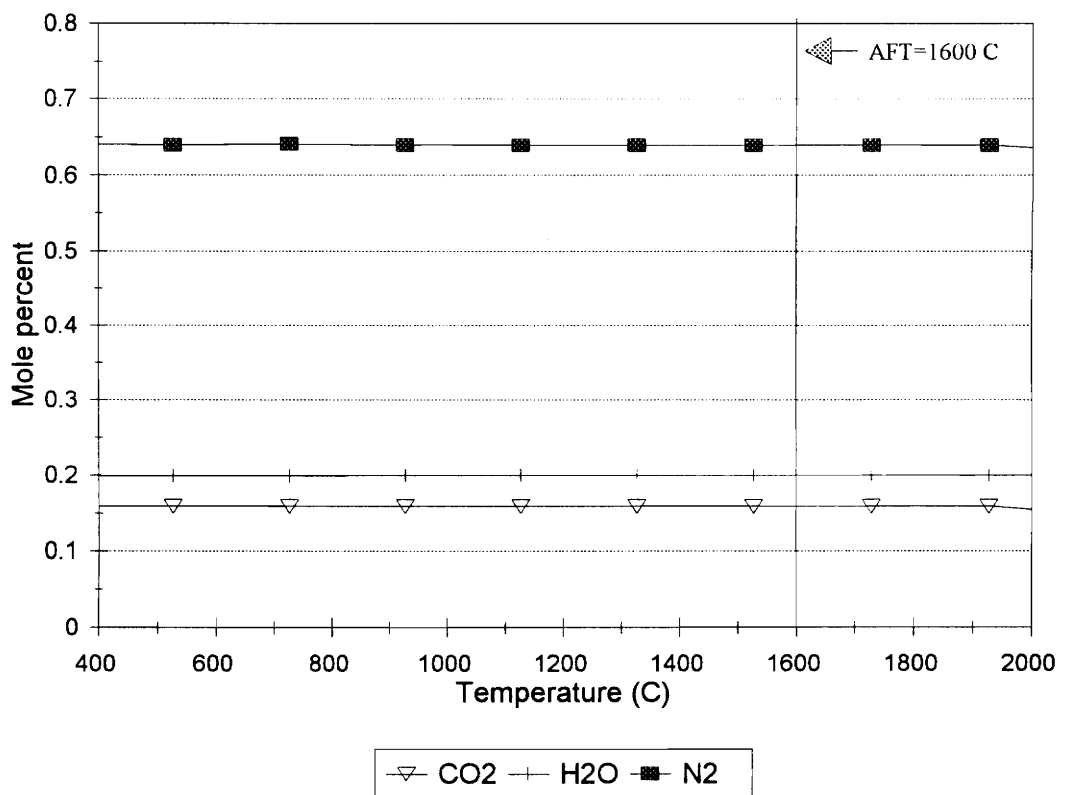
Figure 10.   Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4, ϕ =1.0

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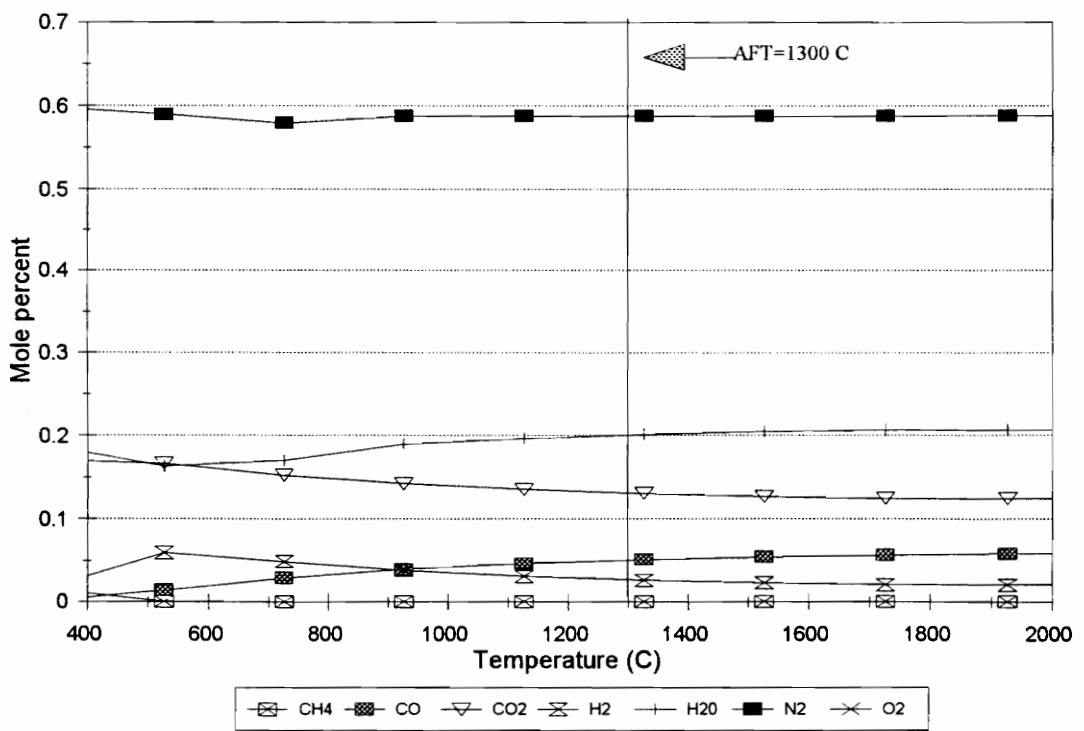
Figure 11. Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4,

ϕ =1.25.

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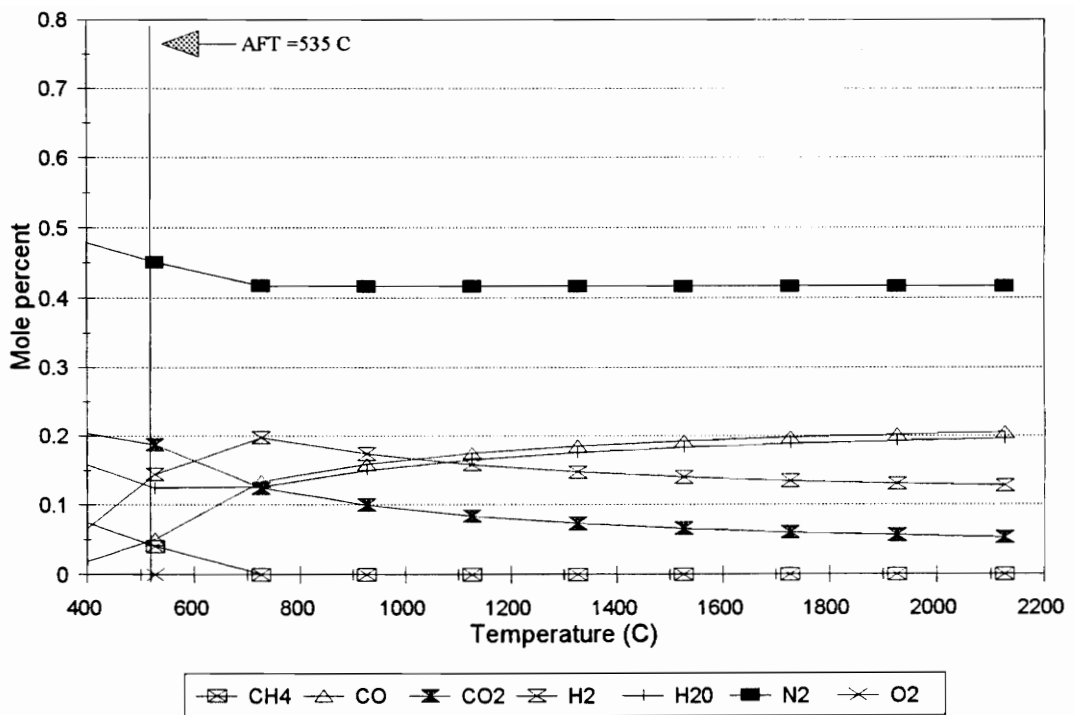
Figure 12.   Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4,

ϕ = 2.5

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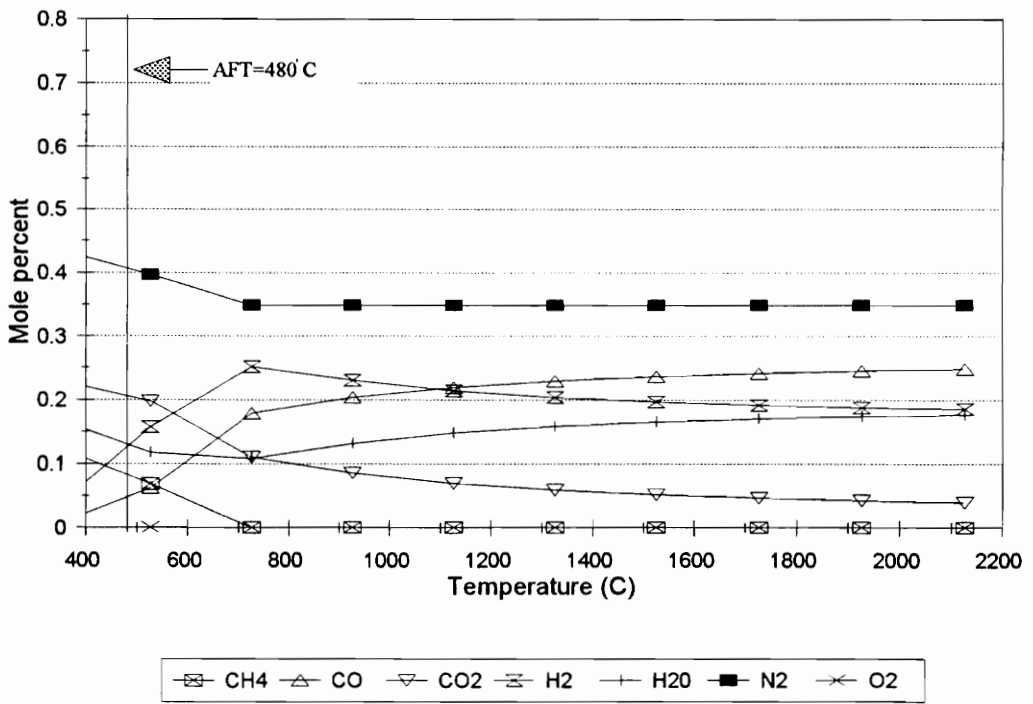
Figure 13. Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4, ϕ =3.3

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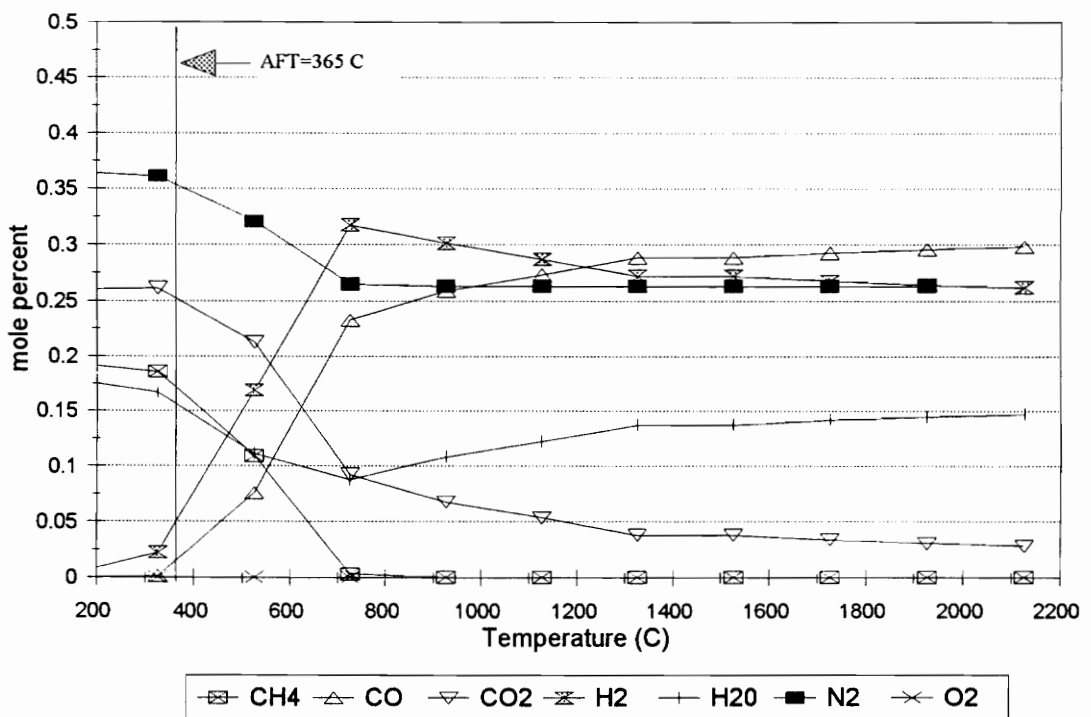
Figure 14.   Equilibrium concentrations, 25% mcwb oak, 20% heat loss, ϕ_g =0.4,

ϕ =5.0

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Chapter 6.0

CONCLUSIONS

Based on the results of this study, the following conclusions were made:

1. The flammability model developed here can be used to predict the performance of a gasifier-combustor. Specifically, the effect of various parameters (moisture, excess air, heat loss) on the equivalence ratio, and adiabatic flame temperature can be predicted.

2. Heat lost by the gasifier-combustor system (to the surroundings or absorbed by the secondary air) has a critical effect on the performance of the system.

3. The flammability of the gas-air mixture is largely determined by the amount of heat loss prior to combustion. The model predicts an adiabatic flammable zone with an equivalence ratio of 0.56 to 1.67 for oak wood with 25% (w.b) moisture content and a maximum adiabatic temperature of 2025°C for dry wood.

4. At equivalence ratios greater than 1.25, carbon monoxide appears in the combustion products. Presence of CO in the combustion products is an indication of incomplete combustion.

5. The model predicts a gasification temperature of 580 to 750°C for red oak at 25% (w.b.) moisture content and 20% heat loss.

6. Adiabatic flame temperatures are largely influenced by moisture and excess air. Temperature is a critical parameter that influences composition distribution of the gasification-combustion products. Composition distributions are important to the designer, for both energy and environmental impact.

7. Problems with the Bioprocessing laboratory's chunkwood gasifier-combustor, as predicted by the model are: excessive heat loss, poor mixing in the SCC, or a combination of the two. With required secondary air supply the combustion temperature (without kerosene flame) was always below 300°C.

8. Optimum performance of a gasifier-combustor system is obtained if heat loss, excess air, moisture content, mixing effectiveness, and residence time are optimized.

RECOMMENDATIONS

The model developed in this study was used to provide a solution to the problem experienced by the author while studying the fuel feedrate of an updraft gasifier. The following are recommended as possible direction for further research for the same gasifier:

1. The secondary combustion chamber was shown by this research to operate inefficiently. A likely solution would be to redesign the SCC. Critical potential areas include, provision for a mixing chamber, and redesign of the secondary air passages. The secondary air should enter at a low velocity and have ample time to mix with the producer gas before combustion. Other areas include increasing the insulation around the primary and secondary combustion chambers, including the fuel loading and the ash removal doors.

2. A study of the effect of a secondary air control mechanism should be done. A damper control mechanism could be used to regulate air supply to the SCC to maintain the air-fuel ratio required for the combustor to operate within the flammable zone.

3. Improve the way the chunks are distributed in the combustor. Investigate the effect of a grate inside the primary chamber. A simple grate positioned above the primary air ways would assist in primary air distribution, in addition, as ash is produced it would fall through the grate, allowing free air movement within the chunks.

4. The flammability model provided in this research should be verified to determine its ability to predict the flammable zone, equilibrium concentrations, and adiabatic flame temperatures.

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Appendix A1

PRESENTATION OF EXPERIMENTAL PROCEDURE AND PRELIMINARY RESULTS
FROM GASIFIER-COMBUSTOR AT THE DEPARTMENT OF
BIOLOGICAL SYSTEMS ENGINEERING.

THE RESULTS LEAD TO THE RESEARCH PRESENTED IN THIS STUDY.

1.0   INTRODUCTION

The objective of the original thesis was to investigate the control of chunkwood feedrate in the gasifier-combustor system (combustor) at the Department of Biological Systems Engineering. To gather the data needed to achieve this objective, the combustor had to operate accurately. Chunkwood pieces were introduced into the primary chamber, and gasified. The gasification product was expected to burn in an adjacent secondary combustion chamber to release thermal energy to be used for crop drying. Charges for chunkwood ranged from 35 to 45 kg per batch per hour. At the end of each test period the primary chamber contents were removed, and weighted; subtracting this amount from the original chunkwood weight yielded wood consumed per given period.

After several trials it was found that the objective could not be realized with the current equipment set-up due to inaccurate results. Consequently, a different thesis was selected to study the nature of the problems encountered in the preliminary tests.

The following is a summary of the study that was done in an attempt to achieve the original thesis objective. Though many tests were carried out, only three test results are presented here with a few details of the experimental procedure.

2.0   EXPERIMENTAL EQUIPMENT

2.1   Overview

The gasification-combustion equipment was built to supply heated water at low pressure for a typical on-farm drying operation. The desired features where the capacity to meet a variable demand, and to maintain high conversion efficiency. Figure 16 shows a plan view of the chunkwood heating system. Referring to Figure 17, fuel was charged in a batch varying from 35 to 45 kg per hour through a rectangular opening at the top left side of the primary chamber. Each charge was spread over the floor. Primary air flow was adjusted such that the chunkwood was gasified. The gases rose into a refractory-lined tunnel which was designed as the secondary combustion chamber. The other side of secondary combustion chamber was connected to a triple-pass heat exchanger. Water was circulated through the heat exchanger to a storage tank which acted as a 'thermal flywheel'. From the storage tank, water was pumped to another heat exchanger simulating demand in a crop drying application.

2.2   Primary combustion chamber construction

Figure 17 shows the cross-sectional view of the primary combustion chamber. The chamber walls were cast using refractory material. It was insulated with 5 cm mineral wool and covered with 18-gage sheet metal. The primary air was distributed through eight injections ports, 1.9 cm in diameter (fig. 18) located 20.3 cm apart. Figure 18 shows the left wall of the chamber, which has two injection ports for introduction of primary air, and a 36.8 cm x 62.2 cm rectangular opening where the wood was introduced. The back wall of the primary chamber is shown in figure 19. It has two injection ports, 18.4
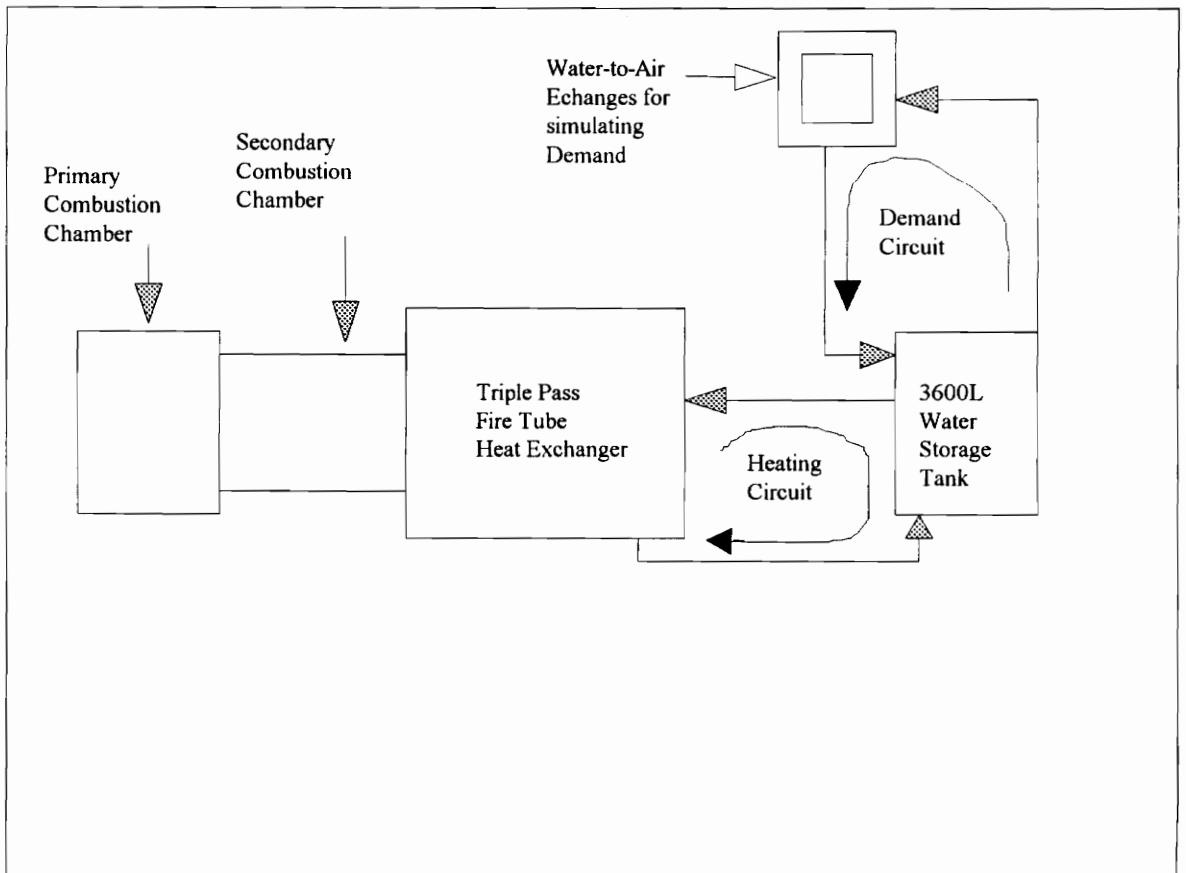
Figure 16.   Plan view of the chunkwood water-heater system

cm from the wall and 40.6 cm apart and a rectangular opening to the secondary combustion chamber. Figure 20 shows the right wall of the primary combustion chamber where the ash removal door was located.

Primary air was delivered by a variable-speed back-inclined fan. A 7.6 cm (3 in) diameter nozzle was mounted on the supply duct. Total air flow was determined by measuring pressure drop across this nozzle. Figure 22 shows the calibration curve of air flow versus pressure drop for the primary fun.

The primary air supplied by the fan flowed through a plenum around the bottom of the chamber (Fig. 21). The air split at the left plenum and flows to the front and back plenums. It was then injected into the fuel bed through eight air distribution ports. Thermocouple probes (chrome Alumel, ANSI Type K) were mounted at two locations in the combustion chamber to monitor producer gas temperatures from the primary chamber and SCC chamber temperature. A multi-channel digital thermometer (Omega Engineering model 2166A) was used to measure the temperatures.

2.3   Secondary combustion chamber construction

A 12.5 cm thick refractory-lined tunnel (fig. 16) cast as a single unit connected the back of the primary combustion chamber to a triple pass heat exchanger. This is where the combustion of gases resulting from wood gasification was expected to take place. To ensure a continuous flame in the SCC, a kerosene burner was installed.

Air into the secondary combustion chamber was supplied by a back-inclined, variable-speed fan. A 10.2 cm (4 in) diameter AMCA nozzle was mounted on the duct. Total air

flow was determined by measuring pressure drop across this nozzle. Secondary air was supplied through 10 injection ports, 1.3 cm in diameter and 6.4 cm apart. The injection ports were balanced to give uniform jets of air into the combustion zone. One additional port was installed midway each side wall. The injection ports were sloped such that air would be injected into the secondary combustion chamber at 45° degrees. The air was expected to gain heat from the refractory and therefore, was injected at a higher temperature. Consequently, it was not expected to cool the combustion zone.

A FOTRAN program 'FANCAL' included in Appendix A2, was used to calculate the secondary and primary air flow. The variables required to run FANCAL were: the fan speed in RPM, the inlet and exit nozzle diameters, the static pressure inside the duct, the pressure drop across the nozzle, the ambient and dry bulb temperatures, and the barometric pressure in inches of mercury. The ambient dry bulb temperature was obtained by measuring the resistance (kΩ) in the middle of the air stream by a thermistor probe connected to an ohmmeter. The ambient wet bulb temperature was obtained by measuring the resistance (kΩ) in the middle of the air stream by a thermistor probe (covered with a wetted wick) connected to an ohmmeter.
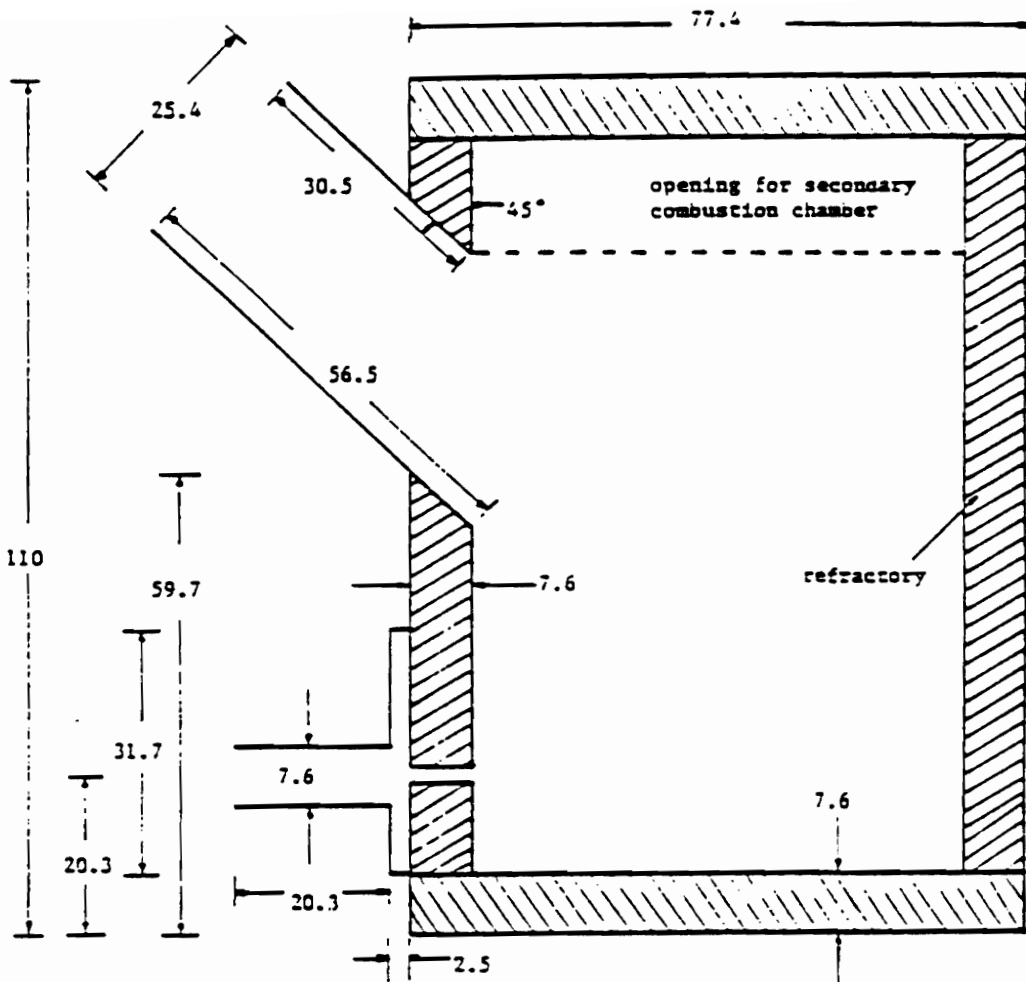
Figure 17.   The cross-sectional view of the primary combustion chamber (all dimensions in cm)

Appendix A1   67
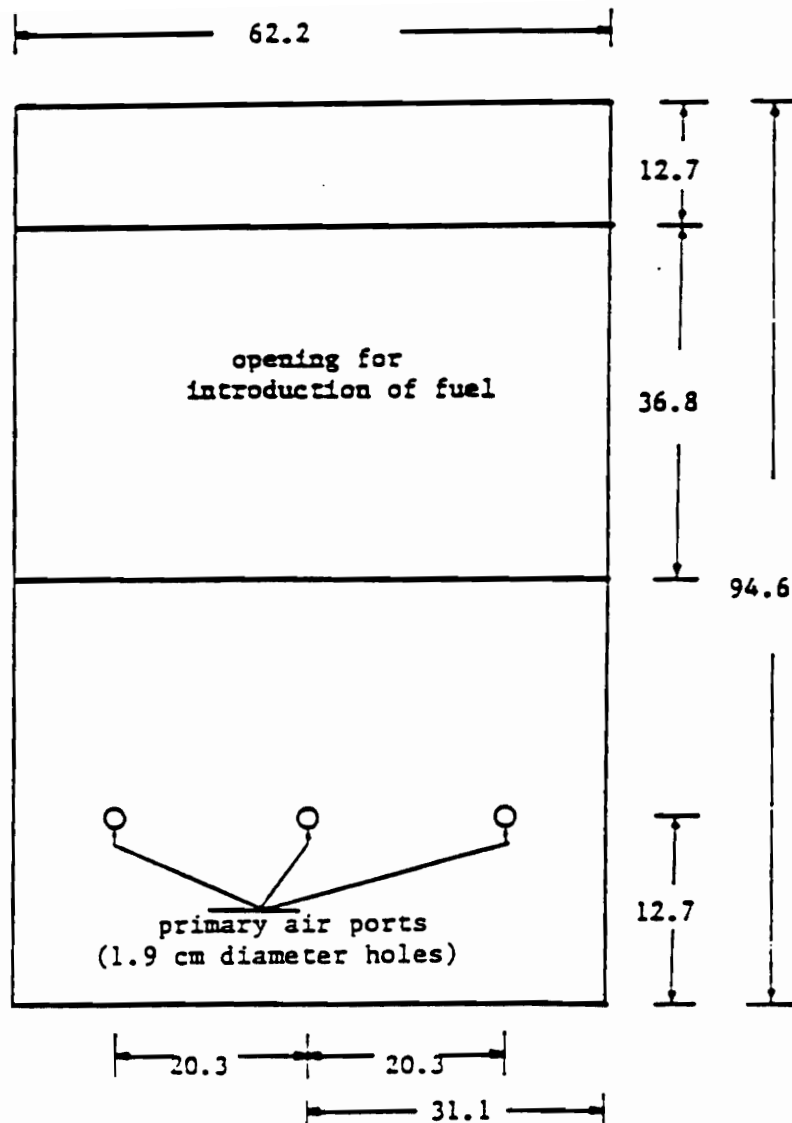
Figure 18.   Left wall of the primary combustion chamber (all dimensions in cm)
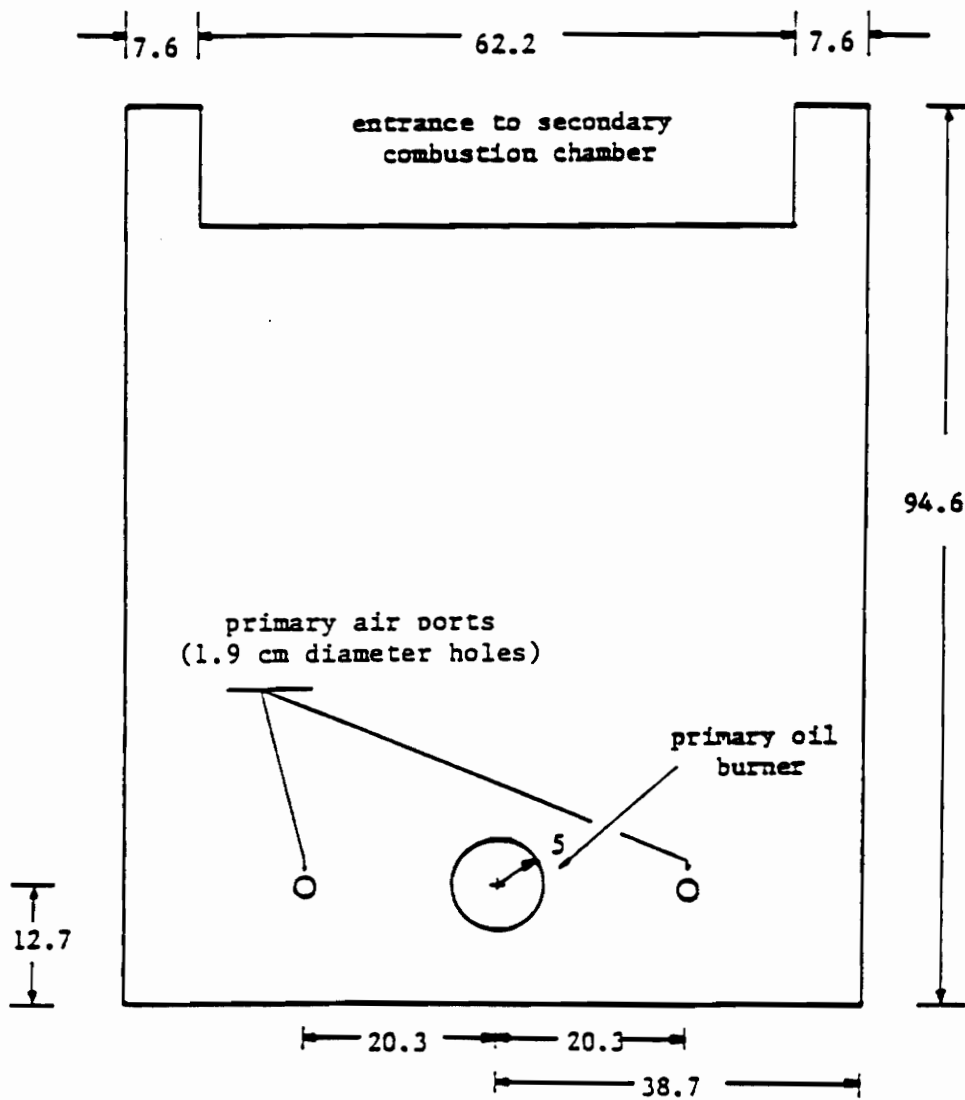
Figure 19. Back wall of primary combustion chamber (all dimensions in cm)

Appendix A1 69
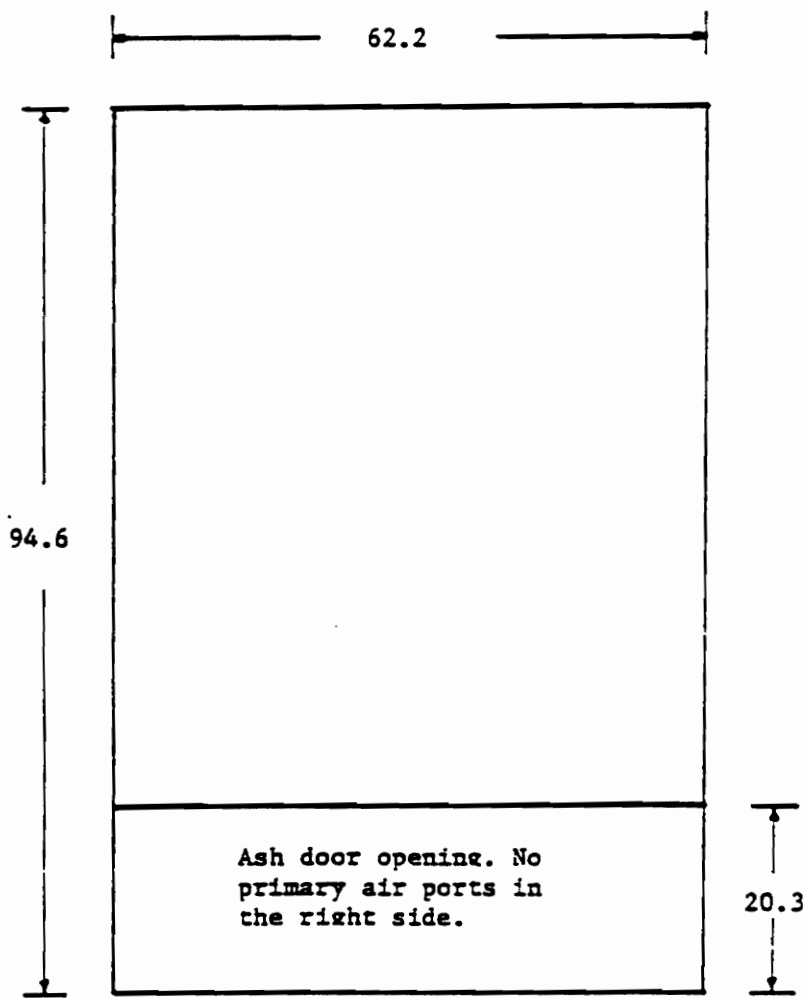
Figure 20. Right wall of primary combustion chamber (All dimensions in cm)

CROSS-SECTIONAL VIEW
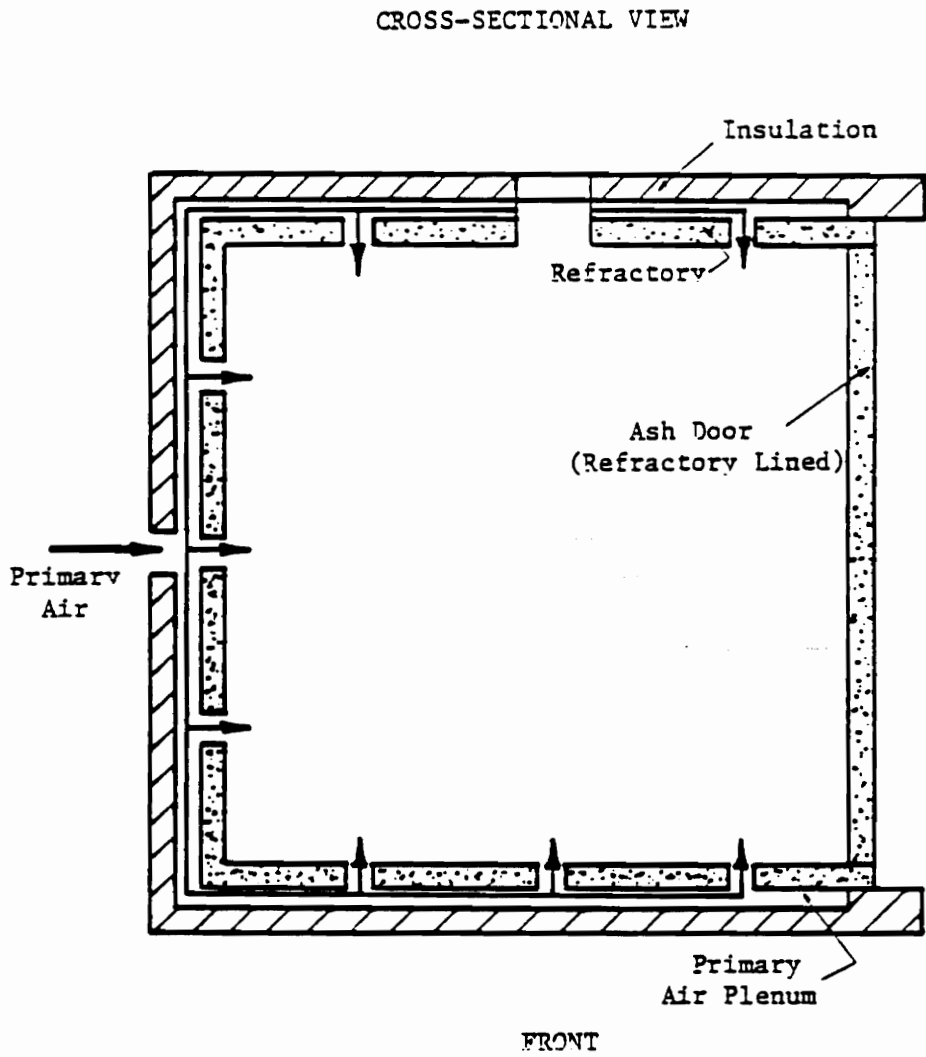
Figure 21.   Primary air plenum and injection ports of primary combustion chamber
              (Cross-sectional View)

The pressure drop across the nozzle and static pressure inside the duct were measured using a micromanometer (Dyer Microtector) with an accuracy of ± 0.0254 mm of water. A 60 tooth gear was mounted on the fan shaft. Each tooth generated a pulse as it passed near a magnetic sensor connected to a frequency counter. The pulses were displayed as RPM. The RPM were controlled by the variable speed drive. For a given RPM the flow rate through the duct was obtained by running the FANCAl program. Figures 22 and 23 shows the calibration curves for the primary and secondary fans.

2.4   Secondary Burner

A burner designed for a small oil furnace was used for the secondary burner, the burner was adjusted such that the flame was blown down stream in the secondary combustion chamber. Kerosene with a higher heating value of 29.9 MJ/kg was burned in the oil burner.

2.5   Heat exchanger

After combustion in the secondary combustion chamber, the hot exhaust gasses passed through the triple pass heat exchanger shown on Figure 24. The heat exchanger had 16, 7.6 cm diameter tubes per pass. The combustion gasses from the secondary combustion chamber pass through the tubes, and transfer heat to the water circulating outside of tubes. The exhaust gases escape to the atmosphere through a 20 cm diameter stack.
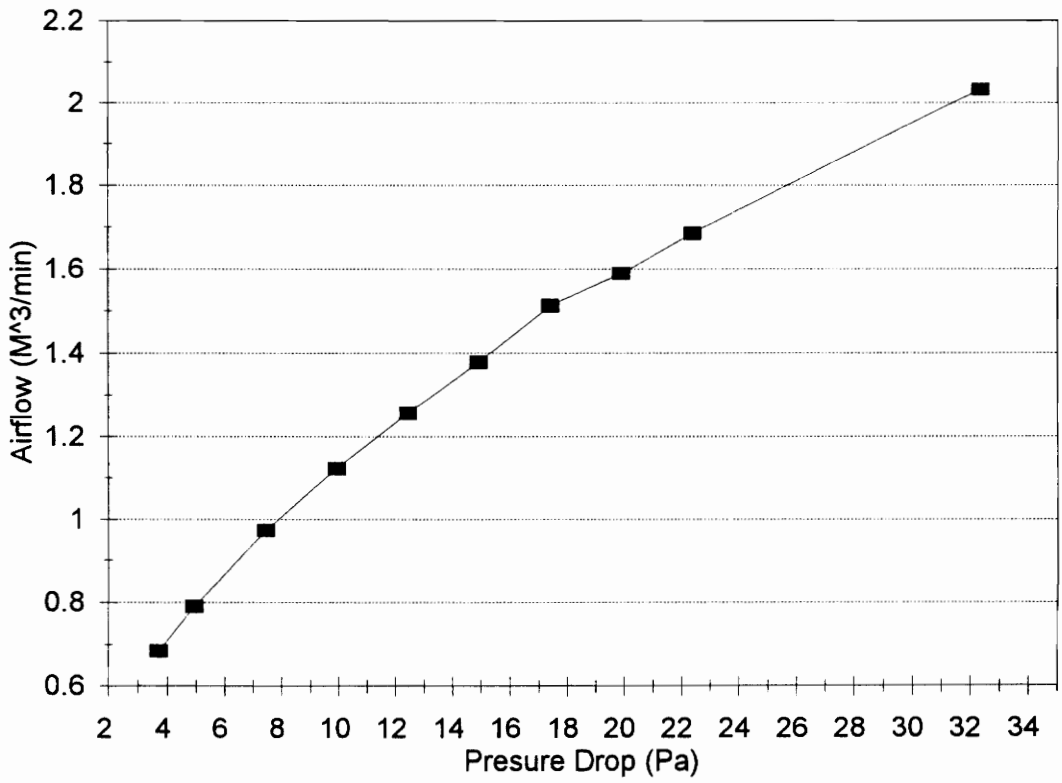
Figure 22.   Calibration curve of air flow versus pressure drop for primary fan
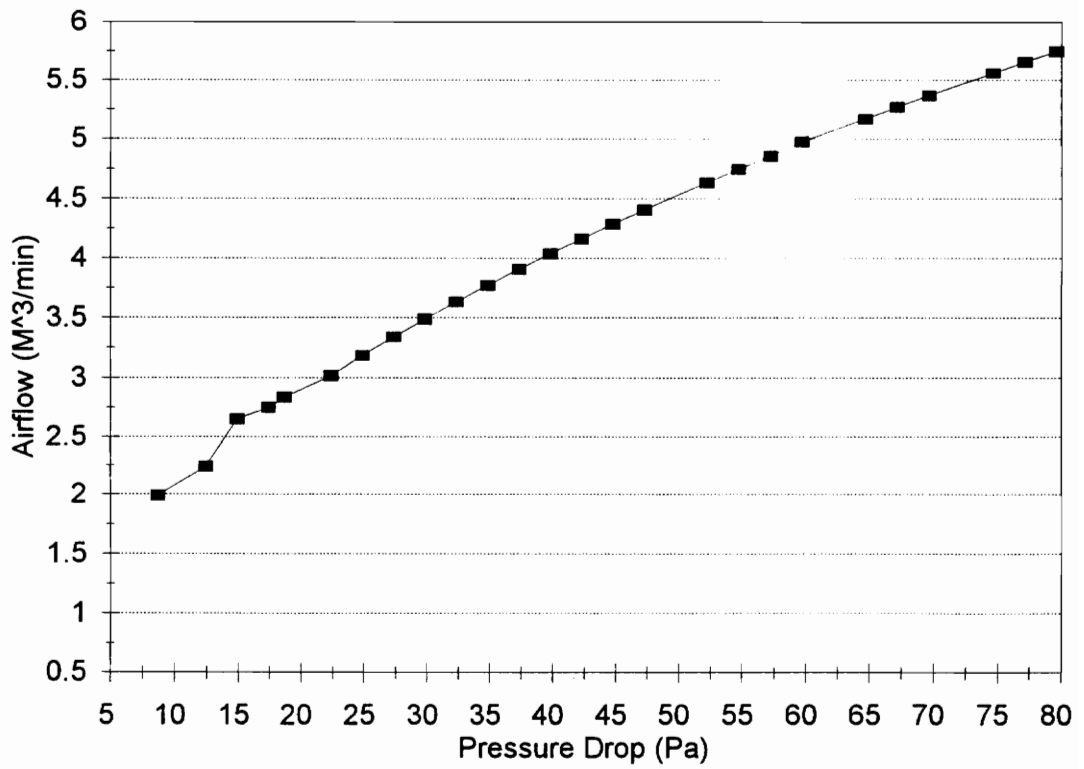
Figure 23.   Calibration curve of air flow versus pressure drop for secondary fan

END VIEW

SIDE VIEW (CROSS-SECTION)

16 TUBES
7.6cm DIAMETER

END PLATE REMOVABLE
TO CLEAN TUBES

Not to Scale
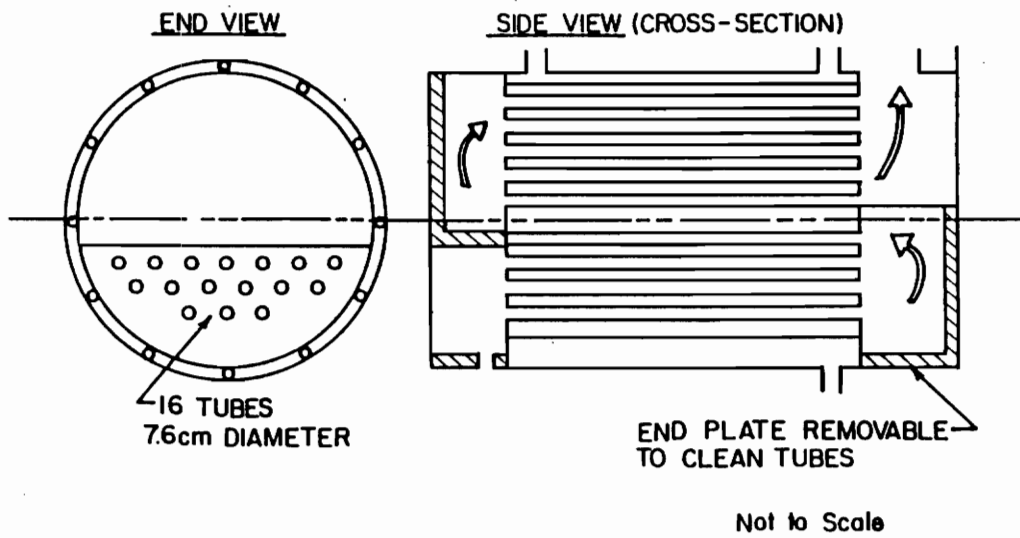
Figure 24.   Triple pass fire-tube heat exchanger

3.0 DESCRIPTION OF EXPERIMENTS

The air-fuel ratio of burning kerosene in air was calculated to be 14.8 kg of air per kg of kerosene ($C_{10}H_{20}$). On the basis of the air-fuel ratio and the amount of kerosene consumed per hour, the secondary burner requires 0.022 cubic meter of air per minute for complete combustion. The kerosene burner was set to introduce 0.076 cubic meters of air into the secondary combustion chamber. Therefore, the burner supplied 0.054 cubic meter per minute as excess air. The excess air was accounted for in the combustion calculations.

3.1 Experimental procedure

Air-fuel ratio for burning Red Oak was calculated based on the empirical formula ($C_{4.12}$ $H_{6.62} O_{2.73}$) and Tillman's general hydrocarbon combustion equation (Tillman, 1991). It was found to be 6.10 kg of air per kg of dry chunkwood.

Primary and secondary air supply requirements were calculated based on Payne's model of the biomass gasification-combustion process (Payne, 1985). The gasification combustion process of 35 kg wet wood with 20.56 percent moisture content (w.t), 40 percent excess air and a gasification equivalence ratio (Φ_g) of 0.36 required 0.871 cubic meters of primary air and 2.517 cubic meters of secondary air, a total of 3.388 cubic meters of air per minute.

A test run based on the above calculations was performed. Percent CO, CO_2, and combustion temperature in the secondary combustion chamber were monitored. The exhaust gases were analyzed using a digital CO/CO_2 analyzer. Data was collected for a period of one hour at a five minute interval. Several tests were done with different combinations of primary and secondary air flow.

The start-up procedure was as follow:

1. Ash was removed from the primary combustion chamber bed via the back door.
2. Primary and secondary air flow was set to the requirement.
3. The gas analyzer, and the temperature sensor were set-up.
4. Wood was weighted and introduced into the primary combustion chamber.
5. The secondary burner was switched on.
6. The fire was started manually.

4.0 RESULT AND DISCUSSION:

4.1   Test 35:36:104:140

The exhaust CO_2, CO, O_2 and temperature measured during test 1, hereafter referred to as test 35:0.36:104:140, are given in table 3. The notation **35:36:104:140** referred to: 35 kg of wet chunkwood with 20.56 percent moisture content, 36 percent of stoichiometric air in the primary chamber, 104 percent of stoichiometric air injected in the secondary chamber, for a total of 140 percent of stoichiometric air injected in the gasification-combustion equipment.

Referring to table 3, the percent CO started from zero, and increased to 0.45 percent within five minutes, and saturated at 1.61 percent (the meter saturation). After 55 minutes the level dropped to 1.10 percent and then increased to saturation level of 1.61 after five minutes. Percent CO_2 started at zero and increased to 4.9 percent within five minutes and then fluctuated between 6.1 and 6.7 percent for 45 minutes. CO_2 percent fluctuated between 0.47 and 0.7 percent in the last five minutes. The secondary combustion chamber temperature increased steadily from 21 to 273°C by the end of the test. Total O_2 and CO_2 concentration was assumed to be 21 percent. Therefore, percent O_2 was estimated by subtracting percent CO_2 reading from 21 (D. Jaasma, 1994 personal communication).

The above results indicate that the exhaust gases contain an average of 15.66 percent O_2 that was not used during gasification-combustion process, as evidenced by an average CO_2 percentage of 5.34. Higher CO_2 percentages (12 to 15 percent), which would indicate clean gasification-combustion process, are preferred (Dr. Jaasma, 1994 personal communication). Most of the air (74 percent) was injected in the secondary combustion

chamber. Consequently, the secondary air may not be necessary since most of it left the SCC unburned.

After one hour the primary chamber fuel level was inspected. It was found that most of the fuel was unburned and the flame was very small, producing light smoke. This observation implied that the gasification process produced inadequate gases for combustion in the SCC. When the primary chamber door was opened, thus letting in extra ambient air, the flame size increased significantly. The secondary combustion temperature increased from 273 to 390°C in less than five minutes. When the primary door was closed the secondary combustion temperature decreased. This implied that the primary chamber was air starved during the experiment, hence the unburned fuel.

Several short tests were conducted with the remaining fuel to examine the effect of the primary and secondary air: the two parameters that controlled the gasification-combustion process. The secondary air was turned off, leaving the primary air and secondary burner on. The secondary combustion temperature increased to 517°C within five minutes, percent CO remained saturated at 1.61 but CO_2 increased to 18.1 percent. The secondary air was turned off, amount of primary air doubled and primary chamber door secured: the SCC temperature stabilized at 432°C within five minutes. With the primary chamber door open the SCC temperature increased to 617°C within five minutes. It was evident that more primary air was needed for gasification than the combustion calculations implied.

Table 3. **Exhaust concentration and secondary chamber temperature for test 35:36:104:140**

Time (min.)	Percent CO	Percent CO_2	Percent O_2	CO/CO_2 Ratio	Secondary Combustion Temperature (°C)
0.00	0.00	0.00	21.00		119.00
5.00	0.45	4.90	16.10	0.09	254.00
10.00	1.61	6.10	14.90	0.26	214.00
15.00	1.61	6.30	14.70	0.26	205.00
20.00	1.61	6.20	14.80	0.26	215.00
25.00	1.61	6.60	14.40	0.24	235.00
30.00	1.61	6.40	14.60	0.25	244.00
35.00	1.61	6.20	14.80	0.26	248.00
40.00	1.61	6.40	14.60	0.25	241.00
45.00	1.61	6.50	14.50	0.25	262.00
50.00	1.61	6.70	14.30	0.24	265.00
55.00	1.10	6.40	14.60	0.17	266.00
60.00	1.61	0.70	20.30	2.30	273.00
		Average:	15.66	0.37	233.92

While the SCC temperature was at 617°C, primary air supply doubled, primary chamber door secured, and secondary air turned on, the temperature decreased to 387°C within five minutes. The primary chamber also started to leak excessively. Percent CO_2 decreased while percent CO remained saturated at 1.16 percent.

Three conclusions were made from these results: (1.) It was evident that the secondary air was cooling the SCC. (2.) Although the secondary air inlet was designed to inject air at an angle of 45 ° towards the exit of the SCC, some of it forced its way back to the primary chamber- hence the leaks. (3.) The rest of the secondary air left the SCC without burning and therefore percent CO_2 decreased a low CO_2 reading means that O_2 level was high in the flue gas.

A similar result was evident on all the previous tests whenever the computed amount of primary and secondary air was injected into the SCC. In view of the above evidence it was not practical to model the gasification-combustion process of the existing equipment after Payne's model.

Three tests were run beginning with 100 percent stoichiometric air in the primary, zero percent in secondary and the kerosene burner running. 54 and 160 percent excess air was injected in the SCC in subsequent tests. SCC temperature, percent CO and CO_2 were observed, and percent O_2 was computed as explained earlier.
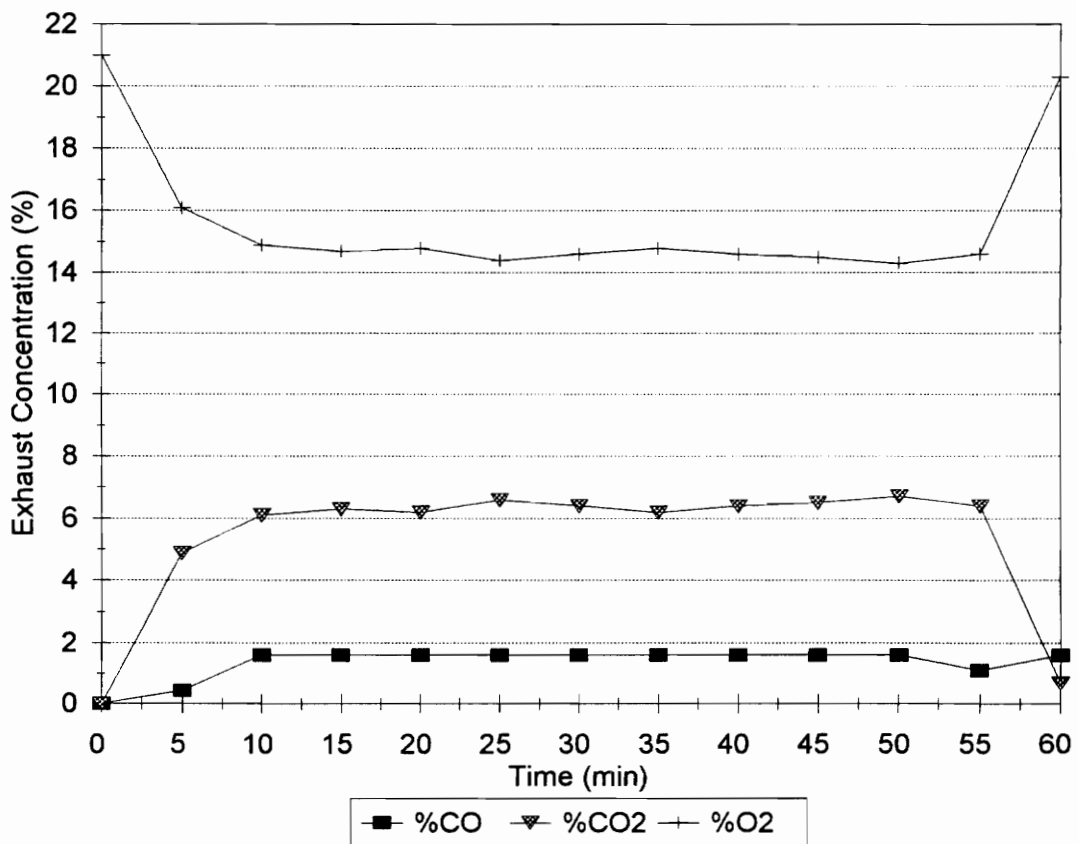
Figure 25.   Percent CO, CO_2, and O_2 concentrations for test: **35:36:104:140**
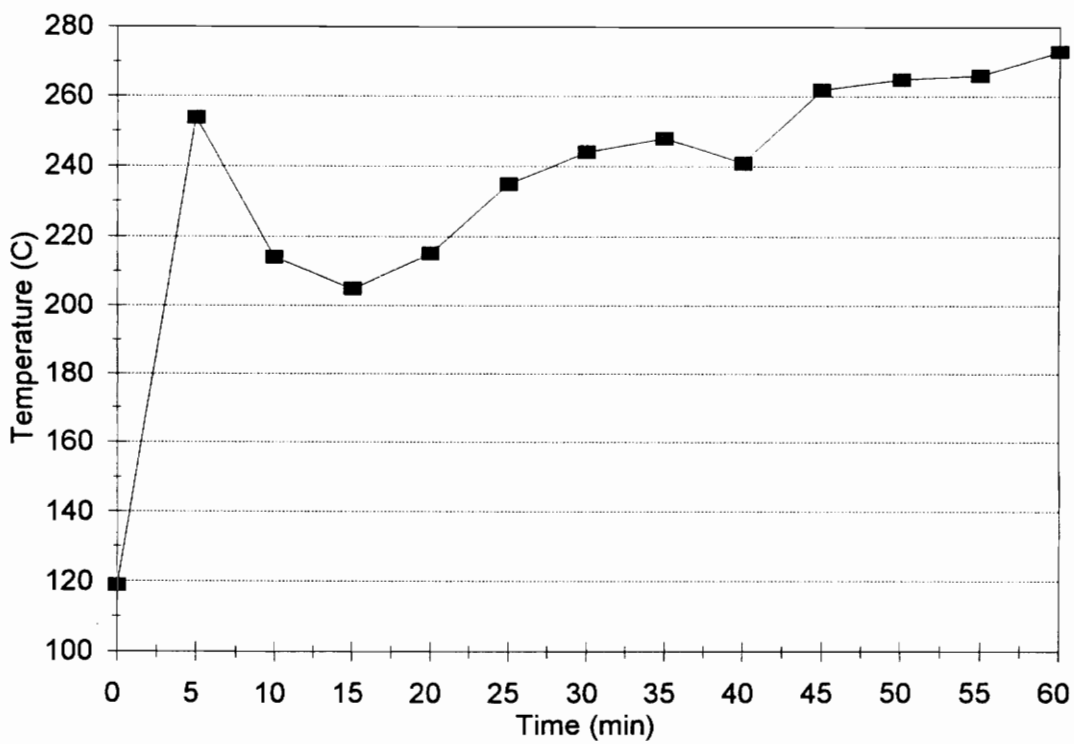
Figure 26.   Temperature profile for test : **35:36:104:140**

4.2   Test 35:72:28:100

The exhaust CO_2, CO, O_2 and temperature measured during test **35:72:28:100**, are given in table 4. Again, the notation refers to: 35 kg of wet chunkwood with 20.56 percent moisture content, 72 percent of stoichiometric air in the primary chamber, 28 percent of stoichiometric air injected in the secondary chamber, for a total of 100 percent of stoichiometric air injected in the gasification-combustion equipment. The secondary burner was set to supply 0.054 cubic meters of excess air per min. This was the only secondary air injected into the SCC for test 35:72:28:100.

Referring to table 4, percent CO got saturated at 1.61 percent immediately after the test begun.   Towards the end of the test percent CO decreased to 0.28 percent. Percent CO_2 started at 7.3 percent, reaching a maximum halfway through the test at 10.1 percent and decreasing to 5.90 after 60 minutes. The SCC temperature started at 794, reaching a maximum of 880 after 40 minutes and decreasing to 833 after 60 minutes. The combustion cleanness for this test as indicated by the CO/CO2 ratio was below average (0.16). A CO/CO_2 ratio of 0.01 is considered excellent combustion, 0.1 average, and 0.3 dirty (D. Jaasma, 1994 personal communication).

4.3   Test 35:52:100:152

The exhaust CO_2, CO, O_2, concentrations, and temperature measured during test **35:52:100:152** are give in table 5. Referring to table 5, percent CO started at 0.1 and increased steadily to saturation level of 1.61 after 20 minutes and remained saturated for 25 minutes decreasing to 0.82 percent at the end of the test.. Percent CO_2 started at 3.6 percent, reaching a maximum after 25 minutes--halfway through the test-- and decreasing

steadily to 9.90 percent after 60 minutes. The SCC temperature started at 745 °C, reaching a maximum of 840 after 40 minutes and decreasing to 799 °C after 60 minutes. The combustion cleanness for this test as indicated by the CO/CO_2 ratio was averaged 0.086.

Visual examination of each test showed that a substantial amount of gas was produced in the primary chamber. However, the expected temperatures in the SCC was not realized. The CO_2 level for tests 2, and 3 indicated that more oxygen was needed, especially at the beginning and towards the end of the a test. However, an attempt to inject any secondary air decreased the SCC temperature immediately and significantly.

Table 4.   Exhaust concentration and secondary chamber temperature for test 35:132:128:260

Time (min.)	Percent CO	Percent CO_2	Percent O_2	CO/CO_2 Ratio	Secondary Combustion Temperature (°C)
0.00	0.10	3.60	17.40	0.03	745.00
5.00	0.08	2.80	18.20	0.03	797.00
10.00	0.35	3.90	17.10	0.09	804.00
15.00	1.45	6.50	14.50	0.22	800.00
20.00	1.61	7.30	13.70	0.22	790.00
25.00	1.61	19.80	1.20	0.08	823.00
30.00	1.61	13.40	7.60	0.12	812.00
35.00	1.61	19.00	2.00	0.08	830.00
40.00	1.40	14.00	7.00	0.10	842.00
45.00	0.66	14.00	7.00	0.05	824.00
50.00	0.07	14.10	6.90	0.00	814.00
55.00	0.09	11.30	9.70	0.01	806.00
60.00	0.82	9.90	11.10	0.08	799.00
	Average:	10.74	10.26	0.09	806.62
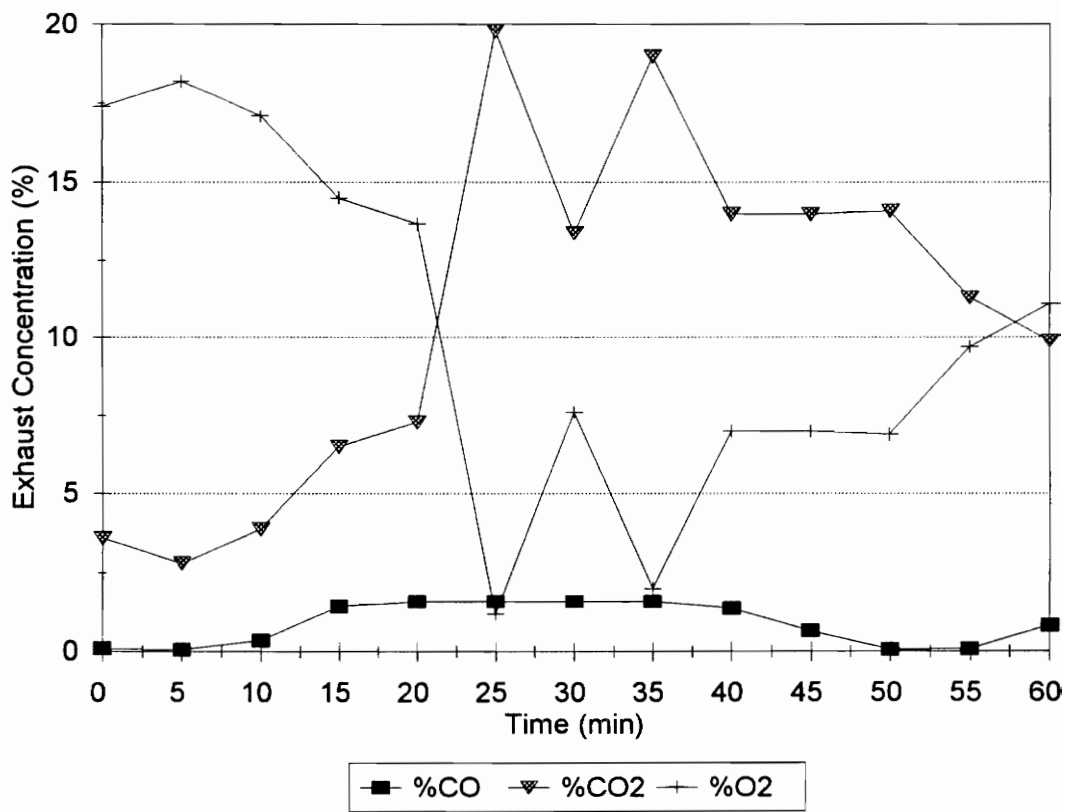
Figure 27. Percent CO, CO_2, and O_2 concentrations for test: **35:132:128:260**
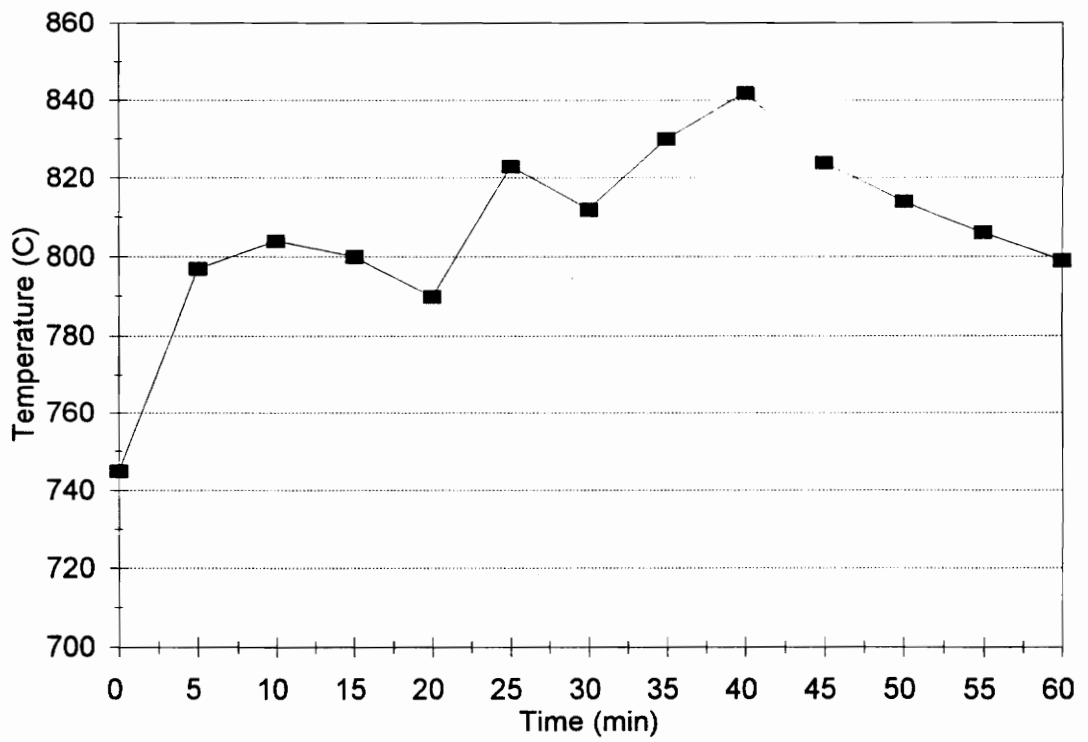
Figure 28.   Temperature profile for test : **35:132:128:260**

Table 5.   **Exhaust concentration and secondary chamber temperature for test 35:52:100:152**

Time (min.)	Percent CO	Percent CO_2	Percent O_2	CO/CO_2 Ratio	Secondary Combustion Temperature (°C)
0.00	1.61	7.30	13.70	0.22	794.00
5.00	1.57	7.10	13.90	0.22	828.00
10.00	1.61	8.20	12.80	0.20	821.00
15.00	1.61	8.30	12.70	0.19	830.00
20.00	1.61	9.20	11.80	0.18	851.00
25.00	1.61	10.10	10.90	0.16	870.00
30.00	1.61	10.30	10.70	0.16	871.00
35.00	1.61	10.10	10.90	0.16	872.00
40.00	1.61	9.60	11.40	0.17	880.00
45.00	1.61	10.00	11.00	0.16	869.00
50.00	0.90	7.80	13.20	0.12	850.00
55.00	0.32	7.20	13.80	0.04	849.00
60.00	0.28	5.90	15.10	0.05	833.00
	Average:	8.55	12.45	0.16	847.54
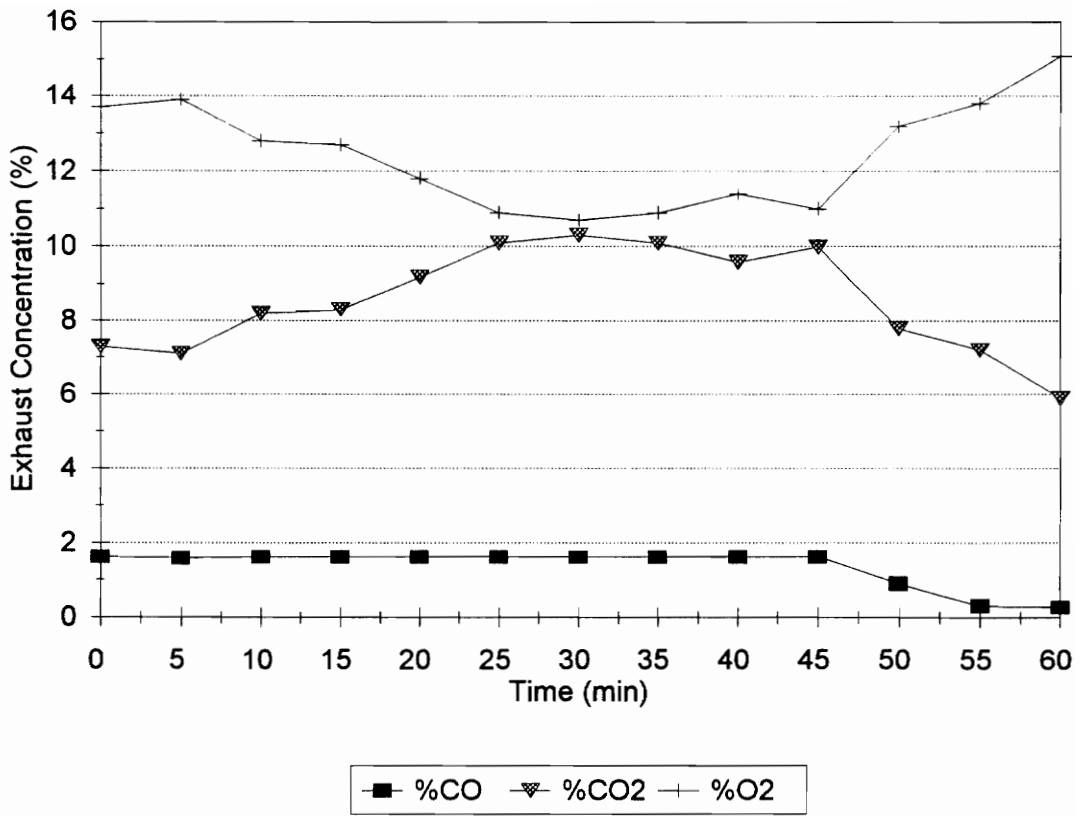
Figure 29.   Percent CO, CO$_2$, and O$_2$ concentrations for test: **35:52:100:152**
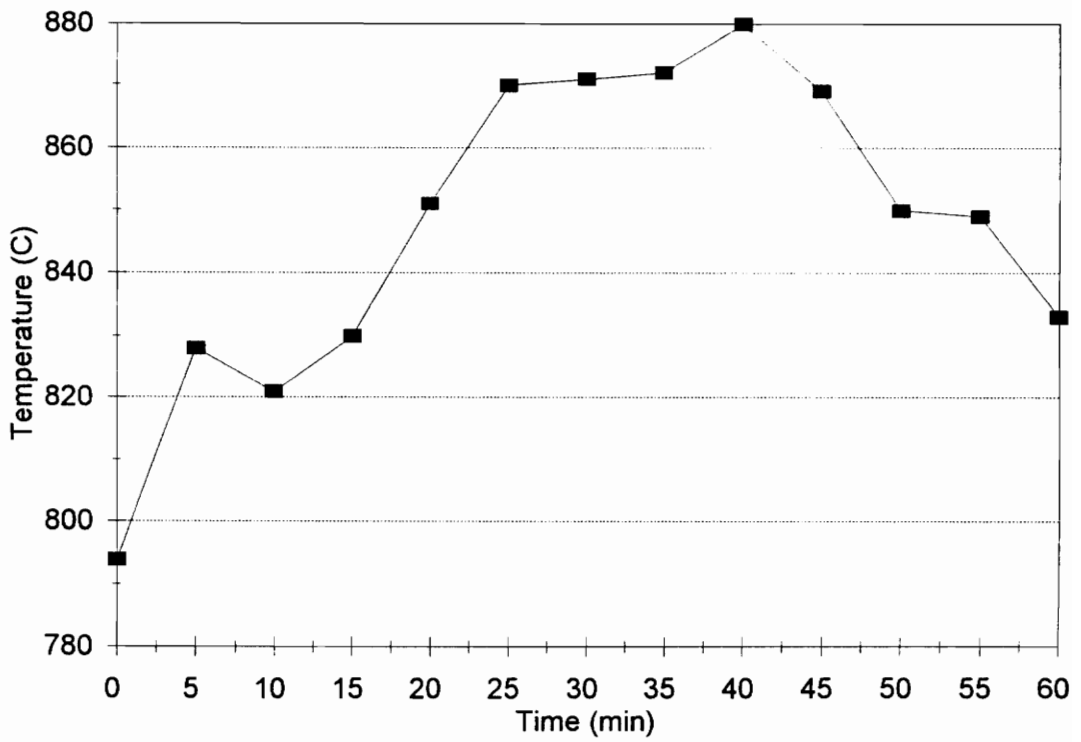
Figure 30.   Temperature profile for test : **35:52:100:152**

Several other tests exhibited the same characteristics as described above. The result made it impossible to proceed with the initial objective of investigating chunkwood feedrate to optimize the combustor.

5.0 Conclusion

Due to the technical problems encountered by the author, it was decided that the initial objective was not feasible within the time frame and economic constrains imposed by my thesis committee. Consequently, a new concept was to be explored. The author opted for a study of the flammable range of the gasification-combustion process using the equilibrium and flammability theory.

Appendix A2

FORTRAN PROGRAM USED TO CALCULATE AIR FLOW

```
C
C**********************************************************************************
C*** FANCAL FORTRAN: USED TO CALCULATE THE AIRFLOW FROM A STANDARD        **
C*** TEST. PROGRAM CAN BE USED FOR A SINGLE NOZZLE MOUNTED IN A           **
C*** DUCT, OR A SINGLE NOZZLE MOUNTED IN A CHAMBER.                       **
C**********************************************************************************
C234567
      DIMENSION PD(100),SP(100),QN(100),QF(100),FS(100),POUT(9)
      INTEGER FS
      DATA PI/3.141592654/
C
C**********************************************************************************
C***  INPUT AND OUTPUT FILEDEFS                                           **
C**********************************************************************************
C
      OPEN(UNIT=34,FILE='FANCAL.DAT',STATUS='OLD')
      OPEN(UNIT=20,FILE='FANCAL.OUT',STATUS='UNKNOWN')
C
C**********************************************************************************
C***  READ IN INPUT DATA FROM FILE (FANCAL DATA A)                        **
C**********************************************************************************
C
      READ(34,1) TDBA,TWBA,TDBD,PB,NP
    1 FORMAT(3F6.3,F6.2,I3)
      DO 100 I=1,NP
  100 READ(34,2) PD(I),SP(I),FS(I)
    2 FORMAT(2F7.2,I5)
      READ(34,3) D4,D6,NCODE
    3 FORMAT(2F3.0,I2)
C
C**********************************************************************************
C*** MOUT = FILEDEF FOR OUTPUT FILE (FANCAL OUTPUT A)                     **
C**********************************************************************************
C
      MOUT=20
C
C**********************************************************************************
C***  INPUT DATA FROM TERMINAL                                           **
C**********************************************************************************
C
C     WRITE(9,1)
C   1 FORMAT('ENTER THE AMBIENT DRY BULB TEMPERATURE(KOHMS)')
C     READ(9,*) TDBA
C     WRITE(9,2)
C   2 FORMAT('ENTER THE AMBIENT WET BULB TEMPERATURE(KOHMS)')
C     READ(9,*) TWBA
C     WRITE(9,3)
```

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```
C  3 FORMAT('ENTER THE BARAOMETRIC PRESSURE FOR THE TEST(IN.HG)')
C    READ(9,*) PB
C
C*******************************************************************************
C*** CALCULATE ATMOSPERIC PRESSURE IN PASCALS                               **
C*******************************************************************************
C
   PATM=PB*3386.53
C    WRITE(9,7)
C  7 FORMAT('ENTER THE DUCT DRY BULB TEMPERATURE(KOHMS)')
C    READ(9,*) TDBD
C    WRITE(9,8)
C  8 FORMAT('ENTER THE NUMBER OF NOZZLE PRESSURE DROP-STATIC PRESSURE R
C   *EADINGS...')
C    READ(9,*) NP
C    DO 15 I=1,NP
C    WRITE(9,9) I
C  9 FORMAT('ENTER READING NUMBER, ',I3,2X,'PRES. DROP, ST. PRESS.,R
C   $PM')
C
C*******************************************************************************
C*** INPUT VARIABLES:                                                       **
C***   1. PD = PRESSURE DROP ACROSS NOZZLE                                  **
C***   2. SP = STATIC PRESSURE(IN. W.G.)                                    **
C***   3. FS = FAN RPM                                                      **
C*******************************************************************************
C
C    READ(9,*) PD(I),SP(I),FS(I)
C 15 CONTINUE
C    WRITE(9,11)
C 11 FORMAT('ENTER THE NOZZLE INLET DIAMETER(IN.)')
C    READ(9,*) D4
C    WRITE(9,12)
C 12 FORMAT('ENTER THE NOZZLE EXHAUST DIAMETER(IN.)')
C    READ(9,*) D6
C    WRITE(9,13)
C 13 FORMAT('IF APPROACH TO THE NOZZLE IA A DUCT, ENTER "0"')
C    WRITE(9,14)
C 14 FORMAT('IF APPROACH TO THE NOZZLE IS A CHAMBER, ENTER "1"')
C    READ(9,*) NCODE
C
C*******************************************************************************
C*** CONVERT TEMPERATURE(K-OHMS) TO TEMPERATURE(DEG C)                       **
C*******************************************************************************
C
   TDBA=THERM(TDBA)
   TWBA=THERM(TWBA)
   TDBD=THERM(TDBD)
C
C*******************************************************************************
C*** CALCULATE AREA OF NOZZLE DISCHARGE(FT**2)                              **
```

```
C*****************************************************************************
C
      D6=D6/12.
      D4=D4/12.
      A6=PI*(D6**2)/4.
C
C********************************************************************** *****
C***  DEFINITION OF VARIABLES:                                            **
C***  1. NCODE = 0 FOR A DUCT APPROACH; NCODE = 1 FOR A CHAMBER           **
C***        APPROACH                                                      **
C***  2. GAMMA = RATIO OF AIR SPECIFIC HEAT AT CONSTANT PRESS.            **
C***        TO AIR SPECIFIC HEAT AT CONSTANT VOL.                         **
C***  3. R = GAS CONSTANT(FT-LB/LBM-DEG R)                                **
C*************************************************************** *****
C
      GAMMA=1.4
      R=53.35
C
C*****************************************************************************
C*** DETERMINE THE SPECIFIC VOLUME OF THE AMBIENT AIR                     **
C*** (SUBROUTINE RETURNS SPECIFIC VOLUME(M**3/KG DRY AIR) AND THIS        **
C***   MUST BE CONVERTED TO FT**3/LBM DRY AIR)                            **
C*****************************************************************************
C
      MODE=2
      CALL PSYC(MODE,TDBA,TWBA,PATM,MOUT,POUT)
      SPV=POUT(9)*16.018
C
C*****************************************************************************
C***  CONVERT TEMPERATURES FROM DEG C TO DEG F                            **
C*****************************************************************************
C
      TDBA=1.8*TDBA+32.
      TWBA=1.8*TWBA+32.
      TDBD=1.8*TDBD+32.
C
C*****************************************************************************
C*** CALCULATE THE DENSITY OF THE AMBIENT AIR                             **
C*****************************************************************************
C
      RHOA = 1.0/SPV
C
C*****************************************************************************
C***  BEGIN LOOP                                                          **
C*****************************************************************************
C
      DO 10 I=1,NP
C
C*****************************************************************************
C*** CALCULATE THE DENSITY OF AIR IN THE DUCT OR CHAMBER (SEC. 9.2.2)     **
C*****************************************************************************
```

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```
C
      RHOX= AIRDEN(RHOA,TDBA,TDBD,SP(I),PB)
C
C*******************************************************************************
C*** CALCULATE ALPHA RATIO                                                   **
C***  ALPHA RATIO IS THE RATIO OF ABSOLUTE NOZZLE EXIT PRESSURE              **
C***   TO ABSOLUTE APPROACH PRESSURE (SEC. 9.3.2.1)                          **
C*******************************************************************************
C
      A= ALPHA(PD(I),TDBD,RHOX,R)


C*******************************************************************************
C*** CALCULATE BETA RATIO (SEC. 9.3.2.2)                                    ***
C*******************************************************************************

      B= BETA(D6,D4,NCODE)
C
C*******************************************************************************
C*** CALCULATE EXPANSION FACTOR (SEC. 9.3.2.3)                              ***
C*******************************************************************************
C
      Y= YFACT(GAMMA,A,B)
C
C*******************************************************************************
C*** CALCULATE ENERGY FACTOR (SEC. 9.3.2.4)                                 ***
C*******************************************************************************
C
      E=EN(NCODE)
C
C*******************************************************************************
C***  CALCULATE AIR VISCOSITY(LBM/FT-S) (SEC. 9.2.4)                         **
C*******************************************************************************
C
      AMU=(11.00+0.018*TDBD)*1.E-6
C
C*******************************************************************************
C*** CALCULATE REYNOLDS NUMBER (SEC. 9.3.2.5)                                **
C*** CALCULATE DISCHARGE COEFFICIENT (SEC. 9.3.2.6)                          **
C*******************************************************************************
C
C*******************************************************************************
C***  MUST USE AN ITERATIVE PROCEDURE TO FIND REYNOLDS NUMBER AND            **
C***   DISCHARGE COEFFICIENT                                                 **
C*** CALCULATE DISCHARGE COEFFICIENT USING FIRST APPROXIMATION OF            **
C***    REYNOLDS NUMBER                                                      **
C*******************************************************************************
C
      ARG=PD(I)*RHOX/(1.-E*B**4)
      RE=1.363E6*D6*SQRT(ARG)
      NCOUNT=0
  201 ARG=PD(I)*RHOX/(1.-E*B**4)
```

```
      C1=60.*AMU*RE/(1096.*D6*Y*SQRT(ARG))
      C2=0.9986-7.006/SQRT(RE)+134.6/RE
c     WRITE(MOUT,50) NCOUNT,C1,C2,RE
c  50 FORMAT(2X,'NCOUNT =',I4,' C1=',F7.4,' C2=',F11.8,' RE=',E13.6)
      IF(NCOUNT.GT.100) GO TO 200
      IF(ABS(C1-C2).LT.0.001) GO TO 200
      NCOUNT=NCOUNT+1
      RE=RE+0.001E5
      GO TO 201
  200 CONTINUE
      C=C2
      RE=REY(C,D6,AMU,Y,PD(I),RHOX,E,B)
C
C*********************************************************************************
C*** CALCUATE FLOW RATE FOR A DUCTED NOZZLE (SEC. 9.3.2.7)                    ***
C*********************************************************************************
C
      IF(NCODE) 20,20,25
   20 Q=QFOUR(C,A6,Y,PD(I),RHOX,E,B)
      GO TO 30
C
C*********************************************************************************
C*** CALCULATE THE FLOW RATE FOR A CHAMBER NOZZE (SEC. 9.3.2.8)               ***
C*********************************************************************************
C
   25 Q = QFIVE(Y,PD(I),RHOX,C,A6)
C
C*********************************************************************************
C*** CALCULATE THE FAN FLOW RATE (SEC. 9.3.2.9)                               ***
C*********************************************************************************

   30 CONTINUE
      QN(I)=Q
      QF(I) = Q*(RHOX/RHOA)
      QFF=QF(I)*0.02832
C*********************************************************************************
C***        PRINT OUT THE NECESSARY OUTPUT          ******************************
C*********************************************************************************
C
      WRITE(MOUT,54) A,B,GAMMA,PB
   54 FORMAT(80('*'),/,' ALPHA FACTOR = ',F7.4,/,' BETA FACTOR = ',F4.2,
     */,' GAMMA FACTOR = ',F4.2,/,' BAROMETRIC PRESSURE(IN. HG) =',F6.2)
      WRITE(MOUT,55) RHOA,Y,E,AMU
   55 FORMAT(' AIR DENSITY(LBM/FT**3) = ',F8.5,/,' EXPANSION FACTOR = ',
     *F8.4,/,' ENERGY FACTOR = ',F8.4,/,' AIR VISCOSITY(LBM FT-S) = ',E1
     *3.6)
      WRITE(MOUT,56) RE,C
   56 FORMAT(' REYNOLDS NUMBER = ',E13.6,/,' DISCHARGE COEFFICIENT = ',F
     *8.4)
      WRITE(MOUT,57) TDBA,TWBA,TDBD
   57 FORMAT(' AMBIENT DRY BULB(DEG F) = ',F5.1,/,' AMBIENT WET BULB(DEG
```

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```
    * F) = ',F5.1,/,' DRY BULB AT NOZZLE INLET(DEG F) = ',F5.1,/)
      WRITE(MOUT,53) PD(I),SP(I),QN(I),QF(I),QFF.FS(I)
   53 FORMAT(' PRESSURE DROP ACROSS NOZZLE(IN.W.C.) = ',F5.3,/,' DUCT ST
     *ATIC RESSURE(IN.W.C.) = ',F7.2,/,' NOZZLE AIRFLOW(CFM) = ',F7.2,/.
     *' FAN AIRFLOW(CFM) = ',F7.2,/,'FAN AIRFLOW(CUBIC M/MIN)=',F7.2,/,'
     * FAN SPEED(RPM) = ',I4,/,80('*'),/)

   10 CONTINUE
      print*,'YOUR OUTPUT IS IN FILE FANCAL.OUT'
      STOP
      END
C
C*******************************************************************************
C*** FUNCTION THERM CALCULATES TEMPERATURE FROM K-OHM MEASUREMENTS   **
C*******************************************************************************
C
      FUNCTION THERM(R)
       DOUBLE PRECISION A,B,C,D,E
       A=-2077.423
       B=4681.344
       C=498.4451
       D=21.7356
       E=100.1315
       S=-1.0
       ARG=A+B*R+C*R**2
       IF(ARG.GT.0.0) GO TO 100
       THERM=100.0
       GO TO 200
  100 THERM=D*R+E+S*SQRT(ARG)
  200 RETURN
      END
C
C*******************************************************************************
C*** DUCT OR CHAMMBER AIR DENSITY (SEC. 9.2.2)                           ***
C*******************************************************************************
C***                                                                    ***
C*** PURPOSE:                                                           ***
C***  CALCULATES THE DENSITY OF AIR AT PLANE-X. IF THE STATIC          ***
C*** PRESSURE AT PLANE-X IS LESS THAN 4 IN. WATER, THEN RHX=RHOA.       ***
C***                                                                    ***
C*******************************************************************************
C***                                                                    ***
C***  INPUT VARIABLES:                                                  ***
C***  1. RHOA = DENSITY OF AMBIENT AIR(LB/CU. FT.)                      ***
C***  2. TDO = DRY-BULB TEMPERATURE OF AMBIENT AIR, DEGREES             ***
C***     FARENHEIT.                                                     ***
C***  3. TDX = DRY-BULB TEMPERATURE AT PLANE-X, DEGREES FARENHEIT.      ***
C***     PLANE-X IS PLANE-4 FOR A DUCT APPROACH, OR PLANE-5 FOR A       ***
C***     CHAMBER APPROACH.                                              ***
C***  4. PSX = FAN STATIC PRESSURE, IN. W.G.                            ***
C***  5. RHOB = BAROMETRIC PRESSURE, IN. HG                             ***
```

```
C***                                                                          ***
C*****************************************************************************
C***                                                                          ***
C***  RETURNED VARIABLE:                                                       ***
C***  1. RHOX = AIR DENSITY AT PLANE X(LB/FT**3)                               ***
C***                                                                          ***
C*****************************************************************************

      FUNCTION AIRDEN(RHOA,TDO,TDX,PSX,PB)
        F1= (TDO+459.7)/(TDX+459.7)
        F2= (PSX+13.63*PB)/(13.63*PB)
        AIRDEN= RHOA*F1*F2
      RETURN
      END
C
C*****************************************************************************
C*** ALPHA RATIO (SEC. 9.3.2.1)                                                ***
C*****************************************************************************
C***                                                                          ***
C*** THIS FACTOR IS THE RATIO OF ABSOLUTE NOZZLE EXIT PRESSURE TO              ***
C*** ABSOLUTE APPROACH PRESSURE.                                              ***
C***                                                                          ***
C*****************************************************************************
C***                                                                          ***
C***  INPUT VARIABLES:                                                         ***
C***  1. DELP = PRESSURE DIFFERENTIAL ACROSS THE NOZZLE, IN. WATER.            ***
C***  2. TDX = DRY-BULB TEMPERATURE IN PLANE -X, DEGREES FARENHEIT.            ***
C***      PLANE-X IS PLANE-4 FOR DUCT APPROACH, OR PLANE-5 FOR                 ***
C***      CHAMBER APPROACH.                                                    ***
C***  3. RHX = FAN AIR DENSITY(LB/CU.FT.)                                      ***
C***  4. R = GAS CONSTANT, 53.35 (FT-LB)/(LBM DEG R)  FOR AIR                  ***
C***                                                                          ***
C*****************************************************************************
C***                                                                          ***
C***  RETURNED VARIABLE:                                                       ***
C***  1. ALPHA = DIMENSIONLESS.                                                ***
C***                                                                          ***
C*****************************************************************************

      FUNCTION ALPHA(DELP,TDX,RHX,R)
        F1=5.187*DELP
        F2= RHX*R*(TDX+459.7)
        ALPHA= 1.- F1/F2
      RETURN
      END
C
C*****************************************************************************
C*** BETA RATIO (SEC. 9.3.2.2)                                                 ***
C***                                                                          ***
C*****************************************************************************
C***                                                                          ***
```

```
C*** INPUT VARIABLES:                                                 ***
C*** 1. D6 = DIAMETER OF NOZZLE DISCHARGE STATION,FT.                 ***
C*** 2. D4 = DIAMETER OF DUCT PIEZOMETER STATION,FT.                  ***
C*** 3. NCODE = NOZZLE CODE INDICATING DUCT APPROACH OR CHAMBER       ***
C***    APPROACH.                                                     ***
C***       A. NCODE = 0 : DUCT APPROACH                               ***
C***       B. NCODE = 1 : CHAMBER APPROACH                            ***
C***                                                                  ***
C*********************************************************************** 
C***                                                                  ***
C*** RETURNED VARIABLE:                                               ***
C*** 1. BETA = DIMENSIONLESS                                          ***
C***                                                                  ***
C***                                                                  ***
C*********************************************************************** 
C
      FUNCTION BETA(D6,D4,NCODE)
      IF (NCODE) 2,2,3
    2 DX=D4
      BETA=D6/DX
      RETURN
    3 BETA=0
      RETURN
      END
C
C*********************************************************************** 
C*** EXPANSION FACTOR (SEC. 9.3.2.3)                                  ***
C***                                                                  ***
C*********************************************************************** 
C***                                                                  ***
C*** INPUT VARIABLES:                                                 ***
C*** 1. GAMMA = RATIO OF SPECIFIC HEATS, DIMENSIONLESS.               ***
C*** 2. ALPHA = RATIO OF ABSOLUTE NOZZLE EXIT PRESSURE TO             ***
C***    ABSOLUTE NOZZLE APPROACH PRESSURE, DIMENSIONLESS.             ***
C*** 3. BETA = RATIO OF NOZZLE EXIT DIAMETER TO APPROACH DUCT         ***
C***    DIAMETER, FEET.                                               ***
C***                                                                  ***
C*********************************************************************** 
C***                                                                  ***
C*** RETURNED VARIABLES :                                             ***
C*** 1. Y = EXPANSION FACTOR, DIMENSIONLESS.                          ***
C***                                                                  ***
C*********************************************************************** 
C
      FUNCTION YFACT(GAMMA,ALPHA,BETA)
      F1 = GAMMA/(GAMMA-1.)
      F2 = ALPHA **(2./GAMMA)
      F3 = 1. - ALPHA**((GAMMA-1.)/GAMMA)
      F4 = 1. - ALPHA
      F5 = SQRT(F1*F2*F3/F4)
      F1 = 1. - BETA**4
```

```
      F2 = 1. - (BETA**4) * (ALPHA**(2./GAMMA))
      F3 = SQRT(F1/F2)
      YFACT = F3*F5
      RETURN
      END
C
C*****************************************************************************
C*** ENERGY FACTOR (SEC. 9.3.2.4)                                         ***
C***                                                                      ***
C*** DETERMINE THE VELOCITY PRESSURE UPSTREAM OF THE NOZZLE AT THE        ***
C*** STANDARD TRAVERSE STATIONS. SUFFICIENT ACCURACY FOR THIS             ***
C*** STANDARD CAN BE OBTAINED FOR SETUPS QUALIFYING UNDER THIS            ***
C*** STANDARD BY SETTING E=1.0 FOR CHAMBER APPROACH, E=1.043 FOR          ***
C*** DUCT APPROACH.                                                       ***
C*****************************************************************************
C***                                                                      ***
C*** INPUT VARIABLES:                                                     ***
C*** 1. NCODE = NOZZLE CODE INDICATING DUCT APPROACH OR CHAMBER           ***
C***    APPROACH.                                                         ***
C***        A. NCODE = 0 : DUCT APPROACH                                  ***
C***        B. NCODE = 1 : CHAMBER APPROACH                               ***
C***                                                                      ***
C*****************************************************************************
C***                                                                      ***
C*** RETURNED VARIABLE:                                                   ***
C*** 1. EN = ENERGY FACTOR, DIMENSIONLESS                                 ***
C***                                                                      ***
C*****************************************************************************
C
      FUNCTION EN(NCODE)
      IF (NCODE)3,3,4
    3 EN = 1.043
      RETURN
    4 EN = 1.0
      RETURN
      END
C
C*****************************************************************************
C*** REYNOLDS NUMBER (SEC. 9.3.2.5)              ⌐                        ***
C*****************************************************************************
C*** PURPOSE:                                                             ***
C*** AN APPROXIMATION FOR THE REYNOLDS NUMBER IS DESCRIBED BY THIS        ***
C*** FUNCTION.                                                            ***
C*****************************************************************************
C***                                                                      ***
C*** INPUT VARIABLES:                                                     ***
C*** 1. C = DISCHARGE COEFFICIENT,DIMENSIONLESS.                          ***
C*** 2. D6 = DIAMETER OF THE DUCT AT STATION 6, FT.                       ***
C*** 3. VIS6 = VISCOSITY OF THE AIR AT STATION 6, LB/(FT-S).              ***
C*** 4. Y = EXPANSION FACTOR, DIMENSIONLESS.                              ***
C*** 5. PD = PRESSURE DROP ACROSS THE NOZZLE, IN. WATER.                  ***
```

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```
C*** 6. RHOX = DENSITY OF THE AIR AT STATION X, LB/CU.FT.          ***
C*** 7. E = ENERGY FACTOR, DIMENSIONLESS.                          ***
C*** 8. BETA = BETA RATIO, DIMENSIONLESS.                          ***
C***                                                               ***
C******************************************************************************
C***                                                               ***
C*** RETURNED VARIABLE :                                           ***
C*** 1. REY = REYNOLDS NUMBER, DIMENSIONLESS.                      ***
C***                                                               ***
C******************************************************************************

    FUNCTION REY(C,D6,VIS6,Y,PD,RHOX,E,BETA)
      F1 = 1096./(60.*VIS6)
      F2 = C*D6*Y
      F3 = (PD*RHOX)/(1.-E*BETA**4)
      REY = F1*F2*SQRT(F3)
    RETURN
    END
C
C******************************************************************************
C*** FLOW RATE FOR A DUCTED NOZZLE ********************************************
C******************************************************************************
C*** PURPOSE:                                                      ***
C*** CALCULATES THE VOLUME FLOW RATE AT THE ENTRANCE OF A DUCTED   ***
C*** NOZZLE.                                                       ***
C***                                                               ***
C******************************************************************************
C***                                                               ***
C*** INPUT VARIABLES:                                              ***
C*** 1. C = DISCHARGE COEFFICIENT, DIMENSIONLESS.                  ***
C*** 2. A6 = THROAT AREA AT STATION 6, SQ. FT.                     ***
C*** 3. Y = EXPANSION FACTOR, DIMENSIONLESS.                       ***
C*** 4. PD = PRESSURE DROP ACROSS NOZZLE, IN. WATER.               ***
C*** 5. RHO4 = AIR DENSITY AT STATION 4, LB/CU.FT.                 ***
C*** 6. E ENERGY FACTOR, DIMENSIONLESS.                            ***
C*** 7. BETA = BETA FACTOR, DIMENSIONLESS.                         ***
C***                                                               ***
C******************************************************************************
C***                                                               ***
C*** RETURNED VARIABLE:                                            ***
C*** 1. QFOUR = VOLUME FLOW RATE AT THE ENTRANCE OF THE DUCTED     ***
C***    NOZZLE, CFM.                                               ***
C***                                                               ***
C******************************************************************************

    FUNCTION QFOUR(C,A6,Y,PD,RHO4,E,BETA)
      F1 = 1096.*C*A6*Y
      F2 = SQRT(PD/RHO4)
      F3 = SQRT(1.-E*BETA**4)
      QFOUR = F1*F2/F3
    RETURN
```

```
      END
C
C****************************************************************************
C*** FLOW RATE FOR A CHAMBER NOZZLE ***************************************
C****************************************************************************
C*** PURPOSE:                                                           ***
C*** 1. Y = EXPANSION COEFFICIENT, DIMENSIONLESS.                       ***
C*** 2. PD = PRESSURE DROP ACROSS NOZZLE, IN. WATER.                    ***
C*** 3. RHO5 = DENSITY OF AIR AT STATION 5, LB/ CU.FT.                  ***
C*** 4. C = DISCHARGE COEFFICIENT, DIMENSIONLESS                        ***
C*** 5. A6 =THROAT AREA AT STATION 6 , SQ.FT.                           ***
C***                                                                    ***
C****************************************************************************
C***                                                                    ***
C*** RETURNED VARIABLE:                                                 ***
C*** 1. QFIVE = VOLUME FLOW RATE AT THE ENTRANCE OF NOZZLE WITH         ***
C***    A CHAMBER APPROACH, CFM.                                        ***
C***                                                                    ***
C****************************************************************************

      FUNCTION QFIVE(Y,PD,RHO5,C,A6)
        F1 = 1096.*Y*C*A6
        F2 = SQRT(PD/RHO5)
        QFIVE = F1*F2
      RETURN
      END
C
C****************************************************************************
C***    PSYC                                                            **
C****************************************************************************
C
C****************************************************************************
C***    SUBROUTINE PSYC CALCULATES THE PSYCHROMETRIC PROPERTIES OF      **
C***    THE DRYING AIR BASED ON THE ASAE DATA SHEET D271.2 IN THE       **
C***    AGRICULTURAL ENGINEERING YEARBOOK.                              **
C****************************************************************************
C***                                                                    **
C***    RECEIVES:                                                       **
C***                                                                    **
C***      1. MODE = FLAG TO SIGNAL WHICH VALUES ARE TO BE EN-           **
C***         TERED.                                                     **
C***                                                                    **
C***          =1 THEN DRY BULB AND DEW POINT TEMPERATURES               **
C***           ARE TO BE INPUT.                                         **
C***                                                                    **
C***          = 2 THEN DRY BULB AND WET BULB TEMPERATURES               **
C***           ARE TO BE INPUT.                                         **
C***                                                                    **
C***          = 3 THEN DRY BULB TEMPERATURE AND HUMIDITY RATIO          **
C***           ARE TO BE INPUT.                                         **
C***                                                                    **
```

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```
C***        2. P1 = ENTERING DRY BULB TEMPERATURE (C).                        **
C***                                                                          **
C***        3. P2 = WET BULB TEMPERATURE (C) OR HUMIDITY RATIO                **
C***            DEPENDING ON THE MODE (MODE).                                  **
C***                                                                          **
C***     RETURNS:                                                            **
C***                                                                          **
C***        1. POUT AN ARRAY CONSISTING OF:                                   **
C***                                                                          **
C***        2. RH = RELATIVE HUMIDITY (DEC).                                  **
C***                                                                          **
C***        3. TDP = DEW POINT TEMPERATURE (C).                               **
C***                                                                          **
C***        4. TWB = WET BULB TEMPERATURE (C).                                **
C***                                                                          **
C***                                                                          **
C***        6. HS = SATURATION PRESSURE (PA).                                 **
C***                                                                          **
C***        7. PV = PARTIAL VAPOR PRESSURE (PA).                              **
C***                                                                          **
C***        8. PS = SATURATION PRESSURE (PA).                                 **
C***                                                                          **
C***        9. SH = ENTHALPY (J/KG).                                          **
C***                                                                          **
C***        10. VSA = SPECIFIC VOLUME (CUBIC M/KG).                           **
C***                                                                          **
C******************************************************************************
C
    SUBROUTINE PSYC   (MODE,P1,P2,PATM,MOUT,POUT)
    LOGICAL ERROR
    DIMENSION POUT(9)
    ERROR=.FALSE.
    TE=P1
    PS=PWS(TE)
    HS=0.6219*PS/(PATM-PS)
    GOTO(10,20,30),MODE
C
C******************************************************************************
C**      MODE 1   TA,TDPA   RH,TDP,TWB,H,HS,PV,PS,SH,VSA                       **
C******************************************************************************
C
C******************************************************************************
C***     IF MODE = 1 THEN P2 = TDP (DEW POINT TEMPERATURE)                     **
C******************************************************************************
C
  10 CONTINUE
    TDPA=P2
    TDP=TDPA
    PV=PWS(TDP)
C
C******************************************************************************
```

```
C***    IF THE PARTIAL VAPOR PRESSURE IS LESS THAN ZERO AN ERROR         **
C***    MESSAGE IS GENERATED.                                            **
C*********************************************************************************
C
      IF (PV.LE.0) GOTO 96
      H=0.6219*PV/(PATM-PV)
      RH=PV/PS
      SH=ENTHAL(H,TDP,TE)
      VSA=287*(TE+273.16)/(PATM-PV)
      GO TO 24
C
C*********************************************************************************
C**     MODE 2 TA,TWBA ---> RH,TDP,TWB,H,HS,PV,SH,VSA                     **
C*********************************************************************************
C
C*********************************************************************************
C***    IF MODE = 2 THEN P2 = TWB (WET BULB TEMPERATURE)                 **
C*********************************************************************************
C
   20 CONTINUE
      TWB=P2
      PSWB=PWS(TWB)
      PV=PVWBDB(TE,TWB,PSWB)
      IF(PV.LE.0.0) GO TO 96
      H=0.6219*PV/(PATM-PV)
      TDP=DEW(PV)-273.16
      RH=PV/PS
      SH=ENTHAL(H,TDP,TE)
      VSA=287*(TE+273.16)/(PATM-PV)
      GO TO 90
C
C*********************************************************************************
C***    MODE 3  TE,HE ---> RH,TDP,TWB,H,HS,PV,PS,SH,VSA                   **
C*********************************************************************************
C
C*********************************************************************************
C***    IF MODE = 3 THEN P2 = H (HUMIDITY RATIO)                         **
C*********************************************************************************
C
   30 CONTINUE
      H=P2
      PV=H*PATM/(H+.6219)
      IF(PV.LE.0) GO TO 96
      TDP=DEW(PV)-273.16
      RH=PV/PS
      SH=ENTHAL(H,TDP,TE)
      VSA=287*(TE+273.16)/(PATM-PV)
C
C*********************************************************************************
C***    AN ITERATIVE PROCEDURE IS USED TO CALCULATE THE WET BULB         **
C***    TEMPERATURE.                                                     **
```

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```
C*******************************************************************************
C
  24 JN=0
     TWB=(TE+2*TDP)/3
  26 CONTINUE
     PSWB=PWS(TWB)
     B=BP(PSWB,PATM,PV,TWB)
     TWB1=TE+(PSWB-PV)/B
C
C*******************************************************************************
C***    A DIFFERENCE OF LESS THAN 0.012 BETWEEN THE PREVIOUS WET            **
C***    BULB TEMPERATURE AND THE ONE BEING CALCULATED MUST BE OB-           **
C***    TAINED TO SATISFY THE CONVERGENCE CRITERIA.                         **
C*******************************************************************************
C
     TWCV=ABS(TWB-TWB1)/TWB
     IF(TWCV.LT..012)GOTO 28
     TWB=(TWB1+2*TWB)/3
C*******************************************************************************
C***    IF THE NUMBER OF ITERATIONS IS GREATER THAN 10 AN ERROR            **
C***    MESSAGE IS GENERATED.                                               **
C*******************************************************************************
C
     JN=JN+1
     IF(JN.LT.10) GOTO 26
     GOTO 96
  28 TWB=TWB1
     GOTO 90
  96 CONTINUE
     ERROR=.TRUE.
  90 CONTINUE
C
C*******************************************************************************
C***    IF THE RELATIVE HUMIDITY IS GREATER THAN 1.00  IT IS               **
C***    SET EQUAL TO 0.999.                                                 **
C*******************************************************************************
C
     IF(RH.GE..999) RH=.999
     POUT(1)=RH
     POUT(2)=TDP
     POUT(3)=TWB
     POUT(4)=H
     POUT(5)=HS
     POUT(6)=PV
     POUT(7)=PS
     POUT(8)=SH
     POUT(9)=VSA
     IF(ERROR) GOTO 98
     RETURN
C
C*******************************************************************************
```

```
C***     ERROR MESSAGE IS GENERATED IF THE PARTIAL VAPOR PRESSURE            **
C***     IS LESS THAN 0 OR THE NUMBER OF ITERATIONS REQUIRED TO              **
C***     CALCULATE THE WET BULB TEMPERATURE IS GREATER THAN 10.              **
C*********************************************************************************
C
   98 WRITE(MOUT,500) MODE,JN,P1,P2
  500 FORMAT(/2X,'**ERROR**PSYC**',2X,'MODE=',I2,2X,'ITER. NO.=',I2,2X,
     $'P1=',F8.4,2X,'P2=',F8.4)
      WRITE(MOUT,501) TWB,TWB1,TWCV
  501 FORMAT(2X,'TWB=',F6.2,2X,'TWB1=',F6.2,2X,'TWCV=',F8.4)
      RETURN
      END
C
C*********************************************************************************
C***      BP                                                                 **
C*********************************************************************************
C
C*********************************************************************************
C***     AN EQUATION REPRESENTING THE WET BULB LINE                          **
C***     (ASAE D271.2).                                                      **
C*********************************************************************************
C
C*********************************************************************************
C***                                                                         **
C***        1. HFGP = LATENT HEAT OF VAPORIZATION AT WET BULB                **
C**            TEMPERATURE (J/KG).                                           **
C***                                                                         **
C***        2. PSWB = SATURATION VAPOR PRESSURE (PA).                        **
C***                                                                         **
C***        3. PATM = ATMOSPHERIC PRESSURE (PA).                             **
C***                                                                         **
C***        4. PV = VAPOR PRESSURE (PA).                                     **
C***                                                                         **
C***        5. TWB = WET BULB TEMPERATURE (C).                               **
C***                                                                         **
C***        6. BP = WET BULB LINE ON PSYCHROMETRIC CHART.                    **
C***                                                                         **
C*********************************************************************************
C
      FUNCTION BP(PSWB,PATM,PV,TWB)
      B1=1006.9254*(PSWB-PATM)*(1.+0.15577*PV/PATM)
      HFGP=2.502553E6-2.385764E3*TWB
      BP=B1/(0.62194*HFGP)
      RETURN
      END
C
C*********************************************************************************
C**       DEW                                                                **
C*********************************************************************************
C
C*********************************************************************************
```

```
C***    DEW POINT TEMPERATURE CALCULATED (ASAE D271.2)                        **
C******************************************************************************
C
C******************************************************************************
C***                                                                          **
C***   1. PS = SATURATION VAPOR PRESSURE (PA).                                **
C***                                                                          **
C***   2. DEW = DEW POINT TEMPERATURE (K).                                    **
C***                                                                          **
C******************************************************************************
C
      FUNCTION DEW(PS)
      X=ALOG(0.00145*PS)
      ARG1=13.6626*X+1.17678*X**2-0.189693*X**3
      ARG2=0.087453*X**4-0.0174053*X**5+2.14768E-3*X**6
      ARG3=-0.138343E-3*X**7+0.38E-5*X**8
      DEW=255.38+19.5322+ARG1+ARG2+ARG3
      RETURN
      END
C
C******************************************************************************
C***         ENTHAL                                                           **
C******************************************************************************
C
C******************************************************************************
C***    ENTHALPY OF AIR CALCULATED (ASAE D271.2)                              **
C******************************************************************************
C
C******************************************************************************
C***                                                                          **
C***        1. HFGDP = LATENT HEAT OF VAPORIZATION AT DEW POINT               **
C***           TEMPERATURE (J/KG).                                            **
C***                                                                          **
C***        2. HR = HUMIDITY RATIO (DEC).                                     **
C***                                                                          **
C***        3. TDP = DEW POINT TEMPERATURE (C).                               **
C***                                                                          **
C***        4. TDB = DRY BULB TEMPERATURE (C).                                **
C***                                                                          **
C***        5. ENTHALPY = ENTHALPY (J/KG).                                    **
C***                                                                          **
C******************************************************************************
C
      FUNCTION ENTHAL(HR,TDP,TDB)
      HFG2P=2.502535E6-2.385764E3*TDP
      ARG1=1.006925E3*TDB+4.1868E3*HR*TDP+HFG2P*HR
      ENTHAL=ARG1+1.875686E3*HR*(TDB-TDP)
      RETURN
      END
C
C******************************************************************************
```
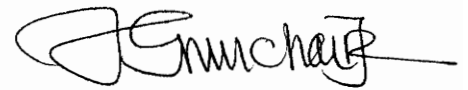
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```
C*** PVWBDB                                                                    **
C*******************************************************************************
C
C*******************************************************************************
C***    PARTIAL VAPOR PRESSURE OF AIR IS CALCULATED (ASAE D271.2)              **
C*******************************************************************************
C
C*******************************************************************************
C***                                                                          **
C***        1. HFGP = LATENT HEAT OF VAPORIZATION (J/KG).                      **
C***                                                                          **
C***        2. TDB = DRY BULB TEMPERATURE (C).                                **
C***                                                                          **
C***        3. TWB = WET BULB TEMPERATURE (C).                                **
C***                                                                          **
C***        4. PSWB = SATURATION VAPOR PRESSURE (PA).                         **
C***                                                                          **
C***        5. PVWBDB = PARTIAL VAPOR PRESSURE OF AIR (PA).                   **
C***                                                                          **
C*******************************************************************************
C
    FUNCTION PVWBDB(TDB,TWB,PSWB)
    HFGP=2.502553E6-2.385764E3*TWB
    ARG1=0.62194*HFGP*PSWB-1.006925E3*(PSWB-101325.0)*(TWB-TDB)
    ARG2=0.62194*HFGP+156.8488*(PSWB-101325.)*(TWB-TDB)/101325.
    PVWBDB=ARG1/ARG2
    RETURN
    END
C
C*******************************************************************************
C***    PWS                                                                    **
C*******************************************************************************
C
C*******************************************************************************
C***    SATURATION VAPOR PRESSURE IS CALCULATED (ASAE D271.2)                  **
C
C*******************************************************************************
C***                                                                          **
C***        1. TC = DRY BULB AIR TEMPERATURE (C).                             **
C***                                                                          **
C***        1. PWS = SATURATION VAPOR PRESSURE (PA).                         **
C***                                                                          **
C*******************************************************************************
    FUNCTION PWS(TC)
    T=TC+273.16
    ARG1=-27405.53+97.5413*T-0.146244*T**2
    ARG2=0.12558E-3*T**3-0.48502E-7*T**4
    ARG3=4.34903*T-0.39381E-2*T**2
    PWS=22105650.*EXP((ARG1+ARG2)/ARG3)
    RETURN
    END
```

VITA

The author, Jesse G. Muchai, was born in Kiambu, Kenya on October 12, 1961. After graduating in 1980 from Chewoyet High School (Kapenguria, Kenya) he began his technical training at the then Jomo Kenyatta College of Agriculture and Technology (Nairobi, Kenya) in Agricultural Machinery Technology. In June, 1986 he obtained a technician Level III certificate. In 1989, he was admitted at Auburn University and in December, 1992 he obtained a Bachelor of Science degree in Mechanical Engineering. In January, 1992 he was admitted to Virginia Polytechnic Institute and State University to work towards a Master's of Science degree in Biological Systems Engineering.