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This report presents the study of wood combustion and combustion systems for a combined cycle wood-fired power plant. The power plant consists of a twin air cycle and a single steam cycle. The fuel for this power plant is wood and wood waste with particle size in the range of three inch minus and a moisture content of 35 to 60 percent wet basis. The distinguishing features of this power plant are: direct combustion of wood, indirect-fired gas turbine, and combined cycle with a gas turbine cycle and a Rankine steam cycle. The three alternative designs of this power plant were considered. System 2 is designed for dry fuel combustion, with a dryer utilized. System 5 is designed for wet fuel combustion; therefore, the dryer of system 2 is not used. In system 7, which is designed for wet fuel combustion, a trimburner is used to increase the inlet temperature of the gas turbine from 1450 F to 1612 F.

The wood fuels have a wide range of particle size and moisture content which affect the combustion of the fuel. In this report the combustion of wood and the effects of variables such as moisture content, particle size, and excess air on combustion of wood have been analyzed, and the results of the analysis are presented.

Because of the variety of wood combustion systems available, in this work several combustion systems have been reviewed with the purpose of recommending a combustion system for the use in this power plant.

An evaluation of wood combustion based on the principles of conservation of mass, the first law of thermodynamics, the second law of thermodynamics and a prescribed extent of completion is presented. A computer program based on this evaluation is provided. The results of analysis based on this model and the computer simulation of the power plant are presented. Recommendations for improved operating conditions of the combustor, selection of the combustion system, and improvements in power plant design and operation are made. Based on the analysis presented in this work, it is concluded that system 7 is the most efficient system among the three systems considered.

Analysis of Wood Combustion and Combustion Systems for a Combined Cycle Wood-Fired Power Plant

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Chapter 1

INTRODUCTION

Until the late nineteenth century, wood and wood-base fuels have been the principal source of energy utilized by man. In this century; however, the discovery of fossil fuel and its low cost have lead to a steady decline in the use of wood as an energy source (although in many countries wood is still a major source of energy in rural areas). The energy crunch of the 1970's and the ever-increasing cost of fossil fuel have renewed the interest in efficient utilization of biomass-fuels, particularly wood and wood waste. Today, the forest industry supplies the U.S. energy market with 1.7 quadrillion Btu (quads) of energy per year [Tillman, 1983].

Wood is a highly flexible fuel which can be used in a variety of ways: such as space heating, steam generation, gasification, liquid fuel production, and electricity generation. Wood was even used to power automobiles and buses in Finland during World War II [Shafizadeh, 1977]. Compared to coal, wood has very little ash, nitrogen, and sulfur content which makes it a very desirable fuel from the standpoint of air pollution control.

In recent years there have been many studies on efficient and competitive ways of utilization of wood to produce other forms of energy, such as improved processes to produce liquid or gaseous fuels, and more efficient power generation cycles.

Oregon State University, under contract with the United States Department of Agriculture, is conducting a feasibility study of a combined cycle biomass-powered power generating facility that may be relocated from one resource site to another. This project is being conducted by a team of engineering and forestry people. The overall project covers an evaluation of the resource site and the power plant system design.

The objectives of this work are to present a study of wood combustion and the combustion systems available for wood fuels, both in general and for this particular power plant.

In this chapter, brief descriptions of the biomass power plant and its alternative designs are given. Chapter two gives a review of the wood fuel characteristics and properties. In chapter three combustion of wood is discussed and in chapter four combustion systems used for wood combustion are reviewed. In chapter five a method of calculation for wood combustion is given.

Chapter six presents the results and conclusions of this study.

Description of the Power Plant

The plant consists of a "twin system" air cycle and a single steam cycle. The biomass fuel is assumed to be hogged wood with the size of the fuel particles in the range of 3 inch minus and variable moisture content (30 to 60 percent wet basis). The power plant is to have an output of about 10 MW at design conditions.

The three distinguishing features of this plant are:

(1) The wood undergoes direct combustion.

(2) The gas turbine system is of the indirect-fired type.

(3) The system is of the combined-cycle type with a gas turbine cycle and a steam cycle.

Eleven alternative designs (systems 1 through 11) for this power plant have been considered. The detailed description and the results of thermal analysis of these systems are given by Bushnell et al. [1984 and 1985].

Since the objective of the work presented here is to analyze the combustion process of wood and combustion systems available for wood fuel, only three of the eleven systems have been chosen for study in this work. These are systems 2, 5, and 7 which are described in the next section.

As mentioned earlier, the plant consists of a twin gas cycle and a single steam cycle. The first section of the power plant consists of two parallel streams of fuel and air, requiring two fuel preparation sections, two combustors, two gas heat exchangers, and two gas turbine-generator sets. The two streams of hot gases exiting the gas heat exchangers are then combined and fed to a single heat recovery steam generator to produce steam which is then expanded through a single steam turbine. In order to simplify the modeling and analysis of the systems, only one of the two twin systems and half of the steam system are modeled.

In the next section brief descriptions of the power plant and the three alternative designs, mentioned above, are given.

System 2

This system is shown in figure 1.1. Table 1.1 presents a listing of the components shown in figure 1.1. In this system the fuel comes from a fuel storage area and is fed into a hammer hog for size reduction to 1/2 inch minus. The fuel is then dried in a triple pass rotary drum dryer, which uses hot (about 454 F) gases exhausted from the heat recovery steam generator. The dry fuel particle size is still further reduced (according to the

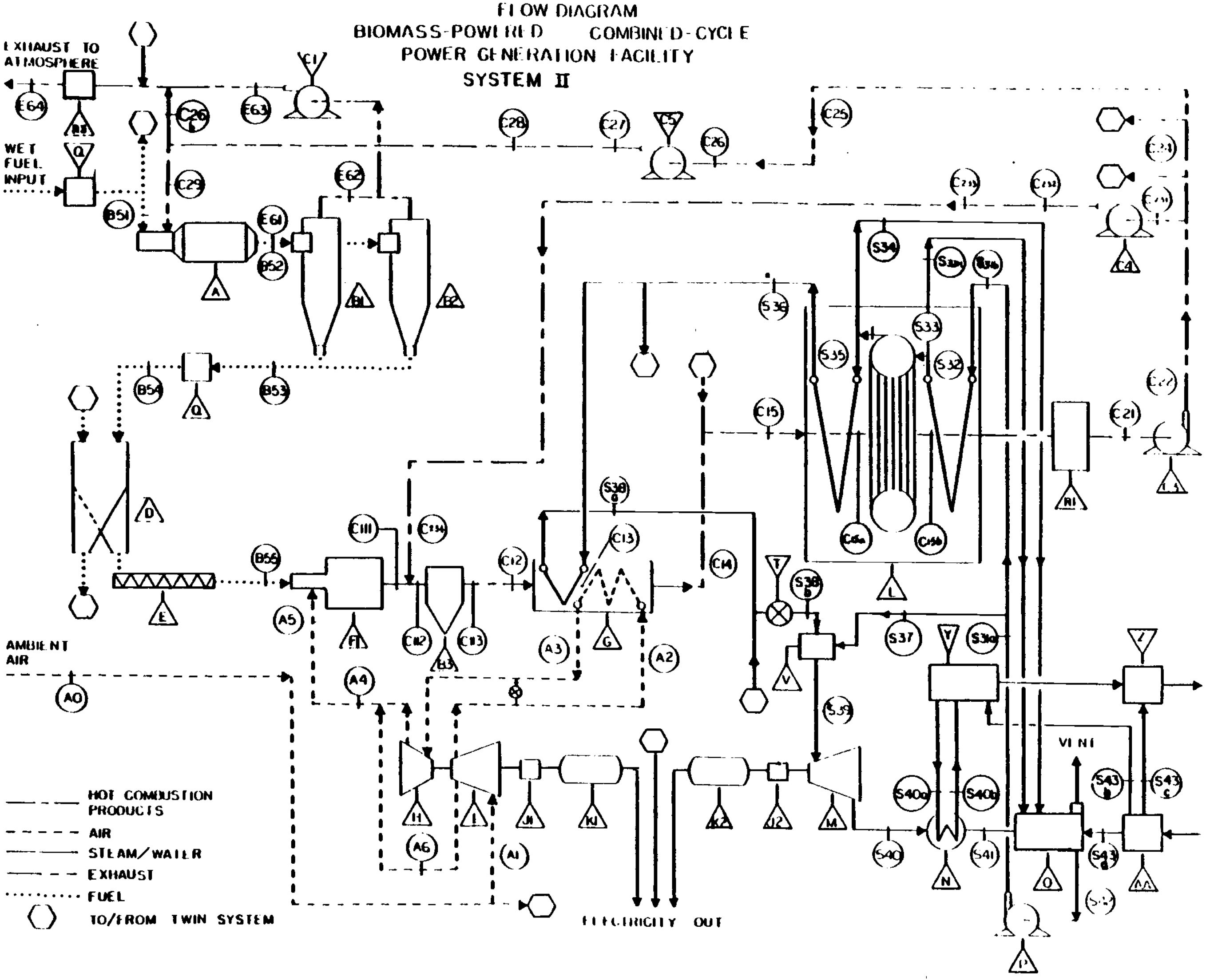

Figure 1.1. Schematic diagram of system 2.

Table 1.1. System Components for Biomass Plant

Dryer - A

Water Treatment Package - AA

Cyclones - B1, B2, and B3

Fans - C1, C3, C4, and C5

Storage Bins - D

Screw Feeders - E

Combustor - F1

Heat Exchangers - G

Trimburner - F2

Gas Turbine - H

Compressor - I

Gearboxes - J1 and J2

Generators - K1 and K2

Heat Recovery Steam Generator - L

Throttling Valve - T

Desuperheater - V

Steam Turbine - M

Condenser - N

Deaerator - O

Pump - P

Hammermill - Q

Air pollution devices - R

Cooling Tower - Y

Solid Waste Treatment Package - Z

combustor specifications) and then stored in a short-term fuel storage. The fuel is then fed to the combustor where it is combusted and hot flue gas is generated. The resulting combustion gases heat the compressed air going from the compressor to the gas turbine, superheat the steam in the steam heat exchanger, produce steam in the heat recovery steam generator, and dry the fuel.

Ambient air enters the system through the compressor. After compression it is heated in an air-gas heat exchanger and then expanded through the gas turbine. The air exiting the turbine, which is still somewhat pressurized and at elevated temperature, is fed to the combustor where it is used as combustion air.

In this system the size of the heat recovery steam generator (consisting of economizer, boiler, and superheater) is chosen such that the flue gas exiting the economizer is at least 454 F, in order to accomplish the drying of the fuel in the dryer.

In the steam cycle, 640 psia feedwater is supplied to the economizer, by the feedwater pump, where it is heated to its saturation temperature. Evaporation takes place in the evaporator. The steam produced in the evaporator is then heated to about 700 F in the primary superheater and to about 900 F in the steam-gas heat exchanger before expansion in the steam turbine. The discharged steam, after passing through the condenser and the deaerator, is then pumped back to the heat recovery steam generator.

The main difference of this system with other systems (systems 5 and 7) is that in this system the fuel is predried and is reduced to smaller particle size; therefore, the combustor system for this system is chosen to be of the suspension type.

System 5

System five, shown in figure 1.2, is designed for wet fuel combustion. Therefore, the extensive fuel preparation equipment and the dryer of figure 1.1 are not used. The remaining portions of the system are nearly the same as system 2 with the exception of the fans, air pollution devices, heat recovery steam generator size, heat exchangers size, and the combustor type.

System 7

In system 7, shown in figure 1.3, a trimburner is utilized to increase the temperature of air entering the gas turbine from 1475 F to 1612 F. The trimburner considered is essentially a duct burner which is supplied with number 2 oil for combustion. The rest of the system is the same as system 5.

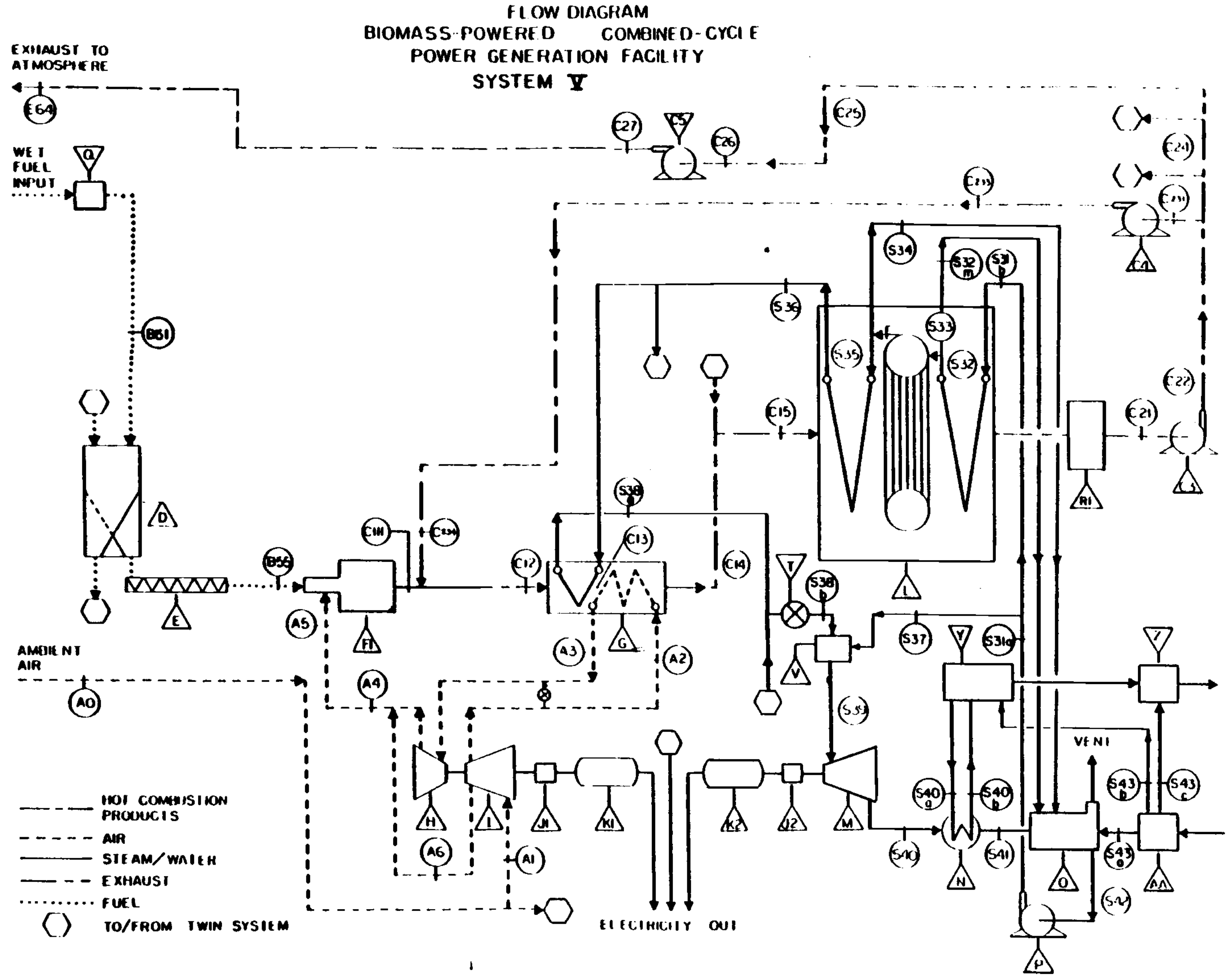

Figure 1.2. Schematic diagram of system 5.

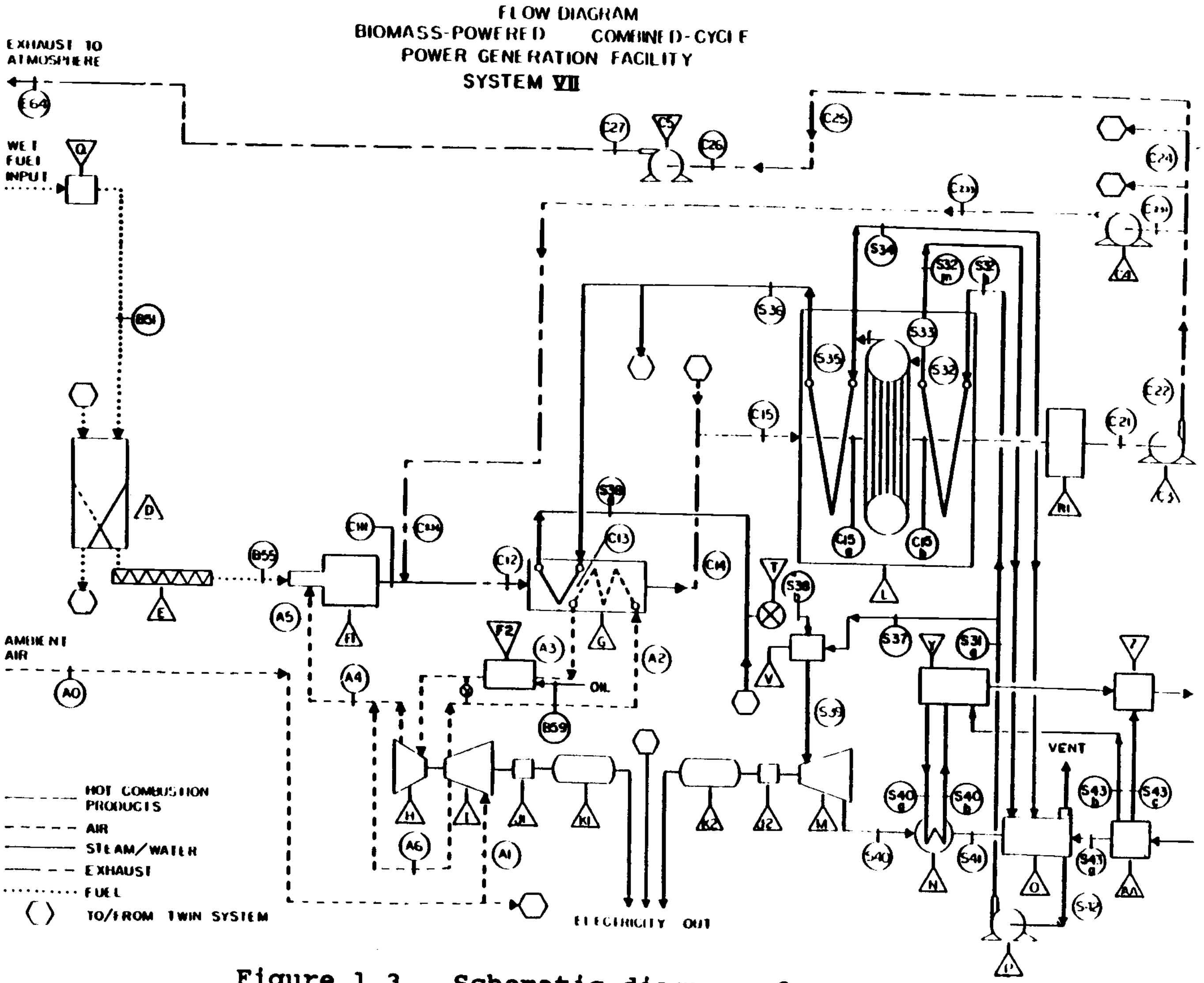

Figure 1.3. Schematic diagram of system 7.

Chapter 2

WOOD FUEL

Wood, in general, is a highly heterogeneous substance, consisting of various types of dead and living plant cells or fibers, with a different structure and composition for each plant species.

Structurally, wood is composed of fibers packed closely together and fairly well-aligned. The bulk of the fibers are made up of holocellulose (combination of cellulose and hemicellulose). Wood contains cellulose, hemicellulose, lignin, extractives, mineral compounds and water. Compared to wood, bark contains much more extractives and lignin, or phenolic compounds, and less holocellulose than wood.

Cellulose

Cellulose is the principle constituent of the cell wall and is the main source of its mechanical and hygroscopic properties [Siau, 1971]. The cellulose component, which is macro-molecular, is the same in all types of wood except for the degree of polymerization. Cellulose has been shown to have an empirical formula somewhat like $(C_6H_{10}O_5)_n$, which is isomeric with starch. The molecular weight of cellulose is about 10^6. Cellulose has a degree of polymerization of about 10000 [Kanury, 1970].

Hemicellulose

Hemicelluloses in the cell wall have a similar composition to cellulose, but have a quite a different molecular configuration. The hemicelluloses are amorphous and have a lower degree of polymerization. They are low-molecular-weight polysacharides, which may preferentially be removed from original or delignified material by aqueous alkali or water.

Lignin

Lignin is a randomly linked amorphous polymer, consisting of phenyl-propane units, which is more abundant and polymeric in softwoods than in hardwoods. Lignin has a lower molecular weight than cellulose (about 1000), and its elementary composition is found to be $C_{47}H_{52}O_{16}$ or $C_{42}H_{32}O_6(OH)_5(CH_3O)_5$ [Kanury, 1970].

Extraneous Components

The term "extraneous components" includes a vast number of organic compounds present in some wood and absent from others. Most of these can be removed by means of neutral solvents, such as ether, benzene, water or steam, and are not a part of the cell wall. Frequently these extraneous substances are also termed extractives, (although in a number of instances their complete extraction is extremely difficult and at times impossible). Extraneous materials, such as resins, play important roles which are out of proportion to their amount. Extraneous components affect wood properties such as durability, color, odor, inflammability, and heating value of the wood [Wise, 1952].

Proximate Analysis

Proximate analysis is a standard method of classification of solid fuels. Proximate analysis describes the content of a solid fuel in terms of volatile matter and ash by direct determination and fixed carbon by difference as a percentage of dry weight of the fuel. ASTM Test No. D-271-48 describes the procedure and equipment for conducting the test [Mingle, 1968].

The fixed carbon, as determined by the proximate analysis, is not the same as the mass fraction of carbon in the fuel which is determined by ultimate analysis of the fuel.

In general, for wood, volatile matter accounts for about 70 to 90 percent of the weight of dry wood. Bark has about 10 to 15 percent less weight in volatile matter as compared to wood of the same species. The difference corresponds to the larger amount of fixed carbon in bark.

Wood and bark contain much higher proportions of volatile matter than coal.

During the combustion, volatile materials burn in the gaseous phase with flaming combustion; whereas, fixed carbon burns in the solid phase with glowing combustion at a much slower rate than flaming combustion.

The proximate analysis of wood and bark for a number of species are presented in tables 2.1.a and 2.1.b.

TABLE 2.1.a. Proximate Analysis of Wood
(percent dry weight)
Source : [Junge, 1980]

SPECIES	VOLATILE MATTER	FIXED CARBON	ASH
Cedar	77.0	21.0	2.0
Douglas Fir	86.2	13.7	0.1
White Fir	84.4	15.1	0.5
Easter Hemlock	72.0	25.5	2.5
Western Hemlock	84.8	15.0	0.2
Maple	76.1	19.6	4.3
Jack Pine	74.3	23.6	2.1
Ponderosa Pine	87.0	12.8	0.2
Redwood	83.5	16.1	0.4

Table 2.1.b. Proximate Analysis of Bark
(percent dry weight)
Source : [Junge, 1980]

SPECIES	VOLATILES	FIXED CARBON	ASH
SOFTWOOD			
Cedar	87.7	13.1	0.2
Douglas Fir	70.6	27.2	2.2
Balsam Fir	77.4	20.0	2.6
Grand Fir	74.9	22.6	2.5
White Fir	73.4	24.0	2.6
Eastern Hemlock	72.0	25.5	2.5
Western Hemlock	74.3	24.0	1.7
Jack Pine	74.3	23.6	2.1
Ponderosa Pine	73.4	25.9	0.7
Southern Pine	72.9	24.2	2.9
Redwood	71.3	27.9	0.8
Black Spruce	74.4	22.5	2.8
Red Spruce	72.9	23.7	3.3
White Spruce	72.5	24.0	3.5
Tamarack	69.5	26.3	4.2
HARDWOOD			
Red Alder	74.3	23.3	2.4
American Beech	75.2	16.9	7.9
Paper Birch	80.3	18.0	1.7
Yellow Birch	76.5	21.0	2.5
American Elm	73.1	18.8	8.1
Red Maple	78.1	18.9	3.0
Sugar Maple	75.1	19.9	5.0
Oak	76.0	18.7	5.3

Ultimate Analysis

The ultimate analysis of a fuel is its elemental composition evaluated from an experimental determination. For wood the ultimate analysis gives the amount of carbon, hydrogen, oxygen, nitrogen, sulfur, and ash present in the fuel as a percentage (or mass fraction) of the dry weight of the fuel.

The ultimate analysis is essential in mass and energy balance calculations of combustion (see Chapter 5).

The ultimate analysis of several species of wood and bark are shown in tables 2.2.a and 2.2.b. As these tables show, wood and bark have extremely low nitrogen, sulfur, and ash content, as compared to coal. This makes wood and bark highly desirable fuels from the standpoint of combustion pollution control.

Table 2.2.a. Ultimate Analysis for Dry Hogged Wood Fuel (percent of dry weight)
Source : [Junge, 1980]

SPECIES	HYDROGEN	CARBON	SULFUR	NITROGEN	OXYGEN	ASH
SOFTWOOD						
White Cedar	6.73	48.80	-	-	44.46	0.37
Cypress	6.54	54.98	-	-	38.08	0.40
Douglas Fir	6.3	52.3	-	-	40.50	0.80
W. Hemlock	5.8	50.4	0.1	0.1	41.40	2.20
Pitch Pine	7.19	59.0	-	-	32.68	1.13
White Pine	6.08	52.55	-	-	41.25	0.12
Yellow Pine	7.02	52.6	-	-	40.07	0.31
Redwood	5.9	53.5	-	0.1	40.3	0.2
HARDWOOD						
White Ash	6.93	49.73	-	-	43.04	0.3
Beech	6.26	51.64	-	-	41.54	0.65
White Birch	6.49	49.77	-	-	43.45	0.29
Elm	6.57	50.35	-	-	42.34	0.74
Hickory	6.49	49.67	-	-	43.11	0.73
Maple	6.02	50.64	-	0.25	41.74	1.35
Black Oak	6.09	48.78	-	-	44.98	0.15
Red Oak	6.62	49.49	-	-	43.74	0.15
White Oak	6.59	50.44	-	-	42.73	0.24
Poplar	6.26	51.64	-	-	41.45	0.65

Table 2.2.b. Ultimate Analysis for Hogged Bark Fuel (percent of dry weight) Source : [Junge, 1980]

SPECIES	HYDROGEN	CARBON	SULFUR	OXYGEN*	ASH
SOFTWOOD					
Douglas Fir	6.2	53.0	-	39.3	1.5
Balsam Fir	6.1	52.8	-	38.8	2.3
E. Hemlock	5.8	53.6	-	40.1	2.5
W. Hemlock	5.8	51.2	-	39.3	3.7
Jack Pine	5.9	53.4	-	38.7	2.0
Scots Pine	5.9	54.4	-	38.0	1.7
Black Spruce	5.8	52.0	-	39.8	2.4
Norway Spruce	5.9	50.6	-	40.7	2.8
Red Spruce	5.7	52.1	-	39.1	3.1
White Spruce	6.4	52.4	-	38.2	3.0
Tamarack	5.9	55.2	-	34.7	4.2
HARDWOOD					
American Beech	5.5	47.5	-	39.1	7.9
Paper Birch	6.8	57.4	-	34.1	1.8
Yellow Birch	6.4	54.5	-	36.8	2.3
American Elm	5.3	46.9	-	39.7	8.1
Red Maple	5.9	50.1	-	41.0	3.0
Sugar Maple	5.9	50.4	-	39.6	4.1
Oak	5.4	49.7	0.1	39.5	5.3

* Nitrogen is included with oxygen

Moisture Content

The moisture content of wood and bark is described in two ways: wet basis and dry basis. The Moisture content on wet (or green) basis is defined as the ratio of the weight of water present in the wood to total weight of the wood plus water:

$$M.C.(wb) = \frac{\text{weight of water in wood}}{\text{weight of dry wood + weight of water in wood}}$$

where M.C.(wb) is the fraction of moisture content on a wet basis.

The dry basis moisture content is defined as the ratio of the weight of water present in wood to the weight of dry wood or:

$$M.C.(db) = \frac{\text{weight of water in wood}}{\text{weight of dry wood}}$$

where M.C.(db) is the fraction of moisture content on a dry basis. The moisture content of wood can also be expressed as a percentage.

Conversion from dry basis to wet basis and from wet basis to dry basis is given in the following equations:

$$M.C.(wb) = \frac{M.C.(db)}{M.C.(db) + 1}$$

$$M.C.(db) = \frac{M.C.(wb)}{1 - M.C.(wb)}$$

Throughout this work the moisture content of wood refers to the wet basis unless it is specified otherwise.

The moisture content of wood fuels can vary greatly depending on the species of the wood, manufacturing processes that produce the wood fuel, storage, handling, site, and season of the year. For example, residues generated from dried wood may have a moisture content of less than 10 percent; whereas, the bark from water-stored logs may have a moisture content of up to 75 percent [Howlett, 1977]. Typical seasonal variations of moisture content of wood residues in the Pacific North West are shown in figure 2.1. Table 2.3 shows the range of moisture content of selected biomass fuels.

Table 2.3. Approximate Range of Moisture Content for Typical Wood Fuel Source : [Junge, 1980]

COMPONENT	RANGE OF MOISTURE CONTENT (percent wet basis)
Sanderdust	2 - 8
Shavings	10 - 20
Sawdust	25 - 40
Bark (hogged)	25 - 75
Coarse Wood Residue	30 - 60
Log Yard Cleanup	40 - 60
Forest Residuals	30 - 60

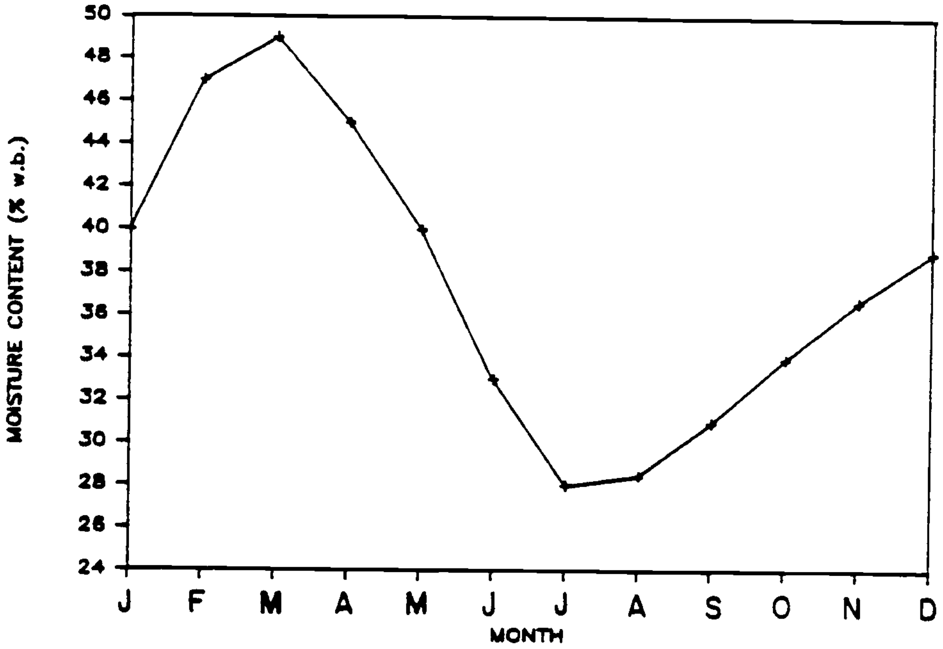

Figure 2.1. Typical yearly variation in wood fuel moisture content [Brawn, 1985] .

Particle Size

Wood fuel particle size depends on its source and the manufacturing process which produces it. It can range from 1/100 inch for very fine sanderdust to 6 inches for coarse hogged bark.

Particle size of wood fuel not only effects the combustion process, but also determines the type of fuel handling equipment and the combustion system. For example, smaller particles like sanderdust are suitable for burning in suspension, but larger particles require grate type burners. Table 2.4 shows the typical range of the hogged fuel particle sizes.

Table 2.4. Approximate Size Range of Typical Components of Wood Fuel
Source : [Junge, 1980]

COMPONENT	SIZE RANGE (inches)
Sanderdust	less than 1/32
Shavings	1/32 - 1/2
Sawdust	1/32 - 3/8
Bark (hogged)	1/32 - 3/8
Coarse Wood Residue	1/32 - 4
Log Yard Cleanup	up to 4
Forest Residuals	needles to stumps

Heating Value

An important property of any fuel is its heating value, which is a measure of the energy released during complete combustion of the fuel at a reference temperature and pressure. There are three different heating values used in this work.

1- Higher Heating Value (HHV) or Gross Heating Value (GHV). Higher heating value is the experimentally determined value of heat released during the combustion per unit weight of the fuel using an oxygen bomb calorimeter (ASTM Standard D2015-77). It is important to note here that in the products of combustion the water formed is in the liquid phase and, therefore, the heat of vaporization for the water is not subtracted from the heating value. In most cases, higher heating value is reported for dry wood. To account for the moisture content of the fuel and the energy loss due to vaporization of water formed during the combustion, lower heating values have been defined.

Although analytical formulas for the calculation of higher heating values have been suggested [Shieh, 1982], it is the experimentally determined value which gives a more accurate result.

2- Lower Heating Value One (LHV1). Lower heating value or net heating value is defined the same as HHV except that here it is assumed that, in the products of combustion, the water "formed" is in vapor phase. Therefore, if the heat of vaporization at reference temperature is equal to Hvap, then the lower heating value one and the higher heating value are related by the equation:

$$LHV1 = HHV - Hvap * W$$

where W is the pound of water formed per pound of fuel. At 60 F and a pressure of one atmosphere, the heat of vaporization of water is equal to 1059 Btu/lb, so LHV1 in Btu per pound of dry fuel is equal to

$$LHV1 = HHV - 1059 * W$$

3- Lower Heating Value Two (LHV2). Since the wood or bark fuels entering the combustion chamber are not dry, they have to be dried before combustion takes place. In this process, a certain amount of energy is used to vaporize the moisture in the wood. Lower heating value two is defined in order to indicate the net heating value of the wood available during the combustion to raise the temperature of the products of combustion.

LHV2 is equal to LHV1 minus the energy required to vaporize the moisture of the wood, that is:

$$LHV2 = LHV1 - Qvap$$

where LHV2 is the lower heating value two in Btu per pound of dry wood and Qvap is the energy required to vaporize the moisture in the wood . The method of calculation of

Qvap is discussed in Chapter 5. Note that in previous reports on this project, LHV2 defined above, is designated as LHV3.

It has been shown that, the higher heating value of different species of wood, on a moisture-free and resin-free basis, are approximately constant and equal to 8300 Btu/lb [Junge, 1980].

Cellulose has a higher heating value of 7460 Btu/lb; whereas, the higher heating value of lignin is about 11470 Btu/lb [Rossi, 1984]. Resin has a higher heating value of about 17000 Btu/lb [Junge, 1980]. Therefore, the difference between the heating values of different species of wood is due to differences in the relative amounts of fiber and resinous material present in the species. Since the higher heating value of resin is much higher than wood fiber, the resinous woods such as pine and Douglas fir have a higher heating value than resin-free woods such as true fir and most hardwoods [Howlett, 1977].

The heating value of bark, in general, is higher than that of wood. Softwood barks have a higher heating values than hardwood barks. Again the difference is probably due to the higher proportion of resin-like compounds and lignin in the bark.

Table 2.5.a, 2.5.b, and 2.5.c present the higher heating value for the dry wood and bark of several species.

Table 2.5.a Higher Heating Value of Wood
(in Btu/lb of dry wood)
Sources :
A : [Junge, 1980]
B : [Rossi, 1984]
C : [Howlett, 1977]

SPECIES	HIGHER HEATING VALUE A	HIGHER HEATING VALUE B	HIGHER HEATING VALUE C
SOFTWOOD			
Western Red Cedar	9700	8840	9700
White Cedar	8400	-	-
Cypress	9870	-	-
Douglas Fir	9050	8760	8890
Easter Hemlock	8885	-	-
Western Hemlock	8620	8550	8410
Pine Sawdust	9130	-	-
Jack Pine (Wood Refuse)	8930	-	-
Loblolly Pine (Stemwood)	8600	-	-
Pitch Pine	11320	-	-
Ponderosa Pine	9100	9020	9110
White Pine	8900	-	-
Yellow Pine	9610	-	-
Redwood	9040	-	-
HARDWOOD			
White Ash	8920	-	-
Beech	8760	-	8150
Birch (Wood Refuse)	8870	-	-
White Birch	8650	-	-
Elm	8810	-	8170
Hickory	8670	-	8050
Maple	8580	-	-
Maple (Wood Refuse)	8190	-	-
Black Oak	8180	8020	-
Red Oak	8690	-	8050
White Oak	8810	-	8150
Poplar	8920	-	-

Table 2.5.b. Higher Heating Value of Bark of Softwoods (in Btu/lb of dry weight)
Sources :
A : [Junge, 1980]
B : [Rossi, 1984]
C : [Howlett, 1977]

SPECIES	HIGHER HEATING VALUE		
	A	B	C
Douglas Fir	9800 10100	9430	9790
Balsam Fir	8861 9477 9100	-	8950
Eastern Hemlock	8890 9383 9517	-	8870
Western Hemlock	9400 9800	9450	9400
Western Larch	8204 8793	-	8280
Jack Pine	8930 9380	-	-
Lodgepole Pine	10190 10794	-	10260
Ponderosa Pine	9100	10350	-
Scots Pine	8595	-	-
Slash Pine	9002 9618	-	-
Southern Pine	8837	-	9360
Spruce Pine	8617	-	-

Table 2.5.b continued. Higher Heating Value of Bark of Softwoods
(in Btu/lb of dry weight)
Sources :
A : [Junge, 1980]
B : [Rossi, 1984]
C : [Howlett, 1977]

SPECIES	HIGHER HEATING VALUE A	 B	 C
Virginia Pine	8424	-	-
E. White Pine	9647	-	-
Western White Pine	9085	-	9090
Western Red Cedar	8700	8960	8790
Black Spruce	8846 8819 8610 9000 9143	-	-
Engelmann Spruce	8359 8846	-	8420
Norway Spruce	8568	-	-
Red Spruce	8630	-	-
White Spruce	8530	-	-
Tamarack	9010	-	-

Table 2.5.c. Higher Heating Value of Bark of Hardwoods
(in Btu/lb of dry weight)
Sources :
A : [Junge, 1980]
B : [Rossi, 1984]
C : [Howlett, 1977]

SPECIES	HIGHER HEATING VALUE		
	A	B	C
Red Alder	7947 8436 8760	8360	8410
Green Ash	8367	-	-
White Ash	8453	-	-
Quaking Aspen	8433 8924	-	8500
American Beech	7640 7993	-	-
Paper Birch	9434 9910 10310	-	9490
European W. Birch	10422	-	-
Yellow Birch	9200 9574	-	9140
Blackgum	7936	-	-
Black Cottonwood	9000	-	9000
American Elm	6921 7418 7600	-	-
Soft Elm	7600	-	-
Shagbark Hickory	8423	-	-

Table 2.5.c. continued. Higher Heating Value of Bark of Hardwoods (in Btu/lb of dry weight)
Sources :
A : [Junge, 1980]
B : [Rossi, 1984]
C : [Howlett, 1977]

SPECIES	HIGHER HEATING VALUE		
	A	B	C
Hard Maple	8230	-	-
Silver Maple	8360	-	-
Sugar Maple	7301 7767 8230	-	7370
Black Oak	8340	7350	-
Pin Oak	8883	-	-
Post Oak	6773	-	-
Northern Red Oak	8030 8400	-	8080
White Oak	6995 7181	-	7070
Poplar	8810	-	-
Sweetgum	7450 7942 7627	-	7530
Sycamore	7403	-	-
Black Tupelo	7942 8102	-	-
Black Willow	7168 7683 8137	-	7250

Chapter 3

COMBUSTION OF WOOD

Combustion is a fast and highly exothermic chemical reaction between fuel and an oxidant. In complete combustion of wood, hydrogen and carbon in the fuel react with oxygen in the air producing water and carbon dioxide. Combustion can take place in the gas phase, called flaming combustion, or in the solid phase, called glowing combustion. Combustion of wood takes place both in gaseous and solid phase, proceeded by drying of the wood.

Although for centuries wood has been the main source of energy for man (with the exception of solar energy), combustion of wood still is far from being completely understood and formulated in mathematical forms. This is not only due to the heterogeneousness of wood, but also because combustion of wood involves a complex series of mass and heat transfer processes accompanied by a series of chemical reactions. Different stages of wood combustion are discussed in the next sections.

Drying

The first stage of wood combustion is the evaporation of water in the wood. This is an endothermic process in which energy must be supplied to vaporize the water. The method of calculation of the energy required to vaporize the moisture in the wood is given in Chapter 5.

Pyrolysis and Combustion of Volatiles

Since the ignition temperature of wood is higher than the charing temperature [Wise, 1952], under the influence of a sufficiently strong source of energy, wood will pyrolyze to form volatile products. Pyrolysis is "an endothermic irreversible chemical degradation of wood in which virgin wood is transformed into char and combustible vapors" [Kanury, 1972].

In Chapter 2 it was mentioned that wood and bark are mainly composed of hollocellulose and lignin. These materials are not directly combustible, and under the influence of a heat source they decompose to volatile material and char. It has been reported, however, that the cellulosic component is mainly converted to the volatiles; whereas, the lignin component contributes mainly to the char fraction [Shafizadeh, 1977]. Also, the pyrolysis of holocellulose will proceed to completion before the pyrolysis of the lignin reaches a very advanced stage [Roberts, 1970].

The proximate analyses of wood and bark (table 2.1) show that wood and bark contain about 70 to 90 percent volatile materials. When combustible volatiles are driven off they can react with the oxygen and burn in the gas phase with flaming combustion. This reaction provides heat for further drying and thermal degradation of the fuel, as well as the raising of the temperature of the combustion products.

Combustion of Char

The residue remaining after pyrolysis is a highly reactive carbonaceous char. Oxidation of this char in solid phase gives glowing combustion which has a relatively slower rate of combustion than flaming combustion. If the intensity of the heat flow or combustion gases (oxygen supply) fall below a minimum level, smoldering combustion takes place in which unoxidized volatile products and aerosol particles of the are emitted as smoke.

To summarize the process of combustion of wood, figure 3.1 is presented. This is, however, a very simplified presentation of the actual process.

The three successive and overlapping processes of wood combustion mentioned above (water evaporation, pyrolysis and combustion of volatile materials, and char oxidation) are controlled partly by the chemical and physical composition and properties of the and partly by the prevailing conditions of heat and mass transfer

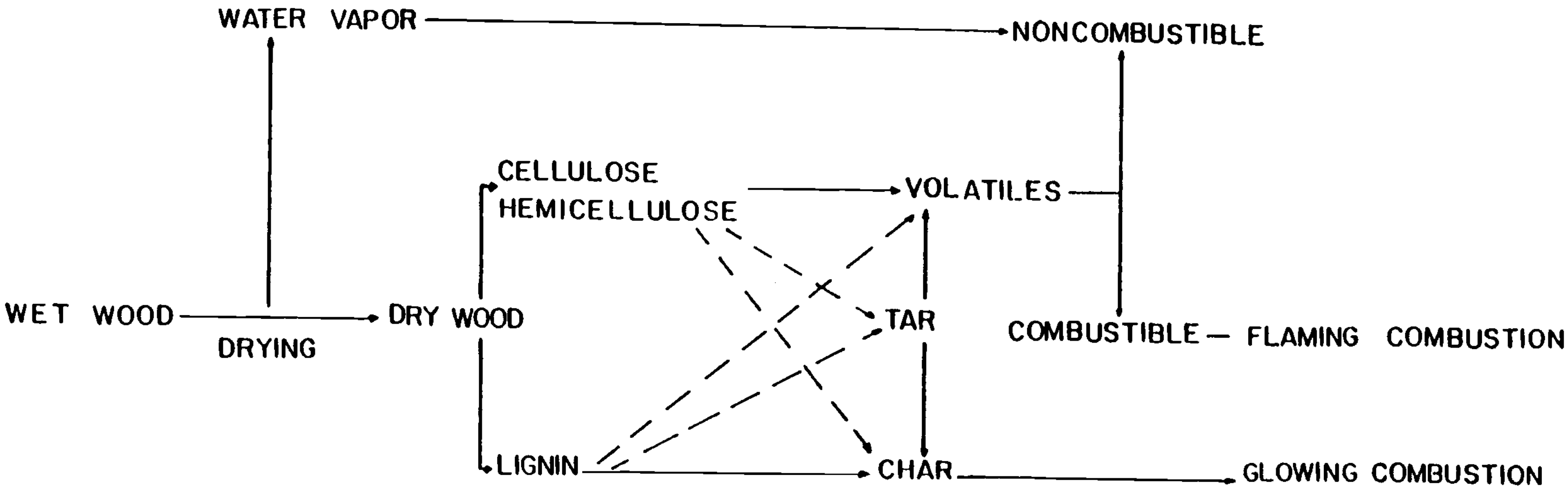

Figure 3.1. Stages of wood combustion.

imposed on the wood particles in the furnace. In the next section some of these factors and their effects on the combustion of wood are discussed.

Factors Affecting the Combustion of Wood

1- Effects of Moisture Content. Perhaps the most important factor in wood combustion is its moisture content. Moisture content affects not only the combustion process and its efficiency, but also greatly affects the net heating value of wood fuel, as well as the selection of equipment used for combustion, fuel handling, storage, transportation, and air pollution.

Evaporation of moisture in wet wood is an endothermic process, requiring approximately 1000 Btu per pound of water vaporized. Therefore, as the moisture content of wood is increased the net heating value of wood is decreased. Figure 3.2 shows a graph of net heating value versus moisture content of wood.

It has been reported that at a moisture content of about 68 percent (wet basis) furnace blackout occurs [Drucker, 1984]; therefore, to sustain the combustion of fuel with moisture content higher than this limit, additional energy (supplemental fuel) must be supplied to the burner.

As moisture content increases the rate of combustion decreases. For example, by increasing the moisture

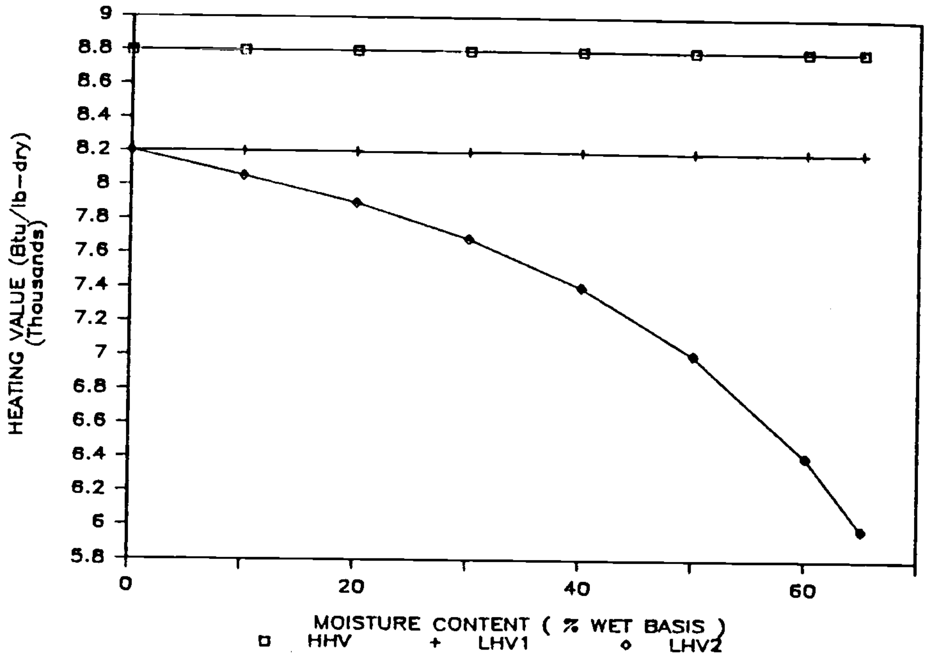

Figure 3.2. Net and gross heating value of Douglas fir (higher heating value is assumed to be 8800 Btu/lb of dry wood).

content of a 10 mm cube of pine wood from 0 to 200 percent dry basis (0 to 67 percent wet basis), the total burn time for the particle in 1100 K air increases from 81 seconds to 160 seconds [Simmons, 1983]. Flame temperature also decreases as moisture content increases, due to the energy used for vaporization of moisture. A slower combustion rate together with decreased flame temperature will further retard combustion, since the driving force for heat transfer to the wood particles is weakened.

Because combustion rate is slower, each particle requires more residence time in the furnace. Therefore, for a fixed residence-time combustor, particulate emissions and the amount of unburned fuel increase, reducing the combustion efficiency and increasing the difficulty of air pollution control.

The volume of water in the vapor phase is approximately 5700 times the volume of water in the liquid phase. Evaporation of water in wood, therefore, results in an increase in the volume of the gases in the combustor. This increased volume (and consequently gas velocity) will result in an increased amount of particulate emission and unburned fuel.

Since the net heating value of the wood decreases, as moisture content increases, the rate of fuel feed for a given combustor load must be increased. This increases the size requirement of the fuel handling system and as a result increases the equipment costs. Higher moisture content also increases the cost of fuel transportation.

Another disservice of variable moisture content is that it makes the control of combustion very difficult [Junge, 1980].

An advantage of high moisture content in wood is that it prevents dusting in open containers and conveyors.

Finally, the moisture content of the fuel (because of the influence of moisture content on the combustion process) determines (to a large extent) the type of the combustion system used for direct combustion of wood. It should also be noted that drying of wood is a costly operation and is not always beneficial to overall efficiency of the system or operating and initial cost of the power plant.

2- Effects of Particle Size. In Chapter 2 it was mentioned that wood fuels have a wide range of particle size. The size of particles entering the combustor not only affects the determination of the type of fuel handling and combustion system, but also greatly affects the combustion process itself.

Heat transfer to wood particles is directly proportional to the exposed surface area of the particles. Smaller particles have a larger surface area to mass ratio than larger particles. This in turn means a faster rate of heat transfer to the particles and, consequently, a faster combustion rate. In addition, larger particles tend to insulate their interiors with the formation of char which further reduces the rate of combustion. As an example, increasing the size of a pine wood cube from 5 mm to 25 mm increases the total burn time from 30 seconds to 400 seconds [Simmons, 1983].

The size of wood fuel particles could range from $8\text{x}10^{-5}$ to 4 inches; this is a ratio of $5\text{x}10^{4}$ [Smith, 1980]. Each type of combustion system is designed for a certain range of particle size. For example, suspension burners are designed for small particles; whereas, grate type burners are designed for larger particles (see Chapter 4). Introduction of larger than specified fuel particles in a suspension burner will result in an increase in the amount of unburned fuel, since the larger particles tend to fall and not be burned in suspension. On the other hand, in grate type burners, if very small particles of wood are introduced the amount of unburned fuel is increased. This increase is due to an increased amount of fine particles leaving the furnace as flyash, which also increases air pollution and reduces burner's efficiency.

Smaller particles burn (since they have a larger surface area per unit mass) at a faster rate than larger particles, producing an increase in excess air requirement. This increases the power requirement of the combustor.

Larger particles also make the control of combustion difficult due to their high thermal inertia. Control of combustion becomes more difficult when moisture content and particle size are increased together. In addition to the effects mentioned above, improperly sized particles could have an adverse effect on fuel feeding systems, air pollution control devices, ash handling systems, and fuel transportation.

3- Effects of Excess Air. To prevent incomplete combustion of fuel in a furnace, oxygen, in excess of the stoichiometric amount needed for combustion, is usually supplied. The amount of air supplied in excess of stoichiometric is termed excess air. The excess air level can range from 0 to over 200 percent depending on the type of the combustion system used. However, if the excess air level exceeds the design limits, it can have adverse effects on combustion.

Increasing the excess air (above the specified amount for the combustor) reduces the flame and furnace temperature. As a result, the rate of combustion is reduced, while the amount of unburned carbon and particulate emissions due to incomplete combustion of the particles are increased.

Since the volume of the furnace is fixed, increased excess air levels increase the volume flow rate of the gas in the furnace which results in an increase in gas velocity. The residence time of the particles is reduced due to increased gas velocity. Reduction in residence time will result in increased particulate carry over and unburned carbon, factors which reduce combustion efficiency and increase air pollution.

Increased excess air also increases the pressure drop through the combustor, resulting in a higher power consumption for the unit.

In addition to the amount of excess air present during combustion, the temperature of the air and the location of the introduction of air into the furnace play an important role in the combustion process. For spreader stoker burners, it has been found that the optimum combustion conditions are achieved when 50 percent excess air is used. Of this amount of air about 38 to 48 percent should be used as underfire air and the rest as overfire air [Junge, 1979].

In this section effects of moisture content, wood fuel particle size, and excess air have been discussed. It should be realized that these are not the only factors affecting combustion of the wood. Physical and chemical properties of a wood, such as chemical composition and thermal properties, play an important role in its combustion. Other factors such as Reynolds number based on particle diameter [Simmons, 1983] also come into play. These were not discussed here since the goal of this work is not to study the combustion of wood for a single particle but for lumped quantities of wood fuel that are used in power generation.

Chapter 4

COMBUSTION SYSTEMS

Since wood fuels, in general, differ greatly in composition, moisture content, and size as well as method of utilization, there has been a variety of furnace designs used for the direct combustion of wood. The variety of furnace designs for wood combustion have steadily increased due to the high cost of fossil fuel and the need for more efficient methods of wood combustion. Until the late 1940's Dutch oven furnaces were utilized almost exclusively for wood combustion [Chermisinoff, 1980]. However, today, there are a large number of different combustion systems for wood fuels. In this chapter some of these combustion systems will be reviewed.

Combustion systems for wood fuel can be classified according to the method of burning wood; these are:

1- Grate Burners
    I- Stationary Grates
        a- Flat Grates
        b- Inclined Grates
    II- Moving Grates
        a- Mechanical Grates
        b- Traveling Grates

2- Suspension Burner

3- Cyclone Burner

4- Fluidized Bed Burner

1- Grate Burners

I- Stationary Grates

a- Flat Grate Burner. In the flat grate burners, which are the oldest and simplest method of wood fuel firing, the fuel pile rests on a flat grate through which primary air (also called underfire or undergrate air) is forced through the pile of fuel. The underfire air generally determines the rate of combustion by providing part of the oxygen for combustion. It also cools the grates, promotes turbulence in the fuel pile, and contributes to drying of the fuel. As fuel moisture content increases, the requirement for primary air also increases. Overfire air (also called secondary air) is introduced above the fuel pile to complete the combustion. Another purpose of the secondary air is to induce turbulence and mixing for better combustion of volatiles. Overfire air is sometimes blown through numerous small nozzles to promote circulation, and therefore better mixing.

Flat grate burners could have one or more chambers. In the first chamber the fuel pile is located and primary air is introduced. The second chamber is used to increase the residence time and is, usually, the place of introduction of secondary air. Some burners even utilize a third chamber where tertiary air is introduced.

Since the thickness of the fuel pile on the grate is uneven, most of the primary air flows through the shallow rim of the fuel pile creating an uneven combustion in the fuel pile. This causes a sudden flow in the fuel pile due to changes of density of the pile, causing further disturbance in combustion. In pile burning it is also very difficult to maintain a constant air fuel ratio, making the control of combustion difficult.

In this type of burner, a variety of particle sizes can be fired, however, baking fuels such as saw are not suitable for this type of burner. Flat grates are suitable for wet fuels with moisture content up to 65% as well as for dry fuels.

In general, the control of the combustion in a pile burner is difficult due to the high thermal inertia of the fuel pile, which results in a slow response to load changes. Because of the high thermal inertia of the fuel pile (and the difficulty of combustion control), pile burners are designed for capacities up to thirty million Btu/hr (30 MM Btu/hr) of fuel input. Utilization of radiant heat transfer to heating surfaces is limited; and, therefore, the heat loss of the system is high.

Important design characteristics of this type of burner are the proportion of cooled and refractory surfaces in the combustion chamber and the location and rate of primary and secondary air. Another important design parameter is the heat release per grate surface area. Typical heat release rates are in the range of 725×10^3 Btu/ft2 of grate surface area.

To overcome some of the weaknesses and operational difficulties of the grate burners, many design improvements have been made. These improvements include use of multiple chambers, pre-heating the combustion air, alternative methods of feeding the fuel, and utilization of water cooled grates. An example of pile burner (Dutch oven) and an example of semi-pile burner (spreader stoker) is now presented:

Dutch Oven. The Dutch oven, or fuel cell, is the oldest type of pile burner in which fuel is gravity fed through a fuel chute into the furnace. A typical Dutch oven is shown in figure 4.1. This method of feeding the fuel increases the particulate emissions and the amount of unburned carbon. It also cools the combustion chamber. This could be corrected with the use of an underfeed stoker (figure 4.2). Dutch ovens have a very low thermal efficiency (about 50 to 60 percent) due to increased heat loss caused by large furnace surface areas and the absence of radiant heating to heat transfer surfaces. Since the thermal inertia of the fuel pile in Dutch ovens is very high, combustion control is very difficult and the response to load variations of the combustor is very slow.

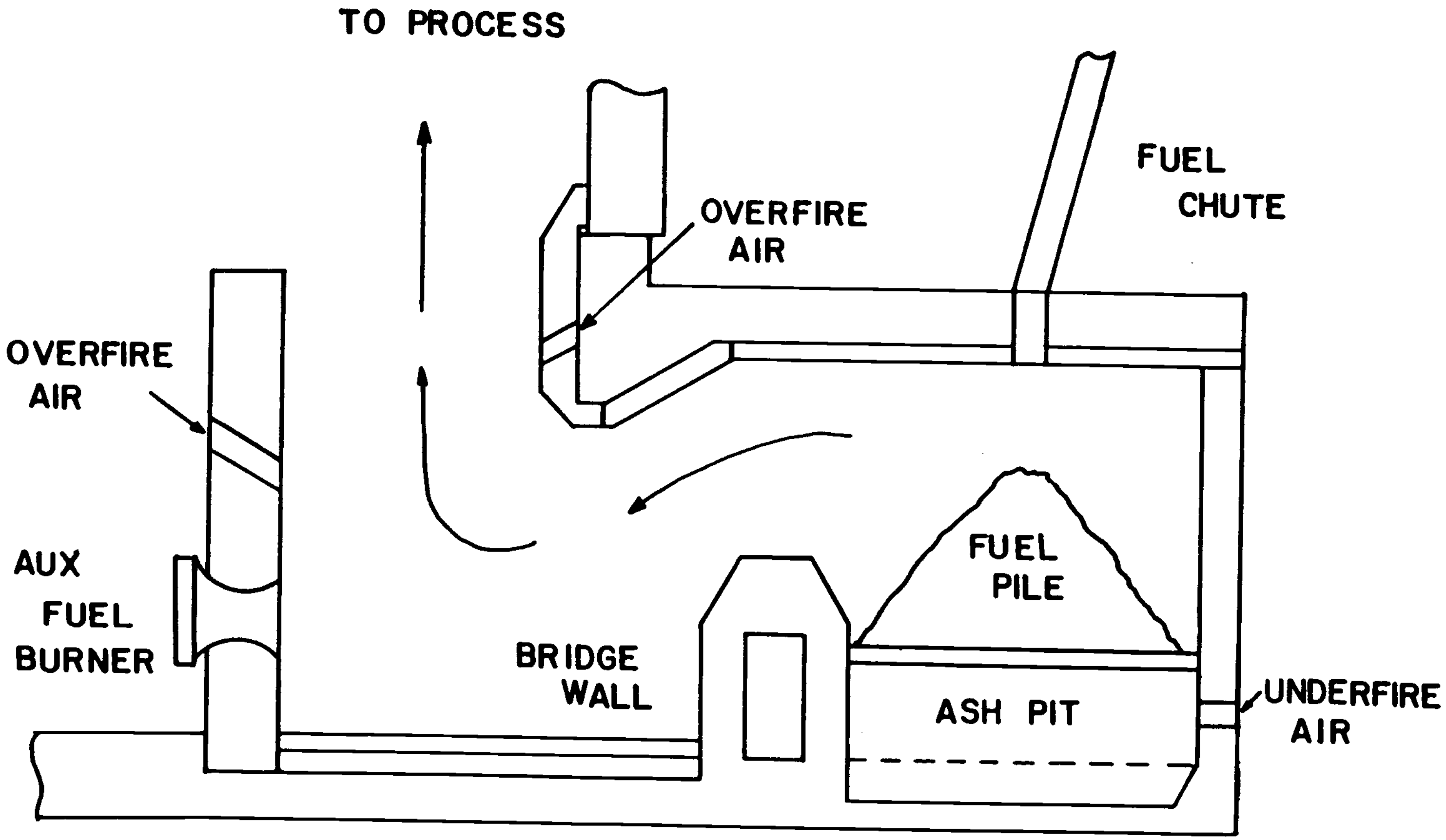

Figure 4.1. Dutch oven.

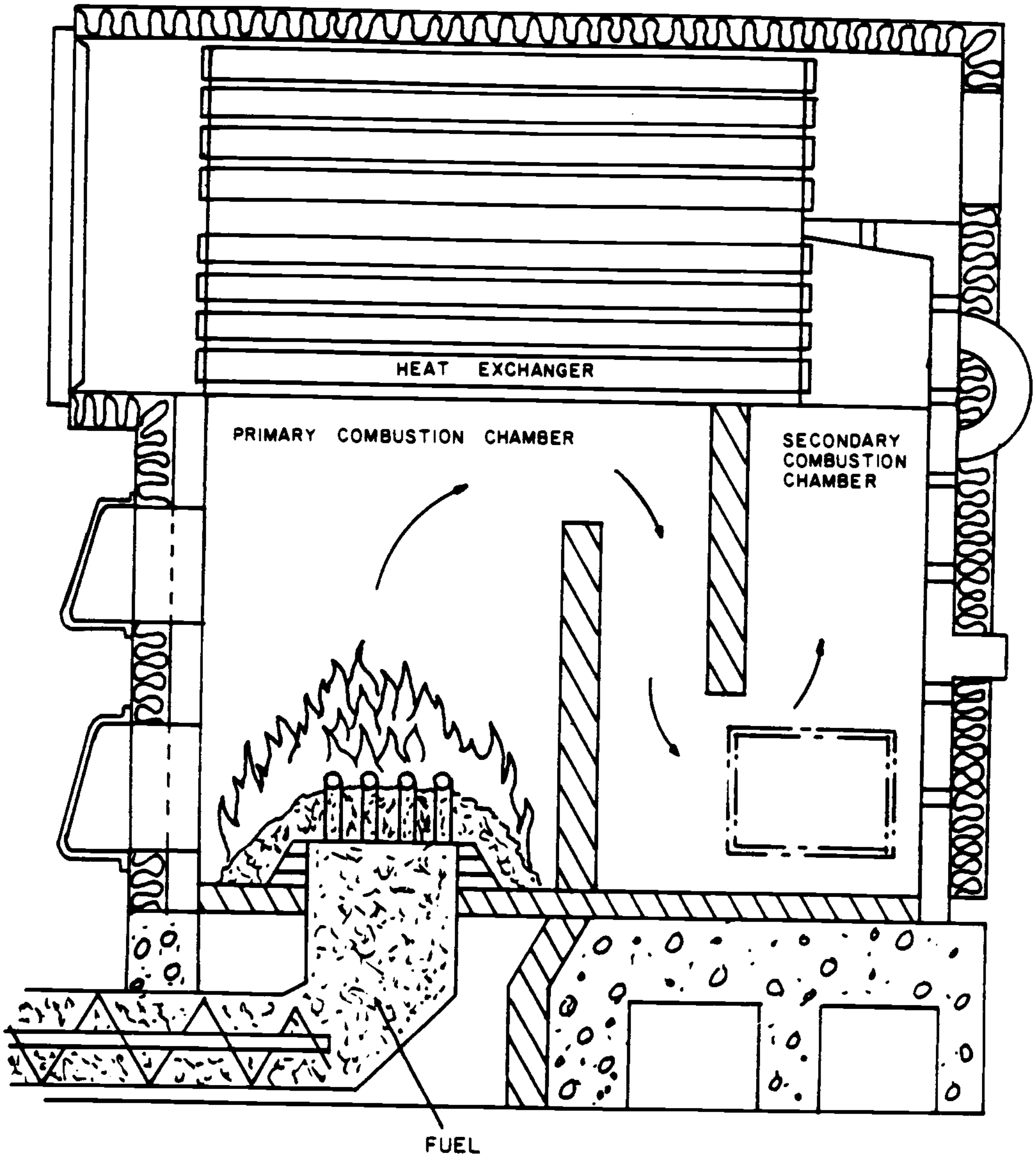

Figure 4.2. Flat grate pile-burning system with an under feed stoker.

Major advantages of the dutch oven are their ability to burn wet fuel and their insensitivity to size variation of fuel particles (with the exception of baking fuels).

Spreader Stoker. In spreader stoker burners the fuel is scattered, or spread, over the burning fuel bed. The stoker can be mechanical or pneumatic. Pneumatic stokers are well suited for particles with inconsistent sizes. When fuel particles are introduced into the furnace, the smaller particles burn in suspension while larger particles fall on the grate where they are burned. Spreader stoker type burners have a much larger capacity than Dutch ovens because of higher rate of heat transfer to suspended particles. Combustion control of spreader stoker burners is easier than for Dutch ovens. However, particulate emission is very sensitive to the size and moisture content of the fuel; as a result, spreader stoker burners require a large combustion chamber to give enough residence time for moisture and larger particles.

Use of water cooled grates in spreader stoker burners will make the use of pre-heated combustion air possible, increasing the rate and efficiency of the combustion as well as increasing the life of the grates.

Typical heat release rates for this type of burner is in the range of 35000 to 25000 Btu/hr per cubic foot of furnace volume. The lower values result in improved efficiency and reduction of the particulate emissions (since the volume of the furnace, and the residence time of the particles is increased). Water cooling of the grates is also essential for improved efficiency since it makes the use of pre-heated air possible and increases the life of the grates.

Spreader stoker burners can be used to burn coal or mixed fuel with minor modifications; however, water cooling of the furnace is a necessity if coal is to be burned.

The use of secondary air through air nozzles to create turbulence is required for optimum performance.

Disadvantages of the spreader stoker include the possibility of particulate emissions due to overfeeding, and extinguishing the flame due to fuel interruptions.

A typical spreader stoker is shown in figure 4.3.

b- Inclined Grates. The inclined grate burners are another type of semi-pile burners. In this type of burner fuel is introduced to the furnace at the top part of the grate. The inclination of the grate is designed according to fuel properties, such as moisture content and size of the particles. The slope of the grate, which could be varying along the grate, determines the velocity and the thickness of the fuel bed. This slope could range between 37 to 55 degrees. The slope of the grates,

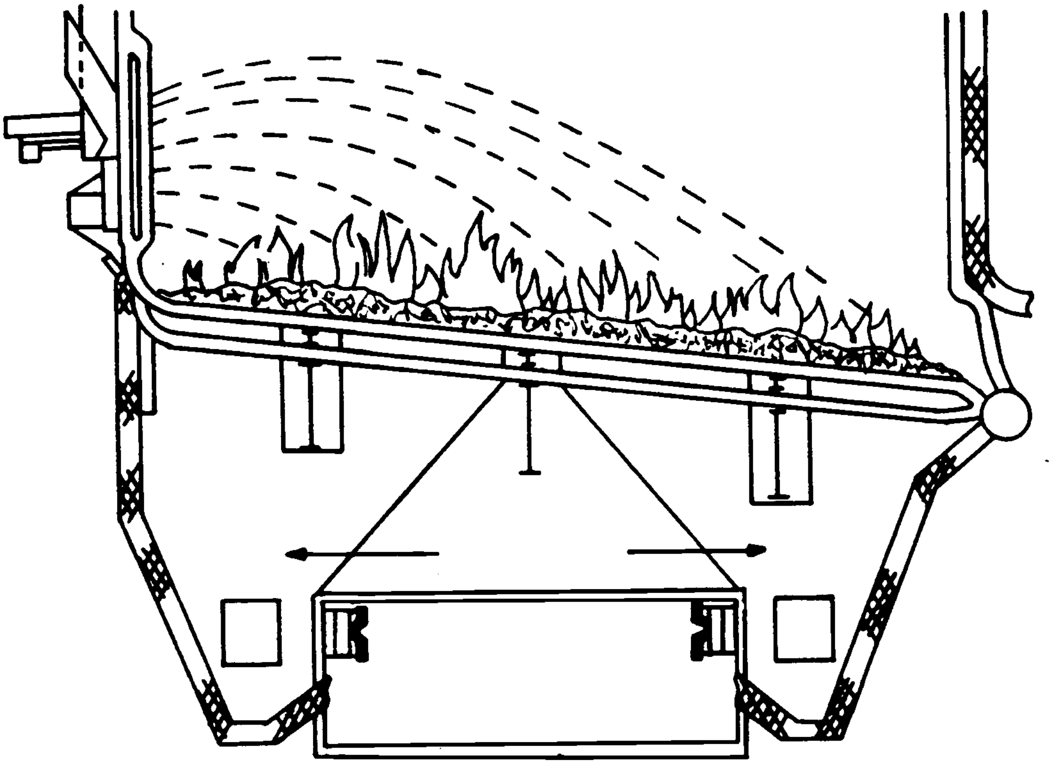

Figure 4.3. Water cooled flat grate with a spreader stoker.

however, remains fixed after it is designed. At the top part of the grate fuel is heated and dried; in the middle section pyrolysis takes place; and at the bottom of the grate char oxidation takes place. Since these processes take place in different sections of the grate, the slope of the grate and the amount of the primary air supply can be chosen such that optimum requirements for each section are met. By controlling the amount of the primary air supplied to each section of the grate, improvements in combustion and its control can be achieved. Use of preheated combustion air (up to 900 F, which requires water cooled grates) can drastically improve the combustion and make the burning of higher moisture content fuels (up to 65%) possible. Water cooled grates, which consist of parallel water tubes running along the slope, have been used in recent designs.

The use of a radiation or ignition arch in European designs is another important improvement in combustor design. This brings the hot refractory surfaces close to the fuel bed to help either in drying, pyrolysis or char combustion depending on properties of the fuel. This type of burner is shown in figure 4.4. A mechanical grate section at the low end of the grate can also be used to remove the ash.

Similar to flat grate designs, inclined grate burners have a large secondary chamber to complete combustion.

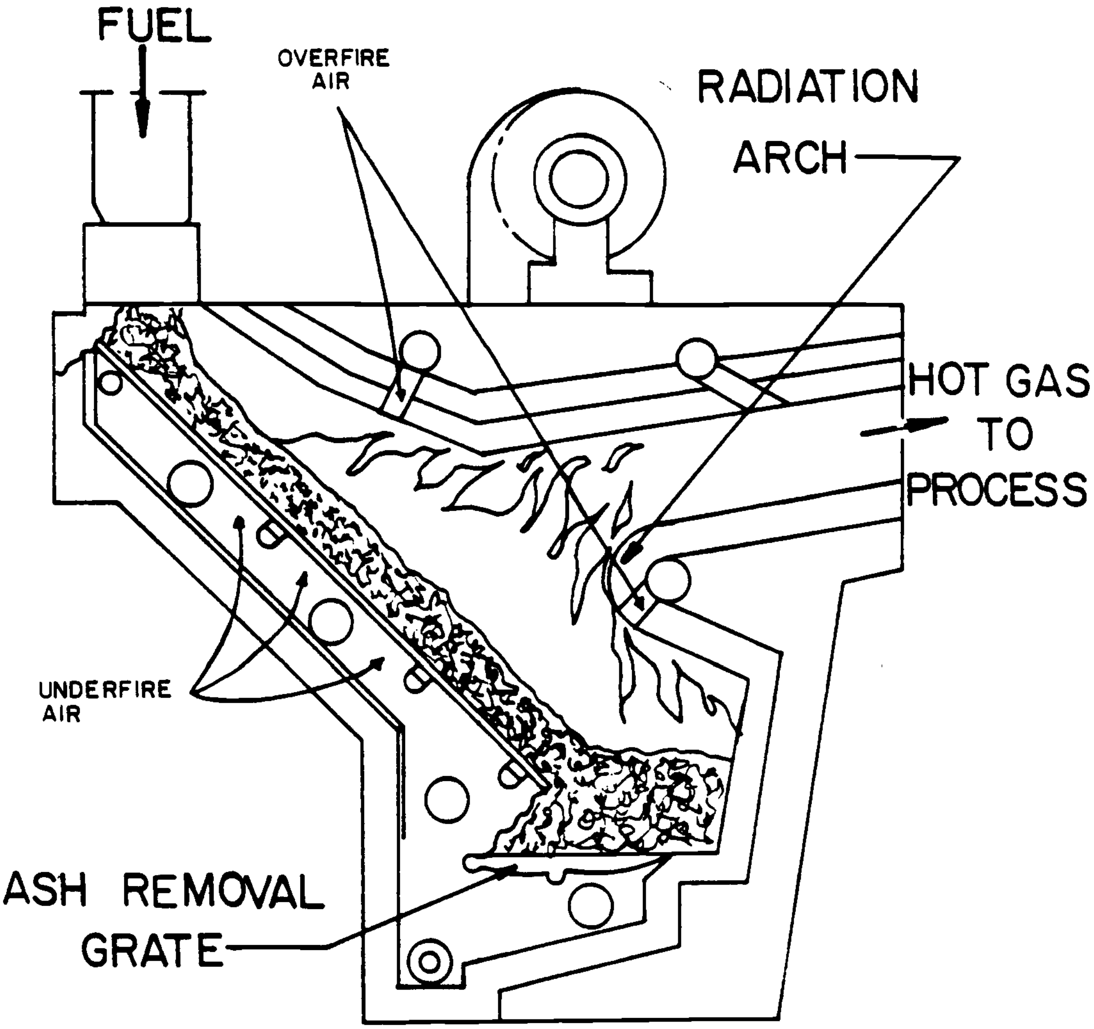

Figure 4.4. Small European inclined grate burner.

The secondary air helps with turbulence and provides the additional oxygen needed for combustion.

The capacity of inclined grate burners ranges from 10 to 400x10^6 Btu/hr of fuel input and is capable of burning fuel with variable size and high moisture content (up to 65%).

Important design features of the inclined grate burner include rate, temperature and location of primary and secondary air; and the proportion of cooled and refractory surfaces in the combustion chamber. Typical heat release rates are in the range of 200 to 400x10E3 Btu/hr per square foot of grate surface area [Leppa, 1982].

Figure 4.5 shows the inclined grate burner of a typical American design. It has a simpler design than European inclined burners and lacks the radiation arch.

II- Moving Grate Burner

The moving grate burner offers another option for wood fuel burning. A moving grate can be mechanical or traveling. These types of burners have moving parts which can mean high maintenance costs. On the other hand, they have better combustion control than stationary grate burners and usually with minor modifications could be used for coal or coal and wood burning.

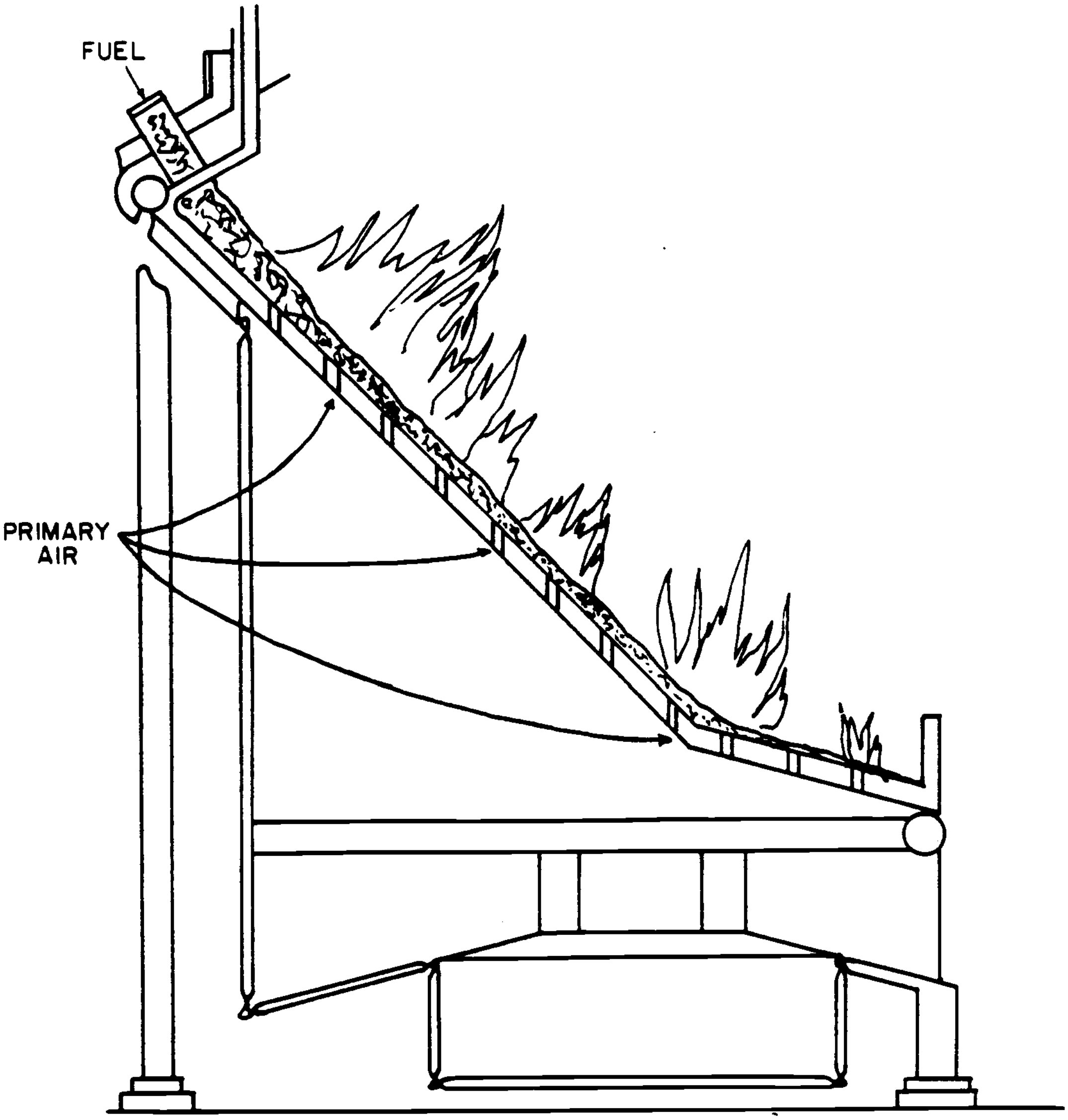

Figure 4.5. American inclined grate burner.

a- Mechanical Grate Burner. This type of burner utilizes grate bars or rotating sector blades to move the fuel forward while turning and stirring the fuel in the furnace. A combination of mechanical and inclined grates can increase the limit of the moisture content of the wood from a maximum of 60 to 65 percent.

The important design parameters of this type of burner, which is mostly used in Europe, are heat release rate per grate area; rate, temperature and location of primary and secondary air; length and speed of grates; and proportion of cooled and refractory surfaces. Figure 4.6 shows a typical mechanical grate burner.

b- Traveling Grate. The traveling grate consists of many small grates attached to chains which pull the grate, thus forming an endless grate moving slowly through the furnace and continuously discharging the ash into the ashpit. The primary air enters through the moving grate while sealing takes place between the stationary and moving surfaces.

This American dominated design has several distinctive features. One of these features is automatic dumping of the ash. With an accelerated motion of the grate, rapid shut-down of the fuel bed can be achieved. The velocity of the traveling grate, upon variation, could change the thickness of the fuel bed and residence time of

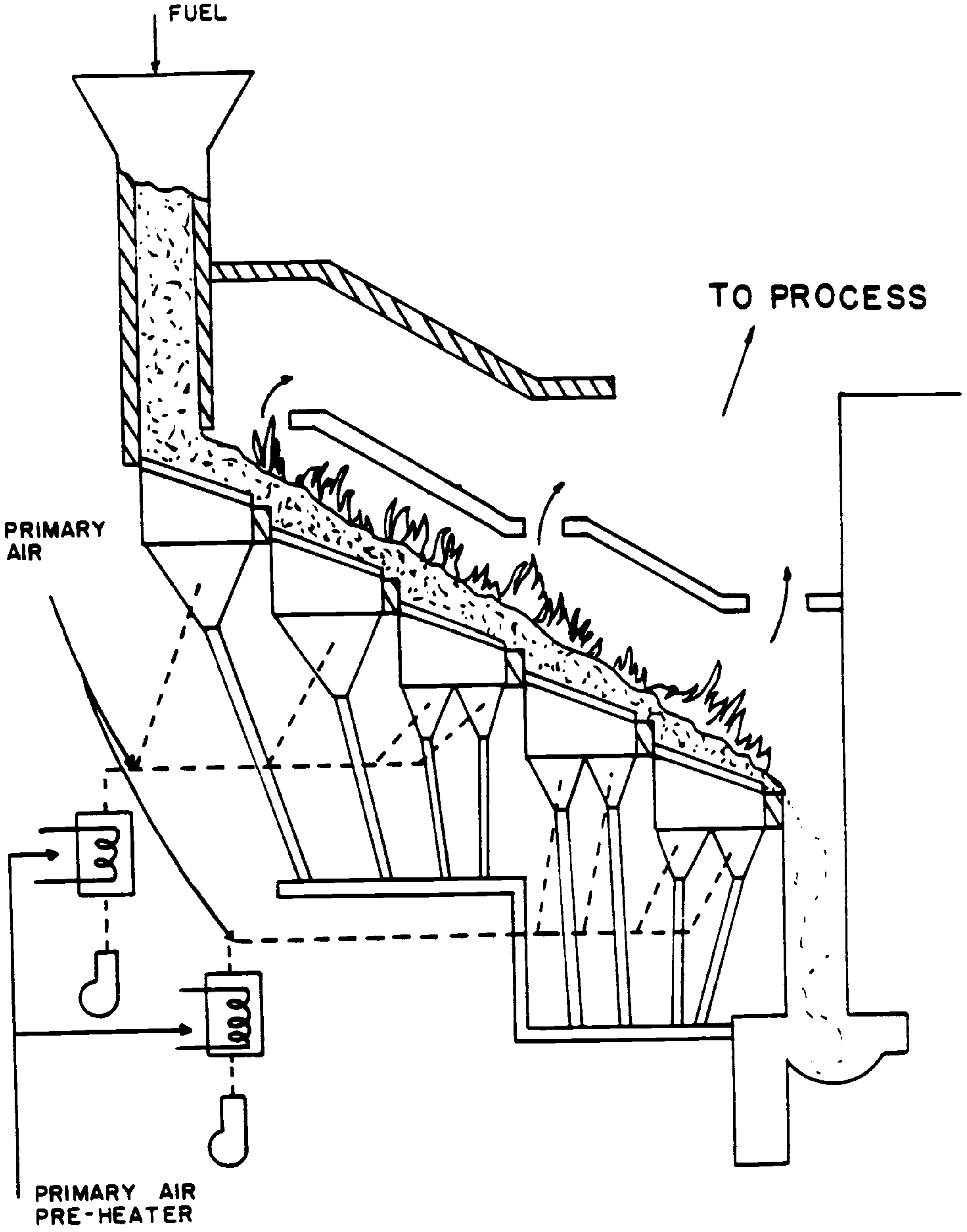

Figure 4.6. Mechanical grate burner.

the particles. Therefore, varying fuel parameters such as size and moisture content can be easily accommodated.

In traveling grate burners two different firing methods can be employed: one is a gravity fed system with forward moving grate (shown in figure 4.7); the other, uses a spreader stoker and a traveling grate moving toward the stoker (shown in figure 4.8). The latter design offers a wide range of available retention time in the furnace, permits better control of combustion, and allows adjustments for varying fuel properties.

Use of a spreader stoker, as it was mentioned earlier, increases the possibility of particulate emissions which are a function of size and moisture content of the fuel particles. Therefore, a larger combustion chamber is required to provide enough time for suspended particles to burn.

Traveling grate burners have a capacity in the range of 50 to 600×10^6 Btu/hr of fuel input. Traveling grate burners are capable of burning wood fuel with particle sizes from 0.25 to 1.5 inch and a moisture content of up to 55% [Leppa, 1982].

Important design parameters of this type of burner are heat release per furnace volume and grate area; rate, location and temperature of primary and secondary air; and location of the stoker.

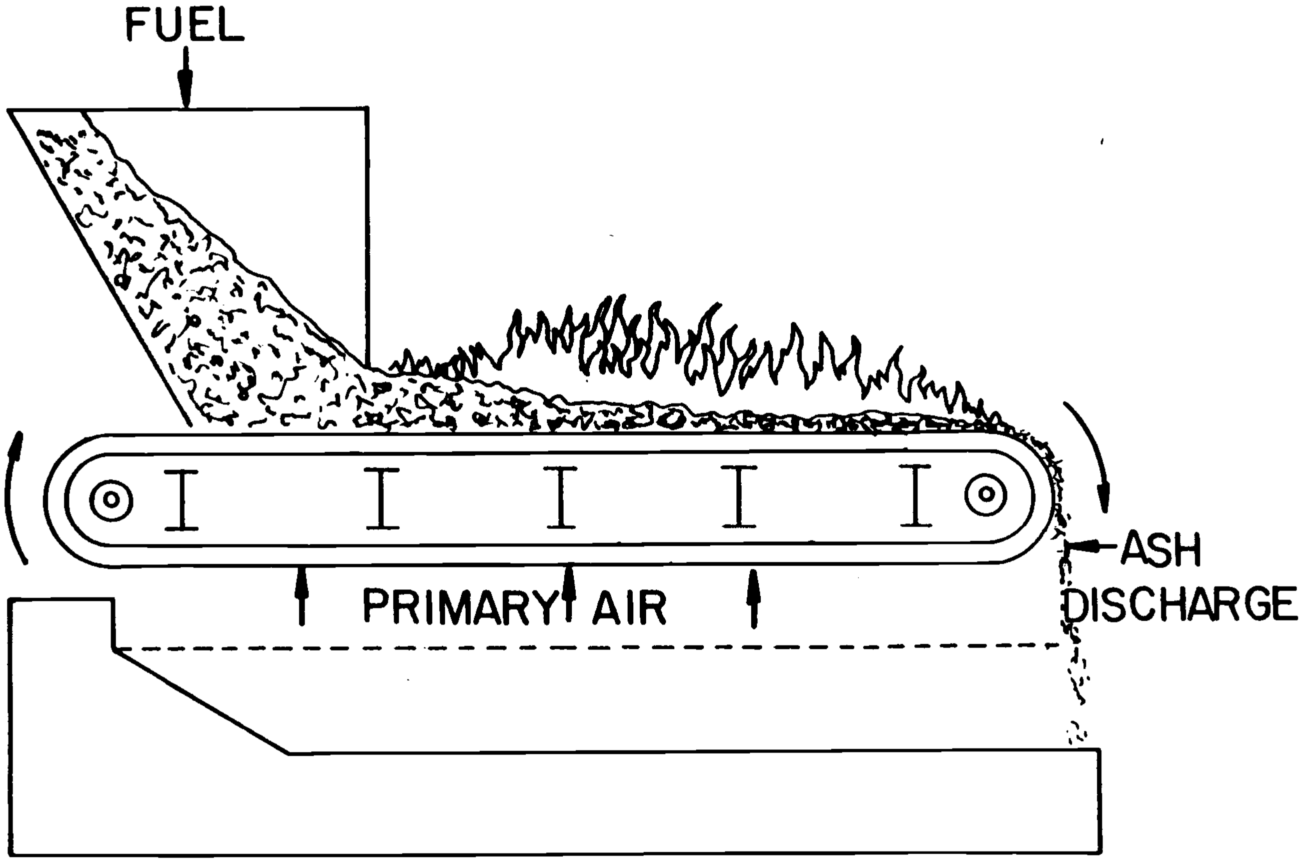

Figure 4.7. Forward (gravity feed) traveling grate.

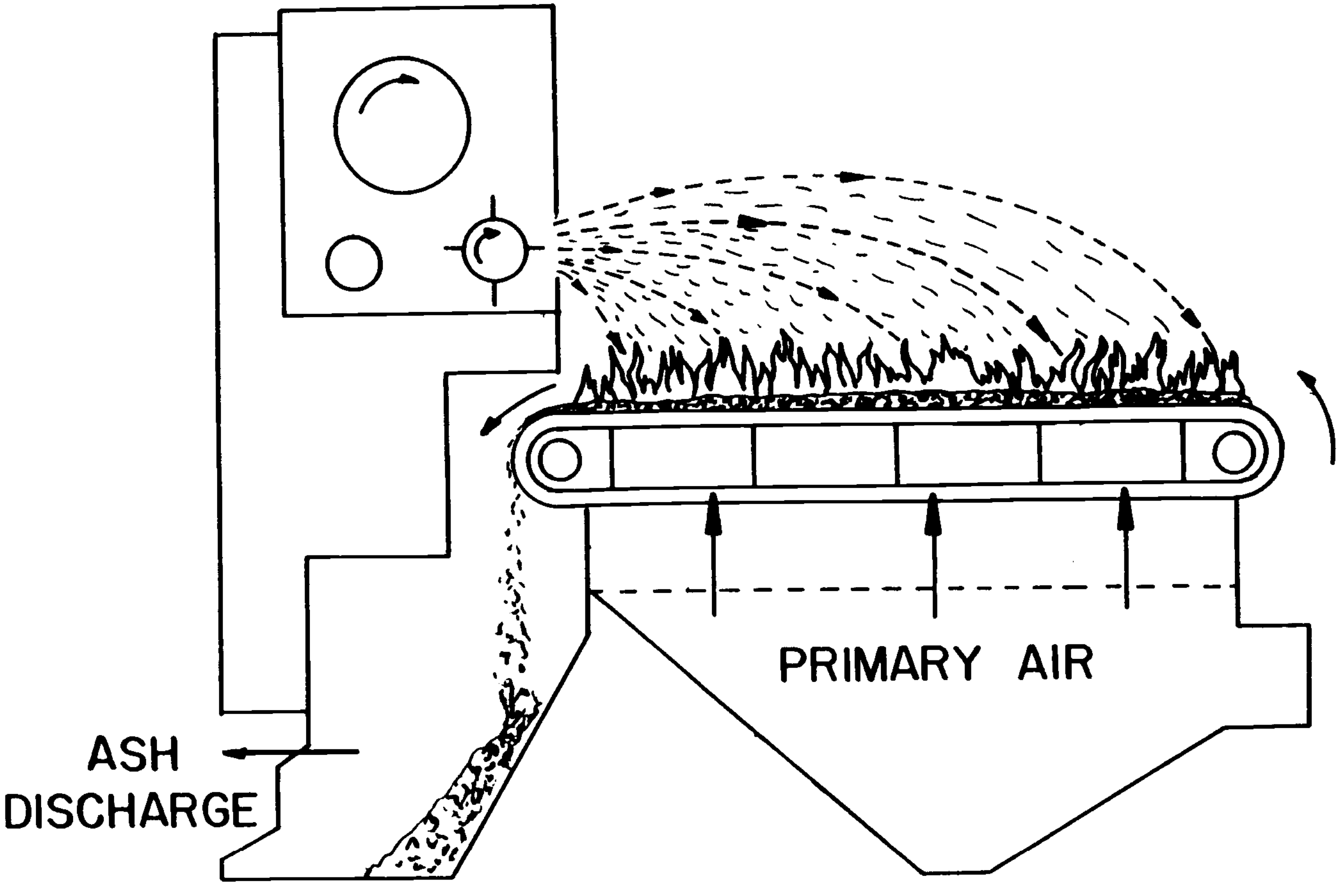

Figure 4.8. Traveling grate with a spreader stoker.

Disadvantages of the traveling grate include high maintenance cost and the possibility of particulate emissions should overfeeding of the furnace occur when using a spreader stoker.

With minor modifications a traveling grate burner could be used to fire coal or a mixture of coal and wood.

2- Suspension Firing

Suspension firing is designed for properly sized and pre-dried wood fuel. Wood particles are mixed with air and are burned in suspension. However, larger or high moisture content particles tend to fall and are not burned in suspension. In many designs a burnout grate is provided . This type of firing requires the most restricted fuel particles both in size (less than 0.25 inch) and moisture content (usually less than 15%). Pre-drying and size reduction of the fuel is necessary since most wood fuels do not meet required conditions when they are received in a power plant. On the other hand, suspension burners have a high turndown ratio due to low thermal inertia, excellent response to load variations, and good efficiency. However, sudden changes in fuel moisture content present very harmful operational disturbances causing incomplete combustion which results in higher particulate emissions and unburned fuel. Since a suspension burner burns dry fuel the combustion temperature could be high enough to generate NOx emissions.

The most important design parameter in a suspension burner is heat release per furnace volume. Typical values are about 14000 Btu/hr per cubic foot of furnace volume. Suspension burners are manufactured with capacities up to 80×10^6 Btu/hr fuel input. Figure 4.9 shows a typical suspension firing system.

3- Cyclone Firing Systems

The distinctive feature of the cyclone firing is the circular or cyclonic motion of secondary air which is introduced tangentially into the cyclone under high pressure. This type of burner has been designed for dry or wet fuel with the ability to burn supplemental fuels. These designs include single stage and double stage with vertical, horizontal, or inclined configurations [Leppa, 1982]. Figure 4.10 shows a European design single-stage cyclonic combustor.

Cyclone firing systems have been built with capacities of up to 125×10^6 Btu/hr of fuel input, and some are capable of burning wet fuel a with moisture content of up to 65% and 4 inches in size. The main design characteristic is heat release per cyclone volume [Leppa, 1982].

Disadvantages of this type of burner are its high

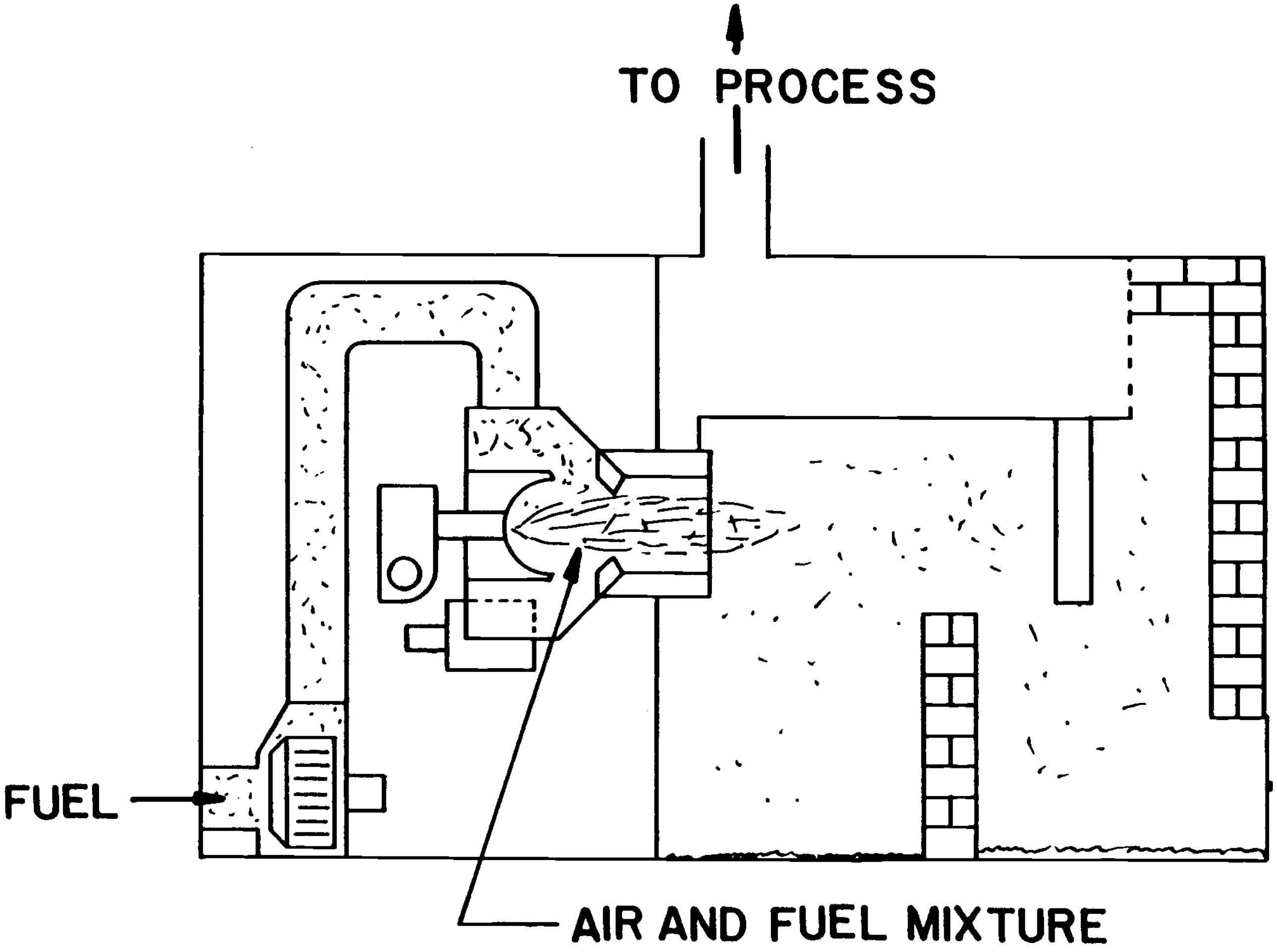

Figure 4.9. Suspension burner.

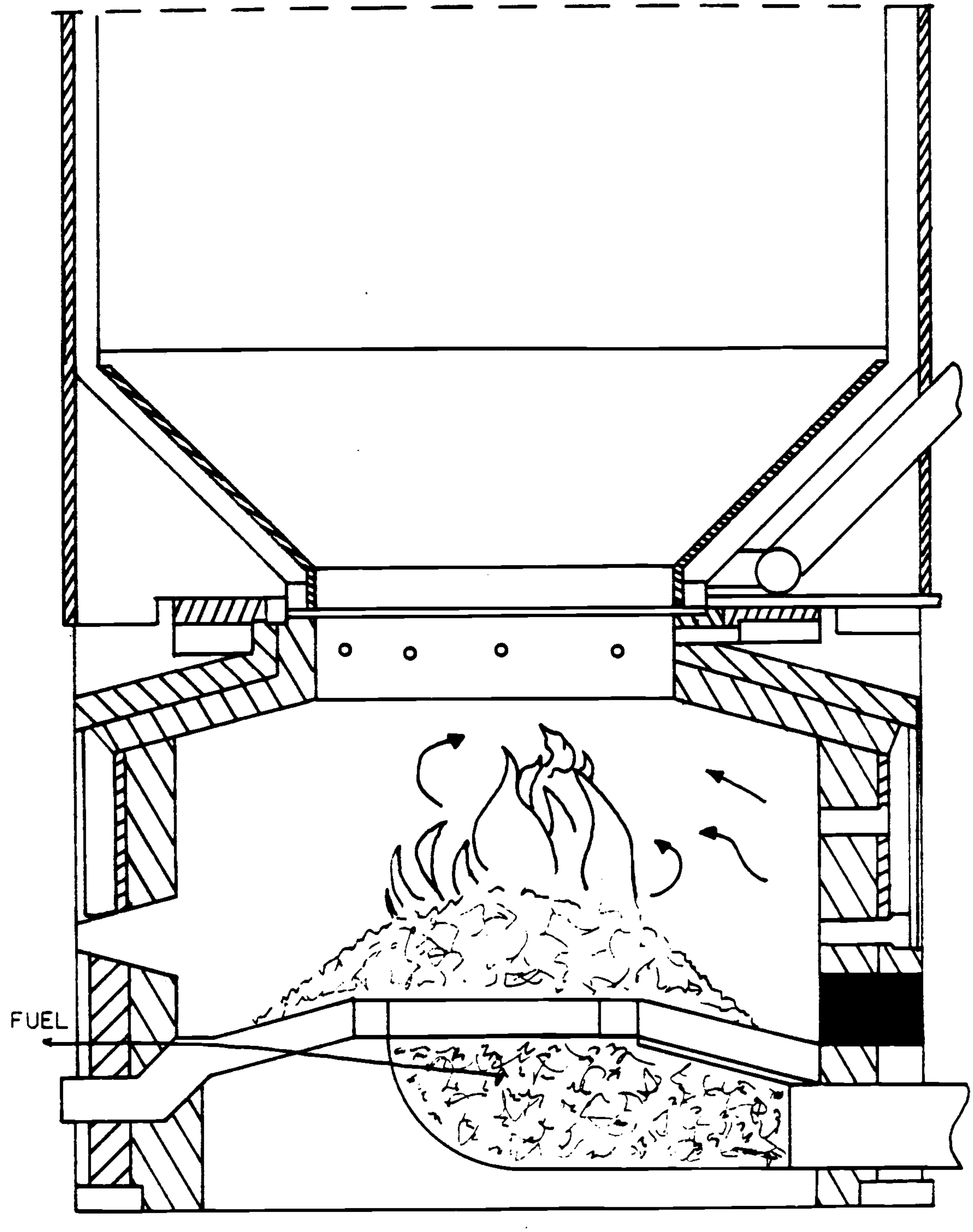

Figure 4.10. Single-stage cyclone burner.

power consumption, poor turn-down ratio, and narrow operational limits for the fuel. Also, high flame temperatures in some designs lead to high NOx emissions.

4- Fluidized Bed

A fluidized bed combustor consists of a chamber with a porous floor (distributor plate) containing a bed of solid inert particles, such as sand or limestone (fluidization medium), through which air under pressure (fluidization gas) is passed at a specific velocity (fluidization velocity) to float and move the particles, resembling the motion of a boiling liquid. A typical fluidized bed is shown in figure 4.11.

To start the combustor the fluidized bed is first heated by an auxiliary fuel. When the bed temperature is high enough to ignite the fuel, the auxiliary fuel is shut off and wood fuel is introduced.

Choice of fluidization medium depends on type and properties of the fuel. For example, the use of limestone in coal combustion results in lower SOx emissions. In wood firing, sand is mostly used because of its low cost and high heat capacity. The high heat capacity of sand permits overnight shutdown of the combustor and re-starting without the use of auxiliary fuel.

The boiling motion of the fluidizing medium produces

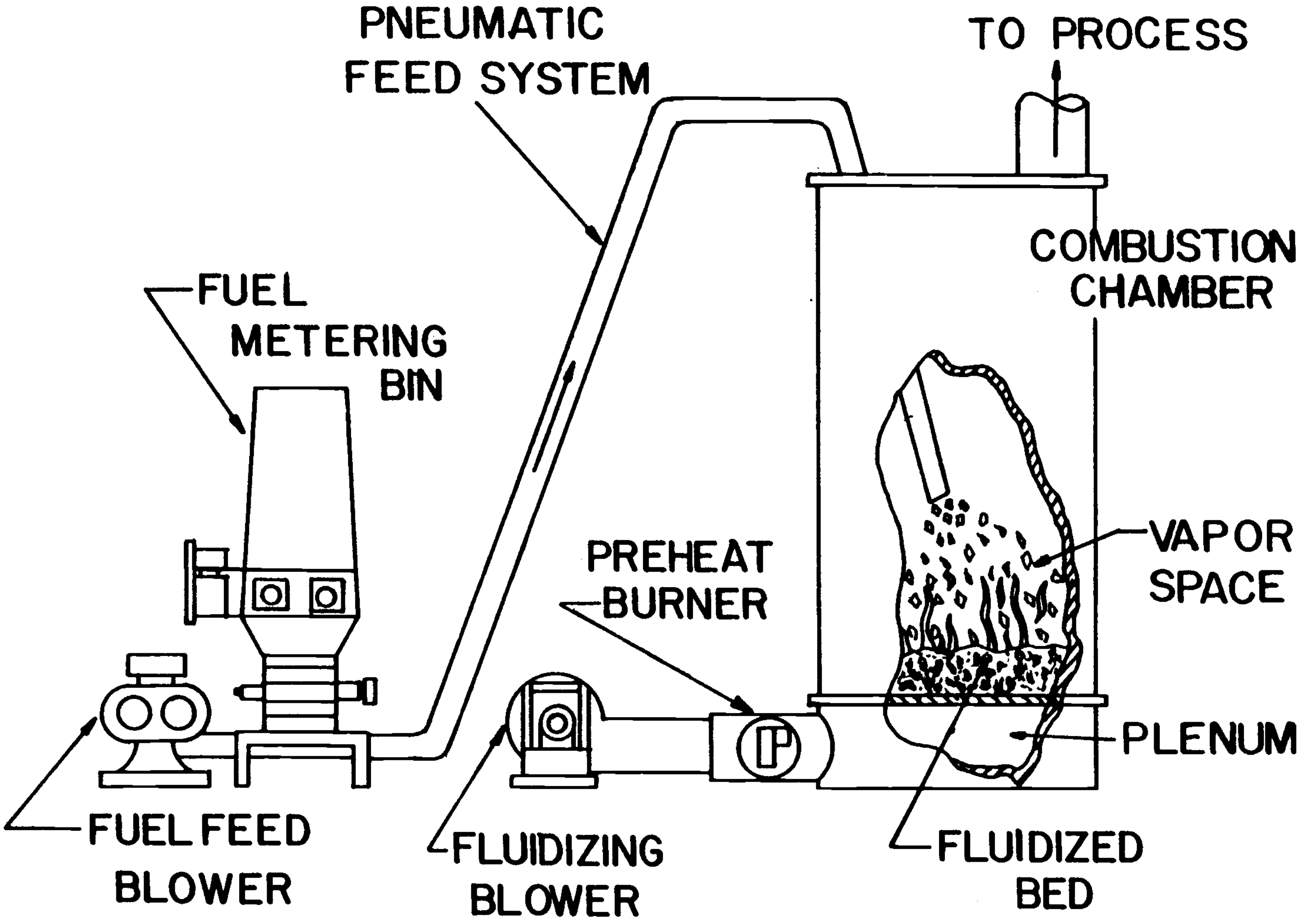

Figure 4.11. Fluidized bed combustor.

high heat and mass transfer rates resulting in better combustion of the fuel. In addition, the impact of inert material on the fuel particles abrades the surface char from the fuel; causing constant exposure of the fresh fuel, decreasing the amount of unburned fuel and, as a result, increasing the efficiency of the combustion.

In contrast with an inclined type burner in which drying, pyrolysis, and char combustion takes place in different regions, in a fluidized bed all combustion reactions take place essentially in the same zone - in the bed and immediately above it.

A fluidized bed combustor offers the least restriction as to type, moisture content, and size of the fuel particles. Mixed firing of wood and coal, or wood and lower grade fuels (such as waste materials) in a series or simultaneously is possible. Wood fuel particles of irregular size and shape with moisture content of up to 65 percent can be used with no preparation.

In a fluidized bed combustor optional bed cleaners, which permit cleaning of the bed while combustor is in operation, are available. This feature is particularly attractive in cases where continuous operation is important or fuel has a very high ash content.

A fluidized bed combustor with minor modification and regulation of combustion air supply can be used for pyrolysis or gasification as well.

The flame temperature in a fluidized bed for wood combustion is kept below 2000 F (usually between 1800 to 1500 F). This low temperature is necessary to prevent slagging of ash and to increase the refractory life.

The two most serious drawbacks of a fluidized bed combustor are its high initial cost and high power consumption.

Fluidized bed combustors have a capacity in the range of 10 to $140\text{x}10^6$ Btu/hr of fuel input. The important design parameter is heat release per bed volume, which is typically about 9000 to 12000 Btu/hr per cubic foot.

Although many of the combustion systems available today do not fall into the categories discussed above, they might have the characteristics of two or more different types of combustion systems discussed earlier. In the next section a few examples are given.

5- Lamb-Cargate Wet-Cell Burner

This burner is manufactured by Lamb-Cargate company of New Westminster, British Columbia, Canada, and is commonly known as a "wet-cell" burner. The wet-cell (shown in figure 4.12) consists of two chambers, one above the other. The functions of these chambers are described below:

Lower Chamber. In the lower or primary chamber fuel is dried and gasified, releasing volatiles for combustion in

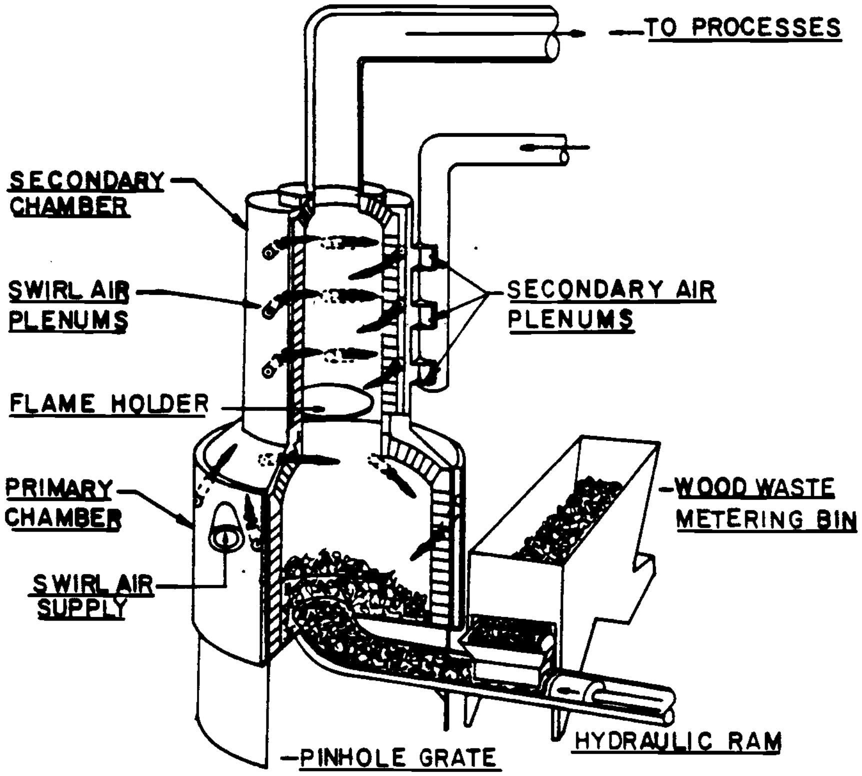

Figure 4.12. Lamb-Cargate wet-cell burner.

the upper chamber. The action of the lower chamber is therefore, gasification of the fuel. The undergrate air and overfire air, totaling less than the stoichiometric amount, are supplied to the lower chamber. This air which helps in drying the fuel also provides enough oxygen for char oxidation. Combustion of fixed carbon in the lower chamber provides the energy required for gasification and drying.

A series of high-pressure "swirl" air jets entering the lower chamber tangentially near the top of the chamber subject the hot gases leaving this chamber to a scrubbing action which separate the particulates in the gases. The cyclonic motion of the swirl air causes the impact of particulates and the furnace wall and, therefore, returns the particulates to the fuel pile. These particles are either burned (combustible particles) or they remain near the outer edges of the pile and are removed periodically (non-combustible particles such as ash and sand).

The amount of air supplied to the lower chamber is regulated so that the gas temperature leaving this chamber remains below 1600 F. This helps to prevent the fusion of ash and prolongs the life of the refractory.

Upper Chamber. In the upper or secondary chamber, which is cylindrical in shape and double-walled with an inner refractory lining, gases from the lower chamber are supplied with excess air (typically less than 15 percent) to complete the combustion. Pre-heating of combustion air is achieved by passing this air between the outer cell wall and the hot refractory. The pre-heated air is introduced near the top of the upper chamber through a series of tangential tuyeres, promoting a strong outer vortex of relatively cool air. The motion of this vortex causes a turbulent mixing of volatiles and air which result in better combustion. This method of pre-heating the combustion air and introducing it into the upper chamber also helps to cool the refractory and reduce particulate emissions.

In a wet-cell burner the fuel is fed with an underfeed stoker and is forced up into the center of the horizontal pinhole grate in the lower chamber where it forms a slightly conical pile.

The gas temperature exiting the wet-cell ranges from 1700 to 2300 F, depending on moisture content of the fuel. Higher temperatures (2300 to 2700 F) can be achieved using auxiliary heat input.

The range of moisture content of fuel for this burner is from 10 to 65 percent. The particle size of the fuel is to be less than 3 inches. The ideal fuel for this burner is fuel with a particle size of 1.5 inches and moisture content of 45 percent. Deviations from these design conditions will result in derating of Btu output. For example, if 50 percent of the fuel particles have a size of less than 0.25 inches, 25 percent derating results, and if all of the particles have a size of less than 0.25 inches, 50 percent derating of the design output results.

Temperature of inlet air is desired to be 60 F and can be as low as -40 F. The maximum inlet temperature is 400 F. It should be noted that since the cooling of the refractory is achieved by pre-heating the inlet air, use of high-temperature inlet air could result in operational difficulties and, therefore, is not recommended.

Typically, a Lamb wet-cell uses about 15 percent excess air. However, a blend box is usually provided to mix the high temperature gases from the wet-cell with cooler air. This helps to provide a steady output temperature and cooler gas where desired.

Control of the wet-cell is based on the amount of air supplied to the primary chamber, such that the exiting gases retain a temperature of about 1600 F, and the amount of air supplied to the secondary chamber to keep the output temperature steady.

Based on manufacturer's claim, wet-cell has an efficiency of approximately 95 to 97 percent with practically no unburned carbon. Radiation losses are reported to be about 2 to 5 percent. Stack emissions are typically less than 0.1 grains/SDCF, but could increase when burning fuels with high fines content at high rates. The Lamb wet cell operates at or near atmospheric pressure.

Table 4.1 shows the capacities and the approximate installed motor power requirements for various sizes of this burner. Actual power consumption is about 50 to 90 percent of installed motor power depending on firing rate and fuel conditions.

Table 4.1. Capacity and Power Requirement for Lamb-Cargate Wet-Cell Burner

CAPACITY (MM Btu/hr)	POWER REQUIREMENT (H.P.)
10	80
25	175
60	465
75	580
100	595

6- Wellons Cyclo-Blast Furnace

The burner shown in figure 4.13 is a cyclo-blast furnace and is manufactured by Wellons, Inc. of Sherwood Oregon. It is a three-stage pile burner in which fuel is continuously augered into the side of a high-temperature refractory tube forming a pile on the water-cooled grates in the primary chamber. The pre-heated

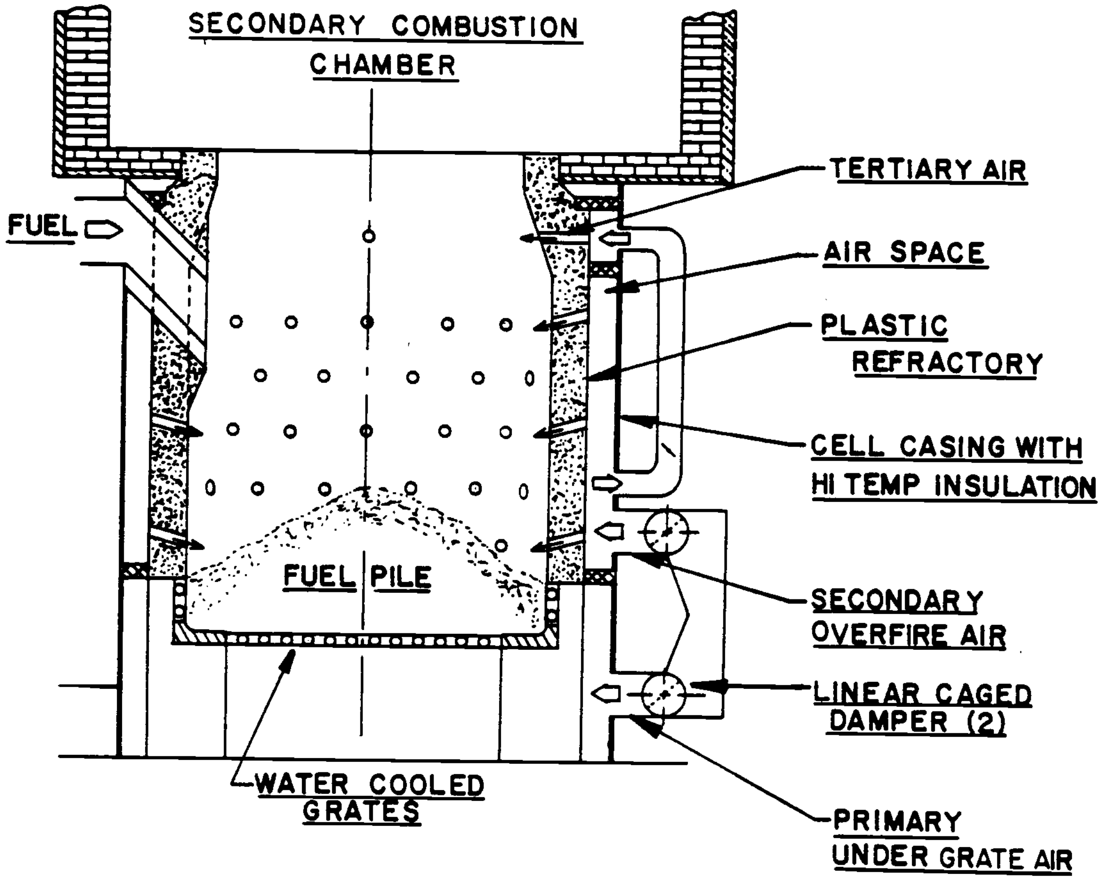

Figure 4.13. Wellons water cooled grate pile burner.

combustion air is introduced in three independently controlled zones.

Primary (underfire) air provides air for drying, gasification, and fixed carbon combustion. Primary air represents about 30 percent of the total combustion air. Use of pre-heated air presents no difficulty since the burner has water cooled grates.

Secondary (overfire) air, which enters tangentially through five stages of a refractory tube with a wall temperature of about 2000 F, is provided for volatiles combustion. Secondary air represents about 60 percent of the total combustion air.

Tertiary air, which is introduced at the top of the cell, completes the combustion and breaks up the cyclonic gas flow.

A large and final combustion chamber is also provided to allow enough time for complete combustion in temperature range of 1400 to 1700 F. To take advantage of radiant heat transfer, heat transfer surfaces (boiler tubes) are placed in this chamber.

This burner is designed to burn fuel with moisture contents up to 50 percent. Higher moisture contents result in derating of the Btu output of the burner. A maximum particle size of 3 inch is specified for use in this burner.

Response to load changes and turndown ratio in this burner is limited due to the pile burning design of the furnace [Drucker, 1984]. However, the manufacturer claims a turndown ratio of 5/1.

7- Roemmc Burner

Manufactured by the Guaranty Performance Company of Independence, Kansas, the Roemmc burner is a cyclone-suspension type burner. This burner (shown in figure 4.14) consists of two sections; a cyclone burner (three zoned ignitor) and a separator furnace. In the cyclone burner section, the fuel is mixed with air and ignited until it is mostly combusted. The remaining unburned fuel along with the ash and hot gases move on to the separator section, where the cyclonic action of gases helps with the ash discharge while the remaining unburned fuel (mostly particles with higher moisture content or larger size) is burned. The clean hot gases leave at the top of the separator furnace at a temperature of about 1800 F.

The Roemmc burner is designed for highly refined and dry fuel. The fuel is brought down to a size of 3 inches or less and is separated from stones using a disk screen and from iron using a magnet. Then, a hammer hog should break the fuel down to 1/2 inch or less. The next step is drying of the fuel to about 15 percent moisture content (maximum allowable moisture content is 20 percent), and

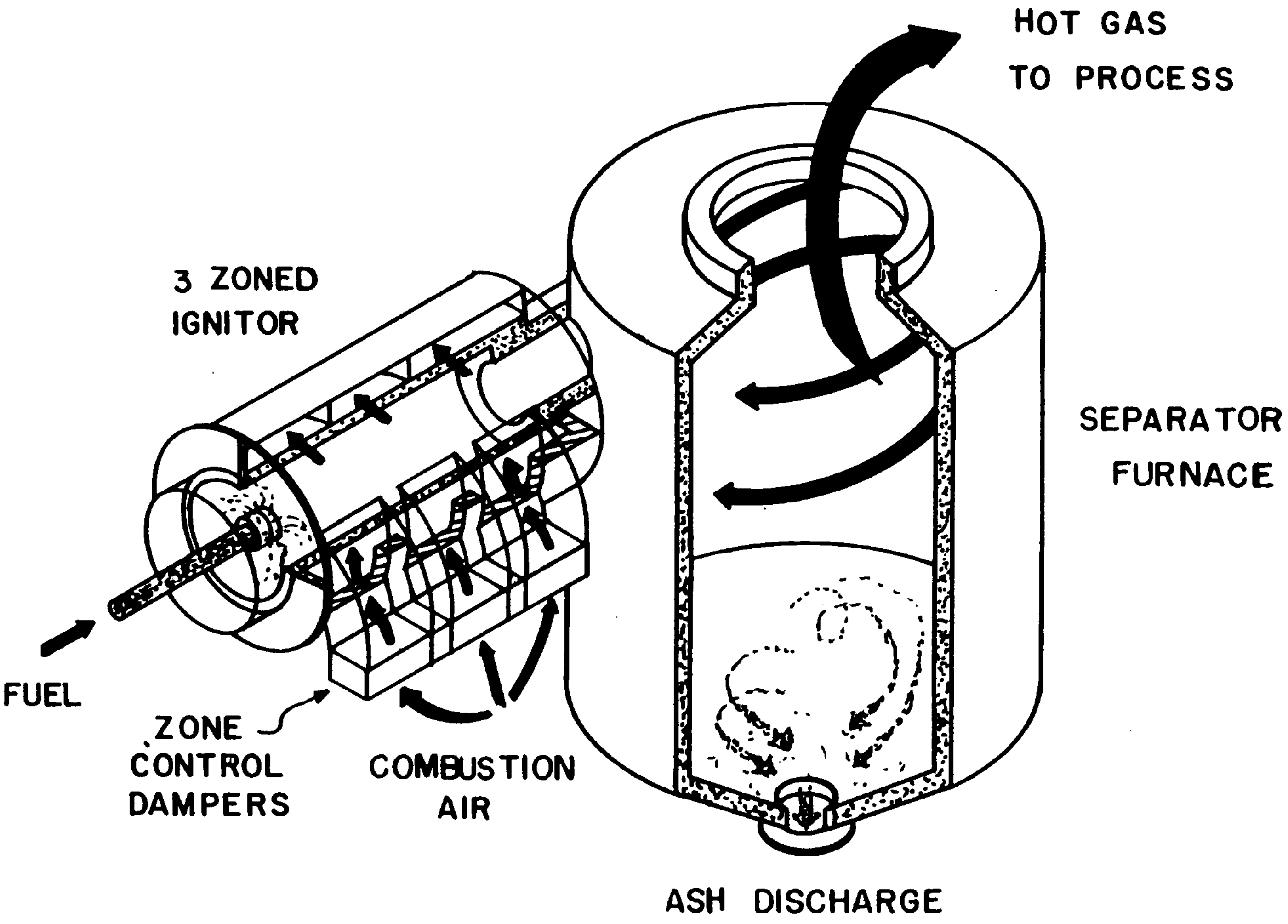

Figure 4.14. Roemmc cyclone-suspension burner.

then further reduced in size. The fuel particles before entering the burner should be able to pass through a 1/8 inch US seive and 50 percent of that has to be able to pass through a 40-mesh screen.

The fuel is pneumatically fed to the center of the burner where it strikes a diffuser plate which fans the fuel radially into the air stream.

Combustion air is supplied through control dampers, tangentially and slightly angled toward the exit of the burner. Pre-heating of combustion air, according to the manufacturer, does not improve the efficiency of the burner significantly. The amount of excess air used in this burner is close to 200 percent.

Control of the burner is achieved by controlling the fuel and the air fed to the burner.

The start up of the burner requires about 5 MM Btu/hr of fossil fuel input for a minimum of 1/2 hour per each start up.

Table 4.2 shows the capacity range of this burner and its power consumption. The power shown is for the combustion air blower only. In selection of this burner special attention should be paid to the power consumption of the preparation equipment such as dryer and hammer hogs.

Response to load for this burner is quite good due to its low thermal inertia.

Table 4.2. Capacity and Power Consumption for Combustion Air Blower of Roemmc Burner

SIZE (MM Btu/hr)	POWER REQUIREMENT (H.P.)
20	75
30	100
40	125
60	150

Chapter 5

COMBUSTION CALCULATIONS

Combustion of wood, as mentioned before, involves drying of the fuel, pyrolysis and combustion of volatiles, and fixed carbon oxidation. Wood particles in a furnace are exposed to intense heat transfer from the surroundings. Inside the solid particles this energy is transferred conductively, resulting in drying and further degradation of the solid. Pyrolysis of wood, therefore, is modeled by an unsteady state heat conduction equation provided with an Arrhenius type pyrolysis term. The same type of modeling, with some modifications, is used for char combustion. It has been shown, however, that although this treatment is justified in certain zones of time and space, there exist several secondary physico-chemical effects of energy and mass transfer that govern other zones of pyrolysis [Kanury, 1970]. This type of modeling, which employs the principles of conservation of mass and energy combined with the kinetics of combustion, is further complicated by the temperature and directional dependence of the thermal properties of wood.

The equations obtained from this type of modeling can be used in predicting the rate of pyrolysis. However, they do not give any clue as to chemistry of the produced volatiles or solid; and, similarly, the equations do not account for the effects of other variables such as supplied air (at least not explicitly).

These shortcomings, together with the difficulty in imposing the proper boundary conditions which exist in a furnace, make this type of analysis unattractive for calculations of wood combustion in a wood burner. However, it should be emphasized that analysis of this type is extremely helpful in the basic understanding of wood combustion.

The principles of chemical equilibrium give another method that can be used in predicting the adiabatic flame temperature and the composition of products of combustion. The basic assumption of this method is that chemical equilibrium in the furnace is achieved. Validity of this assumption depends on the residence time of flue gases in the furnace.

Calculation methods of this type require solutions to a series of non-linear equations. These equations relate the mole fraction, pressure, and equilibrium constants (function of temperature) of the species present in the flue gas. Solution of this set of equations usually requires an iterative method and also requires the use of a computer, especially if the number of species present is more than three or four.

Tillman and Anderson [1983] have used this technique to calculate the adiabatic flame temperature for a number of wood species. Equations 5.1 through 5.5 present part of the results of this study. These equations give the adiabatic flame temperature (in degrees Kelvin) as a function of moisture content and excess oxygen used.

$$DFT = 2405 - 8.7 * MC - 75.5 * EO \qquad (5.1)$$
$$YPT = 2406 - 9.4 * MC - 76.6 * EO \qquad (5.2)$$
$$PT = 2306 - 8.9 * MC - 70.9 * EO \qquad (5.3)$$
$$HT = 2290 - 9.1 * MC - 69.7 * EO \qquad (5.4)$$
$$BOT = 2268 - 9.3 * MC - 68.6 * EO \qquad (5.5)$$

The symbols used in the above equations are defined as follows:

DFT : Adiabatic flame temperature for Douglas fir.
YPT : Adiabatic flame temperature for yellow pine.
PT : Adiabatic flame temperature for poplar.
HT : Adiabatic flame temperature for hickory.
BOT : Adiabatic flame temperature for black oak.
MC : Percent moisture content, wet basis.
EO : Percent excess oxygen in the total stack gas, volume basis.

This method usually overestimates the flame temperature since no heat loss in the furnace is accounted for. In a furnace, flame temperature can have a substantial decrease from the adiabatic flame temperature due to energy losses such as radiation heat loss of the combustor. Also, as it was mentioned above, the condition of chemical equilibrium is a function of the residence time of the flue gases in the furnace , which varies for different types of furnaces, and it is not accounted for in this method.

A mathematical model for combustion of wood chips, using the combination of both methods mentioned earlier, has been developed by Giese and Leesly [1981]. In order to develop and solve the governing equations in this model, several simplifying assumptions have been made (such as plug-flow, zero percent moisture content, and cylindrical shaped particles). These assumptions, therefore, have drastically reduced the usefulness of the method. As the authors have indicated, this method

> "cannot from first principles determine the exact behavior of a furnace. Its usefulness lies, instead, in allowing experimental data to be correlated to provide information useful in optimizing of wood chip size, excess air setting and other combustion parameters."

It should be pointed out that this model deserves more attention than it is given here; it could, conceivably, be a starting point for a comprehensive study on the subject of wood combustion in a wood furnace.

To overcome the difficulties and shortcomings associated with the methods mentioned earlier, another approach has been chosen. This method, which will be discussed in the next few sections, is simple enough to be done by hand calculations. However, since part of the objective is to provide a computer program in the form of a subroutine to be used in conjunction with the power plant simulation program, details of the method are explained so that the reader can easily follow the computer program (given in Appendix A) as well.

It should be emphasized that the method used here neither considers the kinetics of wood combustion nor each wood particle individually; instead, it treats the combustor as a black box with fuel and combustion air entering and the products of combustion (flue gas and ash) leaving under steady state conditions.

The method explained hereafter is based on the principals of conservation of mass, the first law of thermodynamics, and the second law of thermodynamics. This chapter has been divided into the following sections:

1- Known values (input of the computer program)
    a- Combustion air
    b- Fuel input
    c- Combustor specifications
    d- Reference state

2- Conservation of mass

3- First law of thermodynamics

4- Second law of thermodynamics

1- Known Values (computer program input)

The known values for the calculations, which are also the input of the computer program, are as follows:

a- Combustion Air. The combustion air supplied to the combustor is assumed to have known mass flow rate, composition, temperature, and pressure. This stream is designated by the letter " a ". These values are described below (the notations and units used are given in parenthesis):

a. Mass flow rate of air (MRa, lb/sec).
b. Composition of combustion air given by mole fraction of its constituents:
    Mole fraction of carbon dioxide (yCO2a).
    Mole fraction of water (yH2Oa).
    Mole fraction of oxygen (yO2a).
    Mole fraction of nitrogen (yN2a).
    Mole fraction of carbon monoxide (yCOa).
    Mole fraction of argon (yARa).
c. Pressure of combustion air (Pa, inches of water).
d. Temperature of combustion air (Ta, F).

<u>b- Fuel Input</u>. The fuel input is considered to contain moisture and dirt. Composition and mass flow rate of wood on dry, clean (or bone-dry designated by "bdwd") basis as well as moisture content (wet basis) is assumed to be known. Furthermore, the fuel is assumed to be at ambient temperature (Tamb, F). These values are described below:

a. Mass rate of bone-dry clean wood (MRbdwd, lb/sec).
b. Fraction of moisture content of wood, on wet basis (xH2O, lb H2O/ lb wet fuel).
c. Dirt content, as a fraction of dry-clean weight (xdirt, pounds of dirt per pound of dry clean fuel).
d. Fuel composition from ultimate analysis. These values are expressed as a percentage of bone-dry weight of the wood and are listed below:
    Carbon content (xC, lb carbon/ lb bdwd).
    Hydrogen content (xH, lb hydrogen/ lb bdwd).
    Oxygen content (xO2, lb oxygen / lb bdwd).
    Nitrogen content (xN2, lb nitrogen / lb bdwd).
    Ash content (xash, lb ash / lb bdwd).
    Dirt content (xdirt, lb dirt/ lb bdwd).
e. Higher heating value of wood (HHV, Btu/ lb bdwd).
f. Temperature of fuel (Tamb, F).

c- Combustor Specifications. These values are different from one type of combustion system to another. These are:

a. Gas pressure drop through the combustor (dpburn, inches of water).
b. Amount of unburned carbon in fraction of weight of carbon in the fuel (xunbC, lb of unburned carbon/lb of carbon in the fuel).
c. Amount of carbon burned to carbon monoxide in fraction of weight of carbon in the fuel (xCCO, lb of carbon burned to CO / lb carbon in the fuel).
d. Radiation heat loss (radlos, Btu/sec).

In a combustor, as discussed earlier, the amount of carbon burned to carbon monoxide, unburned carbon, and radiation losses are functions of several variables such as moisture content of fuel; fuel particle size; amount, temperature, and location of excess air; flame temperature; and the type of combustion system. Therefore, in an ideal case one should be able to calculate the amount of radiation loss, unburned carbon, and carbon monoxide as a function of variables mentioned. For example, by running a series of tests over a range of moisture content of the fuel, the amount of carbon monoxide generated can be measured, and thus a relation between the two established. Using these relations (performance curves of the combustor), a detailed quantitative analysis of each combustion system and, therefore, comparison between different types of combustors with regard to changes in fuel and operating conditions would be possible. Unfortunately, of the combustor manufacturers who were contacted none provided or were able to furnish this type of information. A situation which leads to the conclusion that these studies have not been done or are not in the public domain.

From a consumer point of view this information is essential for conducting comparisons between different types of combustion systems. But, this is not the only reason for which these studies should be done. By using the results of these studies, design improvements and determination of optimum operating conditions and fuel preparation could also be achieved. It is, therefore, concluded that further research and study in this area is essential to the improvement of wood fuel combustion systems and the utilization of wood fuel as an alternative and efficient source of energy.

Almost all the manufacturers assume a constant for these values; for example, the amount of unburned carbon is assumed to be between 0 and 2 percent, the amount of carbon burned to carbon monoxide about 1 percent, and the radiation heat loss is taken to be about 2 to 5 percent of the capacity of the combustor.

d- Reference State. These values, which are referred to as dead state values, are used in the calculation of thermodynamic properties and the second law analysis. They are represented by the followings :

a. Dead state temperature (Tds, F)
b. Dead state pressure (Pds, Psia)
c. Dead state mole fraction of air:
    Mole fraction of carbon dioxide (ydsCO2)
    Mole fraction of water (ydsH2O)
    Mole fraction of oxygen (ydsO2)
    Mole fraction of nitrogen (ydsN2)
    Mole fraction of carbon monoxide (ydsCO)
    Mole fraction of argon (ydsAR)

2- Conservation of Mass

The conservation of mass principle is used to determine the mass flow rate of flue gas and ash exiting the combustor. In order to determine the composition of flue gas it is necessary to consider the chemical reactions taking place in the combustor.

Since the major contribution of NOx emission in a combustion process comes from the nitrogen in the fuel and at high temperatures; therefore, it is assumed that no nitrogen oxide is formed (in combustion of wood the temperature is relatively low and the amount of nitrogen and sulfur content of wood is negligible). Neglecting the reaction of nitrogen and sulfur with oxygen the reactions in the combustor can be considered to be:

$$C + O2 \longrightarrow CO2$$

$$C + 1/2\ O2 \longrightarrow CO$$

$$H2 + 1/2\ O2 \longrightarrow H2O$$

In a chemical reaction, such as shown in the equations above, reactants combine on a mole basis.

Therefore, after calculating the mass of each constituent entering the combustor, the number of moles of each constituent is calculated. Next, according to the above chemical equations the total number of moles of constituents of the products of combustion is calculated and, finally, converted back to the mass basis.

To determine the mass of each substance in the fuel the following calculations are done:

a. Mass rate of wet clean fuel (MRwcf, in lb/sec).

MRcwf = MRbdwd / (1 - xH2O)

b. Mass rate of moisture in the fuel (MRH2O, in lb/sec).

MRH2O = xH2O * MRwcf

c. Mass rate of dirt in the fuel (MRdirt, in lb/sec).

MRdirt = xdirt * MRbdwd

d. Mass rate of ash in the fuel (MRash, in lb/sec).

MRash = xash * MRbdwd

e. Mass rate of carbon in the fuel (MRcbn, in lb/sec).

MRcbn = xC * MRbdwd

f. Mass rate of carbon burned to CO (MRCCO, in lb/sec).

MRcco = xCCO * MRcbn

g. Mass rate of unburned carbon (MRunbC, in lb/sec).

MRunbC = xunbC * MRcbn

h. Mass rate of carbon burned to CO2 (MRCCO2, in lb/sec).

$MRCCO2 = MRcbn - MRunbC - MRCCO$

i. Mass rate of oxygen in the fuel (MRO2, in lb/sec).

$MRO2 = xO2 * MRbdwd$

j. Mass rate of hydrogen in the fuel (MRH2, in lb/sec)

$MRH2 = xH2 * MRbdwd$

k. Mass rate of nitrogen in the fuel (MRN2, lb/sec)

$MRN2 = xN2 * MRbdwd$

l. Total mass rate of wood to combustor (MRwdin, lb/sec).

$MRwdin = MRwcf + MRdirt$

The next step is to convert these values to mole rates. A detail of these calculations is given in Appendix A. Calculation of mole rate of carbon is given as an example. To calculate the mole rate of carbon (MLC, in lb-mole/sec) from the mass rate of carbon the following equation is used:

$$MLC = MRC / MWC$$

where MWC is the molecular weight of carbon.

To calculate the mass rates of constituents of combustion air (and mole rate), the following steps are taken:

a. Calculation of molecular weight of combustion air (MWa, in lb/lb-mole)

With the ideal gas assumption the molecular weight of combustion air is calculated as follows:

MWa = YARa*MWAR + YCOa*MWCO + YCO2a*MWCO2 +
YH2Oa*MWH2O + YN2a*MWN2 + YO2a*MWO2

where, for example, YARa is the mole fraction of the argon in the combustion air (note that the air stream is designated with "a") and MWAR is the molecular weight of argon.

From these mass rates, the mole rates of each constituent of the combustion air are calculated. For example, the mole rate of carbon monoxide is calculated as:

MLCOa = MRa*YCOa/MWa

Finally, the mole rate and mass rate of flue gas is determined. To do this the following steps are taken (the flue gas stream is designated by "c").

a. Mole rate of argon in flue gas (MLARc, lb-mole/sec).

MLARc = MLARa (since there is no argon present in wood.)

b. Mole rate of carbon dioxide in flue gas (MLCOc, lb-mole/sec).

MLCOc = MLCOa + MLCOb

c. Mole rate of nitrogen in flue gas (MLN2, lb-mole/sec).

$$MLN2 = MLN2a + MLN2b$$

d. Mole rate of carbon dioxide in flue gas (MLCO2c, lb-mole/sec)

$$MLCO2c = MLCO2a + MLCO2b$$

e. Mole rate of water vapor in flue gas (MLH2Oc, lb-mole/sec)

$$MLH2Oc = MLH2Oa + MLH2Ob$$

Note that MLH2Ob is the sum of the moles of water formed during combustion (from hydrogen of the fuel) and the moisture content of the fuel which is evaporated.

f. Mole rate of oxygen in flue gas (MLO2c, lb-mole/sec). This is equal to the sum of the oxygen in the fuel and in the air present minus the oxygen used for combustion.

$$MLO2c= MLO2a + MLO2b - (MLCOb/2 + MLCO2b + MLH2b/2)$$

After calculating the mole rates of the constituents of the flue gas, the mole fraction of the constituents is calculated. From these mole fractions, the molecular weight of the flue gas, in a manner similar to that of combustion air, can be calculated. Having the molecular weight and mole rate of flue gas, the mass rate of flue gas can be calculated. The equations used are:

g. Total mole rate of flue gas (sumMLc, lb-mole/sec)

$$\text{sumMLc} = \text{MLARc} + \text{MLCOc} + \text{MLN2c} + \text{MLCO2c} + \text{MLH2Oc} + \text{MLO2c}$$

h. Mole fraction of the constituents of flue gas

$$\text{YARc} = \text{MLARc} / \text{sumMLc}$$

$$\text{YCOc} = \text{MLCOc} / \text{sumMLc}$$

$$\text{YCO2c} = \text{MLCO2c} / \text{sumMLc}$$

$$\text{YN2c} = \text{MLN2c} / \text{sumMLc}$$

$$\text{YH2Oc} = \text{MLH2Oc}/ \text{sumMLc}$$

$$\text{YO2c} = \text{MLO2c} / \text{sumMLc}$$

i. Molecular weight of flue gas

$$\text{MWcp} = \text{YCOc*MWCO} + \text{YCO2c*MWCO2} + \text{YH2Oc*MWH2O} + \text{YN2c*MWN2} + \text{YO2c*MWN2} + \text{YARc*MWAR}$$

j. Mass rate of flue gas (MRc, in lb/sec)

$$\text{MRc} = \text{sumMLc*MWcp}$$

Excess Air and Air/Fuel Ratio. To calculate the amount of excess air (EA, percent), it is necessary to calculate the theoretical amount of air required for complete combustion. This is done as follows:

a. Theoretical mole rate of oxygen needed for complete combustion of wood (MLO2th, lb-mole/sec)

$$\text{MLO2th} = \text{MLCb} + \text{MLH2b}/2. - \text{MLO2b}$$

where MLCb is the mole rate of carbon in the fuel, MLH2b is the mole rate of hydrogen in the fuel, and MLO2b is the mole rate of oxygen in the fuel.

b. Theoretical mole rate of air required for complete combustion of fuel (MLath, lb-mole/sec)

MLath = MLO2th / YO2a

c. Theoretical mass rate of air (MRath, lb/sec)

MRath = MLath*MWa

d. Theoretical air-fuel ratio (AFth, lb of air/lb of dry fuel)

AFth = MRath / MRbdwd

e. Actual air-fuel ratio (AF, lb of air/lb of dry fuel)

AF = MRa / MRbdwd

f. Percent excess air used in combustion (EA)

EA = (MRa - MRath)*100/MRath

Note that the above calculations are based on the mass of bone-dry fuel.

3- First Law of Thermodynamics Application

The first law of thermodynamics is used to balance the energy input and output of the combustor. In using the first law, it is necessary to calculate the properties of gases such as enthalpy and entropy. These properties are calculated in a separate subroutine called GASPROP, which is provided in Appendix B.

The thermodynamic properties of a mixture of gases is obtained by the ideal gas mixture assumption and utilization of variable heat capacities (function of temperature). Subroutine GASPROP is a collection of functions, each one used for a property. Each one of these functions require temperature, mole fraction, pressure and reference state conditions. The reference conditions (dead state temperature, pressure and mole fraction) are given through the use of a common statement and, therefore, are not repeated each time that a function is called.

To give an example of how to use these functions, suppose the flue gas temperature Tc (in degrees F), pressure Pc (in psia), and mole fraction of its constituents (YCO2c, YCOc, YH2Oc, YN2c, YO2c, YARc) are known. Enthalpy (Hc) and entropy (Sc) of the flue gas are calculated in functions HGAST and SGASTP, respectively, by calling these functions and passing the variables required as follows:

```
    Hc = HGAST (Tc, YCO2c, YH2O, YO2c, YN2c, YARc, YCOc)
    Sc = SGASTP (Tc, Pc, YCO2c, YH2O, YO2c, YN2c, YARc,
         YCOc)
```

To simplify the explanations hereafter, properties are assumed to be calculated as mentioned above and the details of the calculations will not be presented. For a steady state process the first law of thermodynamics is simply stated as:

Energy Input = Energy Losses + Energy Output

Each part of the equation given above is calculated for the combustor and the details are given below:

I. Energy Input

a. Energy Input with Combustion Air. Since the temperature of air entering the combustor could be fairly high (for this plant between 500 to 800 F), a significant amount of energy is added to the combustor by the preheated combustion air. This amount of (heat) energy is calculated as follows:

$$Qa = MRa*(Ha - Hrefa)$$

where Qa is the amount of heat in combustion air (in Btu/sec), Ha is the enthalpy of combustion air (in Btu/lb, calculated at combustion air temperature and composition), and Hrefa is the reference enthalpy (in Btu/lb, calculated at a reference temperature such as dead state or ambient temperature).

b. Energy Input of the Fuel. Energy input of each pound of dry clean fuel is equal to its higher heating value. It should be pointed out that this is based on the assumption of complete combustion (all the carbon in the fuel forms carbon dioxide and all the hydrogen in the fuel forms water). In addition, it is assumed that the water formed is in liquid phase, and it does not account for the energy requirement for vaporization of the moisture content of wood. Therefore, if the products of combustion contain carbon monoxide and water vapor, the losses should be accounted for. The total energy input of the dry clean fuel is then equal to:

$$Qwd = MRbdwd*HHV$$

where Qwd is the energy input of the fuel (Btu/sec), MRbdwd is mass rate of bone-dry wood (lb/sec), and HHV is the higher heating value of the wood (Btu/lb).

Since the the fuel entering the combustor is assumed to be at ambient temperature (which is equal or close to reference temperature), there is no heat input due to temperature difference of the wood (or the moisture content of the wood).

The total energy input of the combustor is then calculated as:

$$Qin = Qa + Qwd$$

where Qin is the total energy input to the combustor (Btu/sec).

II. Energy Losses

The energy that is either lost to the environment or not used to increase the temperature of the flue gas is treated as an energy loss. Such losses include: radiation loss; energy used for evaporation of water formed during combustion; energy used for vaporization of moisture content of the fuel; energy loss due to generation of carbon monoxide (instead of carbon dioxide); energy loss due to loss of fuel (unburned carbon); and energy used to heat the dirt in the fuel. Note that the energy used for evaporation of water in the combustor is treated as a loss of energy (in order to find the temperature of the flue gas); however, this energy is not lost but, rather, is absorbed by the flue gas. This energy can be recovered if, in the process, condensation takes place. If condensation does not take place in the heat transfer equipment, and the flue gas is exhausted to the atmosphere without any condensation, then the energy used for evaporation of water is, in fact, a loss. Methods of calculating these losses are discussed below.

a. Radiation Loss. The amount of radiation heat loss is a function of combustor design and flame temperature. However, as it was mentioned earlier, here a constant value is assumed. Typical values of radiation loss are about 2 to 5 percent of the capacity of the combustor. The radiation heat loss is shown by symbol radloss (in Btu/sec), so

Qrad = radloss

where Qrad stands for radiation heat loss. For this power plant the design capacity of the combustor is 70 MM Btu/hr. Therefore, the radiation loss of the combustor is assumed to be about 350 Btu/sec.

b. Energy Loss Due to Unburned Carbon. For each pound of carbon that is burned to carbon dioxide, about 14,090 Btu of heat is released. Therefore, for each pound of carbon that is not burned (leaves the furnace as particulate or remains in the ash), this amount of energy is lost. So the energy loss due to unburned carbon is:

QunbC = MRunbC*14090

where QunbC is in Btu/sec.

c. Energy Loss Due to Formation of Carbon Monoxide. As mentioned above, each pound of carbon burned to carbon dioxide (CO2) releases 14,090 Btu of heat. Each pound of carbon monoxide (CO), when it goes through complete combustion to produce carbon dioxide (CO2), releases about 4340 Btu of energy. Therefore, for each pound of carbon monoxide generated there is a loss of 4340 Btu. The heat loss due to generation of CO is then calculated as:

QCCO = MRCCO*MWCO*4340

where QCCO is the total heat loss due to generation of CO (Btu/sec), MRCCO is the mass rate of carbon burned to CO, and MWCO is the molecular weight of CO. Note that the product MRCCO*MWCO gives the weight of CO generated during the combustion.

d. Heat Loss Due to Heating of the Dirt. The amount of dirt carried with the fuel into the combustor exits the combustor with the ash discharge at approximately the same temperature as the flue gas. The energy consumed to heat the dirt is a loss since it is not used in raising the temperature of the flue gas and is discharged. This energy loss is calculated as:

$$Qdirt = MRdirt*Cpdirt*(Tc - Tamb)$$

where Qdirt is the energy loss due to heating of the dirt (Btu/sec), Cpdirt is the heat capacity of dirt equal to 0.2 Btu/lb-F, Tc is the temperature of the dirt exiting the combustor (assumed to be the same as flue gas temperature, in degrees F), and Tamb is the ambient temperature (temperature at which dirt enters the combustor, in degrees F).

e. Energy Used to Vaporize the Water Formed During Combustion. The higher heating value is the amount of heat released during the complete combustion of fuel in which water formed during the combustion is condensed to the liquid phase. However, since the flue gas temperature in a combustor is much higher than saturation temperature of water (at combustor pressure), the water formed during the combustion (from hydrogen in the fuel) will be in the vapor phase. The energy used to vaporize this amount of water is not used to increase the temperature of the flue gas and, therefore, is treated as a loss.

The heat of vaporization of water at 60 degrees F (at 1 atm pressure, which is close to the operating pressure of most combustors,) is about 1,059 Btu per pound of water. Therefore, the energy required to vaporize the water formed during the combustion (in Btu/sec) is:

$$QH2 = (MLH2b*MWH2O)*1059$$

where QH2 is energy required to vaporize the water formed during the combustion (from hydrogen content of the fuel), MLH2b is mole rate of hydrogen in the fuel (which is equal to mole rate of water formed during the combustion), and MWH2O is the molecular weight of the water. Note that the product of MLH2b*MWH2O gives the mass rate of water formed during the combustion (in lb/sec).

The lower heating value one of the fuel (LHV1), as defined in Chapter 2, can now be calculated simply by subtracting the amount of energy required for vaporization of water formed during the combustion from the higher heating value, that is:

$$LHV1 = HHV - QH2/MRbdwd$$

where LHV1 is the lower heating value of the fuel in Btu per pound of dry fuel. QH2 is divided by MRbdwd to make the value of LHV1 per pound of dry fuel, so the comparison with HHV (which is also expressed per pound of dry fuel) would be possible without any further calculations.

f. Energy Required for Vaporization of Moisture in Wood.

Since the moisture in the wood is found in three different forms, to make the distinction between these three forms it is helpful to emphasize the definition of the terms "absorption" and "adsorption" as given by Stamm [1964].

"Absorption" is defined to be the mechanical take-up of a liquid by porous solid within the gross capillary structure of the solid as a result of surface tension forces. The energy required to vaporize an absorbed liquid is only slightly greater than required to vaporize the liquid in free form. On the other hand, the intimate take-up of a fluid from the vapor phase, or a solute from solution by a porous material, is called "adsorption". In most cases, this take-up is only one molecule thick (monomolecular) and rarely exceeds an average of ten molecules thick. Adsorption is accompanied by evolution of heat. Often the heat of adsorption is quite large.

The moisture in wood is found in three forms: water vapor, capillary water (also called free or absorbed water), and bound water (also called hygroscopic or adsorbed water) [Skaar, 1972]. Compared to water vapor outside the wood, these three forms are in different energy levels.

Water vapor in the air spaces in the cell cavities of the wood has the highest state of energy (almost equal to water vapor outside the wood). However, since the amount of water vapor in the wood is very small, it will be neglected.

Capillary or absorbed water in the cell wall cavities of the wood is in an energy level slightly lower than liquid water outside the wood (at the same temperature). This is due to the force of capillary-water attraction. Therefore, to vaporize this "absorbed" water, it takes slightly more energy than it does for vaporization of liquid water outside the wood. Because this difference is very small, it is also neglected. Thus, the heat of vaporization of capillary water is assumed to be equal to that of regular liquid water.

The moisture content of the wood at which all the capillary water has been evaporated but no water from the cell wall has been lost is termed the fiber-saturation point. At the fiber-saturation point the moisture content of the wood is about 28 percent of the dry weight of the wood [Junge, 1980]. At moisture contents below the fiber-saturation point the moisture in the wood is in the form of bound (adsorbed or hygroscopic) water. Bound water is still at a lower energy level than capillary water. Therefore, it takes more energy to evaporate this water than it does to evaporate free water. The amount of energy needed to vaporize the bound water is a function of the moisture content of the wood. Figure 5.1 shows the differential heat of vaporization of bound water in wood as a function of moisture content on a dry basis.

The curves in figure 5.1 have been converted to English units (heat of vaporization in Btu/lb of water vaporized, with moisture content given on a wet basis,) and then approximated by a polynomial. Furthermore, the curve for the differential heat of vaporization is integrated to give the heat of vaporization at a given moisture content. The details for calculating the total heat of vaporization are given below.

a. Heat of Vaporization of Free Water. For each pound of water vaporized at 60 degrees F (it has been attempted to base all the calculations on this temperature or dead state temperature which is chosen to be 59 F) about 1,059 Btu of energy is required. Therefore, if the total mass rate of water in the fuel is MRH2O (in lb/sec), the heat

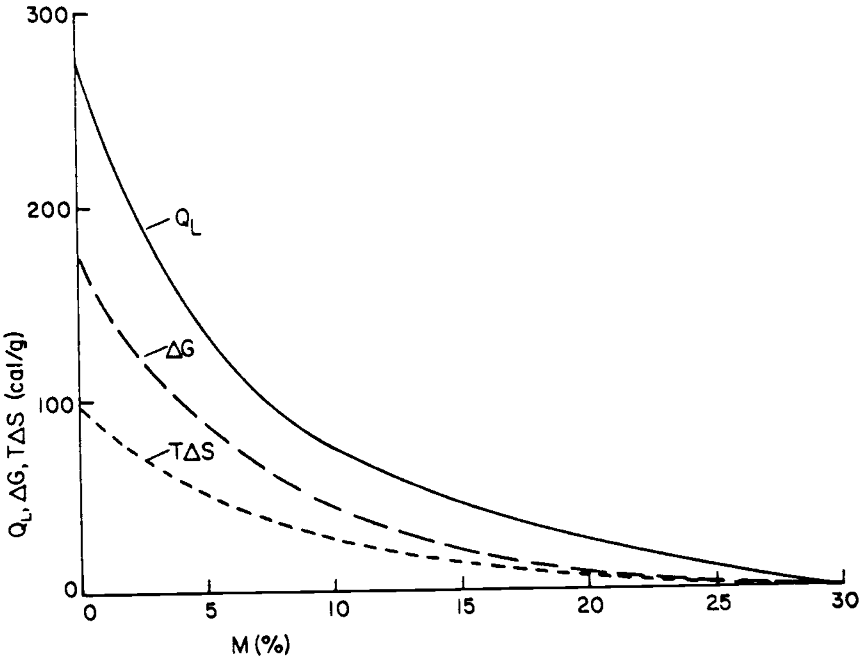

Figure 5.1. Curves of differential heat of sorption Q_1 (cal/g-water), free-energy change ΔG (cal/g-water), and of entropy change $T\Delta S$ (cal/g-water), when liquid water is taken up by wood at various moisture contents [Skaar, 1972].

of vaporization would be:

$$Qfw = MRH2O*1059$$

<u>b. Additional Energy Required to Bring the Bound Water to the Energy Level of Free Water</u>. This is the amount of energy required to bring the bound water to the energy level of free water (could be called the energy required to break the bounds). The energy to vaporize this water from free water level is already accounted for in item "a" above. If the mass rate of bound water is MRbw (in lb/sec), then:

$$Qbw = MRbw*\{(1/MC)*[4.679415E2*MC - 3.231411*10^{1}*(MC^{2}) + 1.040786667*(MC^{3}) + 4.68014*10^{-2} *(MC^{4}) - 6.588278*10^{-3}*(MC^{5}) + 2.569851667*10^{-4}*(MC^{6}) - 3.48937*10^{-6}*(MC^{7})]\}$$

where Qbw is the energy used to remove the bounds (Btu/Sec), and MC is the "percent" moisture content on wet basis. If the moisture content of the wood is higher than the fiber saturation point, MC would be equal to 23.08 which is the FSP. If the moisture content of the wood is less than FSP, then MC would be equal to the moisture content of the wood in percent wet basis.

Now the total energy required to vaporize the moisture content of the fuel and the energy to vaporize the water formed during the combustion is:

$$Qvap = Qfw + Qbw + QH2$$

where Qvap is the total heat of vaporization of the water in Btu/sec.

In order to calculate the lower heating value two (LHV2), as defined in Chapter 2, on dry basis, the following equation is used.

$$LHV2 = LHV1 - (Qbw/MRbw + 1059.)*xH2O/(1. - xH2O)$$

Total heat loss in the combustor. The total heat loss of combustion is simply obtained by adding all the heat losses mentioned above. Calling the total heat loss Qloss in Btu/sec, it would be equal to:

$$Qloss = Qrad + QunbC + QCCO + Qdirt + QH2 + Qvap$$

III. Energy Output

The flue gas leaving the combustor at high temperature is the only stream that carries useful heat outside the combustor. The heat output of the combustor is calculated as:

$$Qc = MRc * (Hc - Hrefc)$$

where Qc is the heat content of the flue gas (in Btu/sec); MRc is the mass rate of flue gas (in lb/sec); Hc is the enthalpy of the flue gas (calculated at flue gas temperature and composition, in Btu/lb); and Hc is the reference enthalpy (calculated at reference temperature and flue gas composition, in Btu/lb). Note that this is the sensible heat of the flue gas. The total energy absorbed by the flue gas can be calculated as:

$$Qflue = Qc + Qvap$$

This equation accounts for the energy used for vaporization of the moisture content of the wood and for the water formed during the combustion.

Calculation of Flue Gas Temperature

To this point there has been no mention of how to calculate the flue gas temperature (Tc). This has been done in order to simplify the explanation of the method described in this section.

The overall equation of energy balance can be written as:

$$Qin = Qloss + Qc$$

This equation can be written in expanded form:

$$Qwd + Qa = Qrad + QunbC + QCCO + Qdirt + Qvap + Qc \tag{5.6}$$

The left side of the equation is the sum of the energy input of the combustor. The right side is the sum of energy losses and output of the combustor. In this equation only Qdirt and Qc are functions of flue gas temperature. To calculate the flue gas temperature, an initial value for flue gas temperature is assumed. Then the value of Qdirt and Qc is calculated and checked to see if it satisfies equation 5.6. If the chosen temperature balances the equation, Tc is found; otherwise, a new Tc is calculated according to following steps:

a. Calculate new Qc from:

Qc = Qin - Qloss

b. Calculate new enthalpy of flue gas:

Hc = Hrefc + Qc / MRc

c. Calculate the new Tc:

Tc = TGASH (Hc, YCO2c, YH2O, YO2c, YN2c, YARc,YCOc)

where TGASH is a function in subroutine GASPROP that calculates the temperature of gas as a function of enthalpy and composition of the gas (see Appendix B).

The new Tc is used to re-calculate Qdirt and Qc, and if equation 5.6 is satisfied the iteration stops; otherwise, the process is repeated until a satisfactory result is obtained. This method converges rather fast, and it is neither complicated nor lengthy. Convergence is usually achieved after 3 to 6 iterations, depending on the initial guesses.

As mentioned in the beginning of this chapter, other quantities such as Qrad, Qunbc, and QCCO are, in fact, functions of flue gas (or furnace) temperature. However, since no such information was available, constant values have been assumed. It should be pointed out that implementation of such information does not pose any difficulty in the method used here.

First Law Efficiency. To point out the importance of the value of the efficiencies, two first law or thermal efficiencies are defined. These two efficiencies are called EFF11 and EFF12. In the next chapter it is explained why two efficiencies for first law and three efficiencies for the second law analysis are defined.

EFF11 is defined as:

$$\mathrm{EFF11} = \frac{\mathrm{Qflue} - \mathrm{Qa}}{\mathrm{Qwd}} * 100$$

And EFF12 is defined as:

$$\mathrm{EFF12} = \frac{\mathrm{Qflue} - \mathrm{Qa} - \mathrm{Qe}}{\mathrm{Qwd}} * 100$$

where EFF11 and EFF12 are in percent. Qe is the energy that is exhausted from the plant (or the process). It is the same as Qc but at exhaust temperature (in this work a temperature of 350 F is assumed). Therefore, Qe is calculated as:

$$Qe = MRc*(He - Hrefc) + Qvap$$

where He is the enthalpy of the exhaust gases (at 350 F).

4- Second Law Analysis

Analysis by the first law of thermodynamics is an essential part of processes in which energy transfer takes place. Based on conservation of energy, first law analysis accounts for input, output, and losses of energy in a process or system. The first law or thermal efficiency was defined as the ratio of output to input energy of the system. This definition, however, does not tell the whole story. It compares the output of the system to its input, without pointing out the real potential of the input energy. In addition, in the first law analysis each form of energy, such as thermal energy of the flue gas and the energy content of the fuel, is treated with equal value. To point out this deficiency consider, for example, a mixture of one pound of a fuel and sufficient air for combustion in an isolated container. If the mixture is at ambient temperature and pressure, the energy content of the container is equal to the higher heating value of the fuel. If the fuel is ignited and combustion is completed, the mixture in the container (products of combustion) will be at a higher pressure and temperature. According to the first law of thermodynamics, the container now contains the same amount of energy as it had before the combustion. However, the fact is that the energy content of the container in the first case (mixture of fuel and air) has more potential than the second case. Mixtures of fuel and air could be used in a variety of ways to produce work or other forms of energy; whereas, ways in which hot flue gas can be utilized are rather limited. Further, consider the container is placed in an environment that has the same temperature and pressure as products of combustion. In this condition the content of the container is not able to do work; whereas, the fuel-air mixture is still capable of doing useful work.

The concept of availability (also called exergy or available energy) derived from the second law of thermodynamics, on the other hand, helps to point out these differences as well as uncover the real deficiencies in many processes such as combustion. It should be emphasized here that, the first law remains an essential tool for analysis of all processes, and in some cases the second law analysis yields the same results as the first law analysis.

Availability (or exergy) of a system is defined as "a property which measures the maximum work which can be obtained from the system when it is allowed to come in equilibrium with the atmosphere" [Reistad, 1970].

Availability can be destroyed (or consumed); whereas, energy is not destroyed but conserved. Destruction of availability is called "irreversibility".

The state of a system when in equilibrium with the environment is called the dead state. Therefore, the availability of the dead state (or environment) is zero.

The choice of a dead state is similar to the choice of a datum for a thermodynamic property such as enthalpy or entropy. Different dead states give different values of availability of a system. The dead state is specified by thermostatic properties and chemical composition (it is assumed that kinetic and potential energy at dead state are zero). Three frequently chosen dead states, stated by Reistad [1970], are given here.

a. Each constituent of the system is in complete equilibrium with the environment:

$Tds = Tenv$ $\qquad$ $Pds = Penv$

$Ydsi = Yenvi$

where Tds, Tenv, Pds, Penv, Ydsi, Yenvi are temperature, pressure, and mole fraction of the ith species of the dead state and the environment, respectively.

b. The system is in thermostatic equilibrium with the environment:

   Tds = Tenv   Pds = Penv

c. Each constituent of the system is in thermostatic equilibrium with the environment:

   Tds = Tenv   Pds = Penv

   Ydsi = 1

When the difference in availability is of concern, the variation resulting from different dead states cancels out. Calculation of absolute availability (such as availability of a fuel or a chemical reaction), however, is affected by the choice of dead state.

In the absence of kinetic and potential energy differences for a steady state process, flow availability is defined as:

$$a = (H - Hds) - Tds(S - Sds)$$

where a is the availability of the flow (Btu/lb); H is the enthalpy of the flow (Btu/lb); Hds is the dead state enthalpy of the flow; S is the entropy of the flow (Btu/lb-R); Sds is the dead state entropy of the flow; and Tdsabs is the temperature of the dead state (deg. R).

In order to calculate the second law efficiency of combustion (also irreversibility or availability destruction), it is necessary to compute the availability of fuel, combustion air, and flue gas.

a. Availability of Combustion Air. The availability of combustion air is calculated as:

$$Aa = MRa\ [(Ha - Hds) - Tdsabs(Sa - Sds)]$$

where Aa is the availability of combustion air (Btu/sec), and Ha and Sa are the enthalpy and entropy of the combustion air, respectively. Note that Hds and Sds are calculated at dead state temperature and pressure with combustion air composition.

b. Availability of Flue Gas. Availability of the flue gas is calculated similar to that of combustion air but at flue gas temperature, pressure, and composition.

$$Ac = MRc\ [(Hc - Hdsc) - Tds(Sc - Sdsc)]$$

c. Availability of Fuel (Wood). Availability of fuel consists of two parts: thermomechanical availability and chemical availability. Since the fuel is assumed to be at ambient conditions when entering the combustor, the thermomechanical availability of the fuel is zero. To simplify the calculations, first the availability of dry fuel is considered, then the effect of moisture content on availability is discussed.

I. Availability of Dry Fuel. The change in Gibbs free energy (at standard state) for a process, such as combustion is defined as:

$$\Delta G = \Delta Hr - T\Delta Sr$$

where ΔHr is the heat of reaction (combustion); ΔSr is the entropy change of reaction; and T is the temperature of reaction. If the reaction is carried out at dead state temperature and pressure, the change in Gibbs free energy is equal (in magnitude) to the chemical availability of the fuel. For many fuels such as pure hydrocarbons, which a have relatively simple composition, this calculation is easily done. Values for availability of these fuels can be found in the literature. Although for more complicated fuels, such as wood and coal, the values of heat of reaction (higher heating value) are experimentally as well as theoretically determined, still there is a lack of information for the change of entropy values. This difficulty arises from the complexity of fuel composition and its wide variation. In many calculations, therefore, the value of availability of fuel is simply chosen as its higher heating value. The higher heating value, although very close to availability (within 10 percent), does not give the accurate value (since it neglects the change in entropy).

Shieh and Fan [1982] have formulated an empirical equation for calculation of availability of fuels with complicated composition such as coal and wood. For fuels that contain only carbon, hydrogen, nitrogen, oxygen, and ash, this equation is:

$$\mathrm{Abdwd2} = \mathrm{MRbdwd}\{ [340.124*\mathrm{xC} + 5.25*\mathrm{xN2} - 5996.25*\mathrm{xH2} + 1062.45*\mathrm{xO2} - 298.15(\mathrm{Sash})(\mathrm{xash})]*1.7998 - \mathrm{HHV} \} \quad (5.7)$$

where Abdwd2 is the availability of the dry wood (Btu/sec); xC, xN2, xH2, xO2, xash are mass fractions of constituents of the dry wood (from ultimate analysis); and Sash is the entropy of the ash. The value chosen for Sash is assumed to be equal to 0.17152 Kcal/Kg, with K being an average of the values of Sash for 12 types of coal [Shieh, 1982].

In order to make a comparison between the values of availability of the wood based on HHV and the value obtained from the above equation, two availabilities for wood are calculated. These are Abdwd1 and Abdwd2. Abdwd2 is defined above, and Abdwd1 is calculated as :

$$\mathrm{Abdwd1} = \mathrm{MRbdwd}*\mathrm{HHV} \quad (5.8)$$

II. Effect of Moisture Content on Availability of Wood.

The free water in wood does not effect the availability of the wood, since it is assumed that wood is at ambient temperature. In another words, the free water in wood is at dead state (or very close to it). However, the bound water is not at the same energy level as water at dead state conditions, and, therefore, the presence of bound water in wood adds to the availability of the wood (availability is always positive).

The first step in analyzing the effects of bound water on the availability of wood is to determine the equilibrium moisture content of the wood (EMC) at dead state conditions. EMC at dead state is the moisture content at which moist wood is in equilibrium with the dead state.

The equilibrium moisture content of wood is a function of several variables such as relative vapor pressure (humidity ratio), and temperature of environment, as well as, type of wood, extractives present in wood, and previous history of the wood.

Figure 5.2 shows the effect of relative vapor pressure and temperature on EMC (dry basis) for wood. Although other factors, mentioned above, also affect the EMC, these factors are not as significant as relative vapor pressure and temperature and, thus, would be neglected. At dead state conditions chosen in this work (59 F and 1 atm), the EMC is found to be about 11 percent (wet basis). Therefore, at dead state conditions the equilibrium moisture content of wood is 11 percent. It

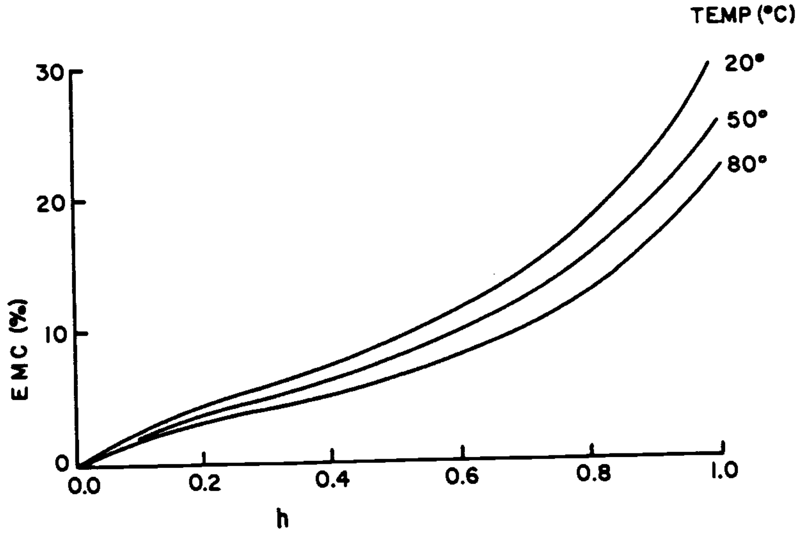

Figure 5.2. Sorption isotherms at three different temperatures showing EMC (%) versus relative vapor pressure h [Skaar, 1972].

should be noted that the EMC determined here is not necessarily the correct value for all dead state conditions but is the value found based on specific conditions chosen here as dead state.

Deviations in the EMC change the value of availability of wood. It is important to realize that this change always increases the value of availability of wood. To clarify this point, assume that one pound of dry wood (or with moisture content less than EMC) is placed in a container which is connected to the atmosphere through a duct in which a (miniature) turbine is located. Furthermore, assume that the turbine is a reversible one that does work with small flow from either side. For wood to gain an equilibrium with the environment (obtain a moisture content equal to EMC), it has to adsorb water. This means that some water vapor has to flow through the turbine. Thus the wood with a moisture content lower than EMC does work to bring the wood to a moisture content equal to EMC.

Now assume that the wood located in the container has a moisture content higher than EMC. To reach EMC under these conditions, some moisture in the wood has to flow out of the container. This means that some water vapor has to flow through the turbine causing the wood with a moisture content higher than EMC to do work to reach a moisture content equal to EMC. Therefore, if the moisture content of the wood is either higher or lower than EMC, to come in equilibrium with the environment (reach EMC) wood will do work; that is, it has a higher availability than wood at EMC. In order to calculate this increase in availability of the wood, the curve for Gibbs free energy of bound water, which is shown in figure 5.1, is approximated by a polynomial, and from this polynomial the change in availability of the wood is calculated (distance between bound moisture content of wood and EMC). This equation is given in Appendix D.

Finally, the overall availability of moist wood is calculated as:

$$Awdin1 = Abw + Abdwd1$$

$$Awdin2 = Abw + Abdwd2$$

where Abw is the availability of the bound water. Awdin1 and Awdin2 are availability of the wet wood based on HHV and equation 5.7 respectively.

Second Law Efficiency. For second law efficiency, three different efficiencies are defined. These are EFF21, EFF22, and EFF23. EFF21 is defined as:

$$EFF21 = \frac{Ac}{Awdin1 + Aa} *100$$

EFF22 is defined as :

$$\mathrm{EFF22} = \frac{\mathrm{Ac}}{\mathrm{Awdin2} + \mathrm{Aa}} \quad {*100}$$

EFF23 is defined as :

$$\mathrm{EFF23} = \frac{\mathrm{Ac} - \mathrm{Ae}}{\mathrm{Awdin2} + \mathrm{Aa}} \quad {*100}$$

where Ae is the availability of the exhaust at 350 F (calculated the same as Ac but at 350 F).

Chapter 6

RESULTS AND CONCLUSIONS

In this chapter the analysis of computer simulated results based on the computer model discussed in Chapter 5 are presented. These results are divided into two sections: (i) the affect of variables, such as moisture content, excess air, and combustion air temperature, on combustor performance; and (ii) the affect of moisture content on power plant performance. Only the affect of moisture content on the power plant performance is considered for analysis —since the mass flow rate of air going through the compressor, gas turbine, and the combustor, as well as, the temperature of air entering the combustor are determined by other components of the power plant (such as gas turbine, compressor and heat exchangers). and cannot be changed independently. For this power plant an air-fuel ratio of about 17 is used.

For this analysis the wood fuel chosen is assumed to be Douglas fir with an ultimate analysis as given in table 2.2.a. The higher heating value chosen is 8800 Btu/lb of dry wood. The radiation loss of the combustor is assumed to be 350 Btu/sec (about two percent of the capacity of the combustor which is 70 MM Btu/hr). It is also assumed that one percent of the carbon in the fuel is burned to carbon monoxide, and two percent of the carbon is not burned and leaves the combustor with ash. An important point to mention here is that the mass flow rate of the bone-dry wood to the combustor is constant, although the moisture content has changed. This means that the mass flow rate of wet fuel entering the combustor is changing as moisture content changes, but the mass rate of bone-dry fuel remains constant.

The exact values of the computer program input are given in Appendix C.

1- Combustor Performance

a- Flue Gas Temperature. Figure 6.1 shows the temperature of the flue gas exiting the combustor versus the excess air used in combustion, for three levels of moisture content. The temperature of combustion air entering the combustor is assumed to be 600 F. Recall that in this model the dissociation reactions of the constituents of the flue gas are not considered. As shown in this figure, the flue gas temperature is drastically reduced as the excess air is increased. For example, for wood with a 10 percent moisture content, the temperature of flue gas decreases from 3640 F at zero percent excess air (theoretical amount of air) to 1884 F at 200 percent excess air. For wood with a moisture content of 50 percent, the temperature ranges from a maximum of 2761 F to a

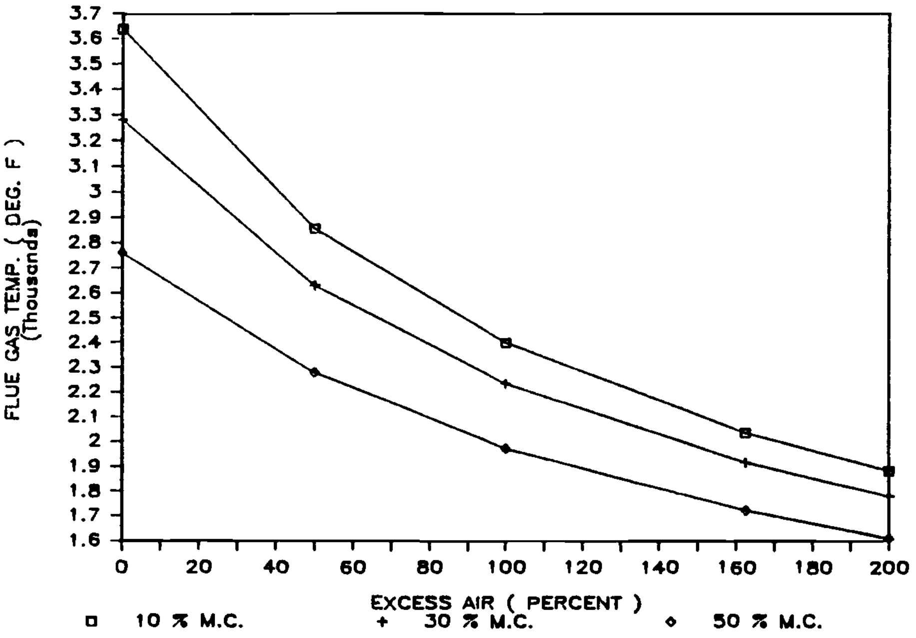

Figure 6.1. Flue gas temperature versus excess air for moisture contents of 10, 30, and 50 percent (combustion air at 600 F).

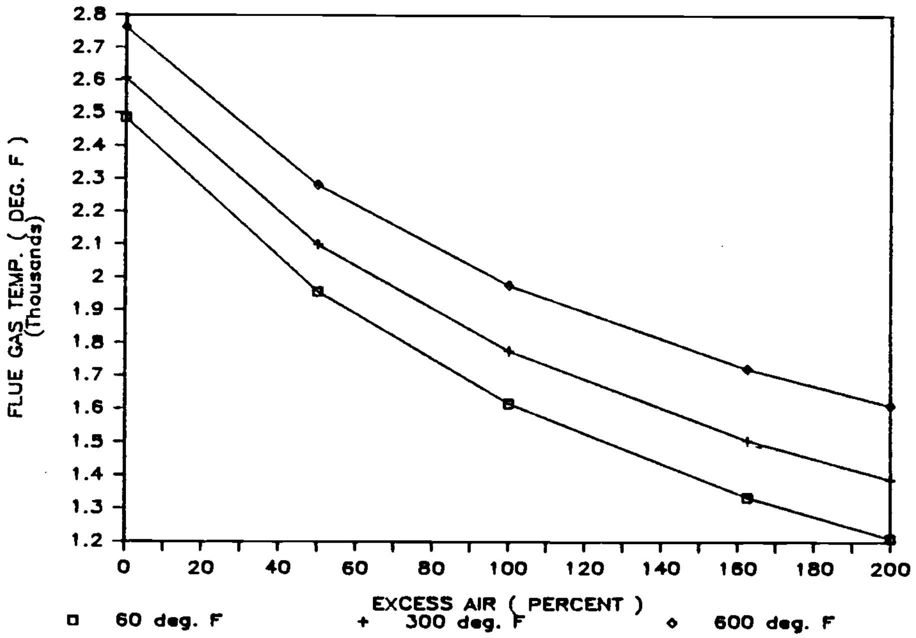

Figure 6.2. Flue gas temperature versus excess air for combustion air temperatures of 60, 300, and 600 F (50% moisture content).

minimum of 1609 F. This figure also shows that the reduction of temperature due to increase of excess air is more pronounced at lower levels of excess air and lower moisture content; whereas, at higher levels of excess air the slope of the lines are lower. This is an expected result since the limiting temperature of the flue gas (combustion with an infinite amount of excess air) is the temperature of the entering combustion air. This figure also shows that the combined effect of moisture content and excess air produces the lowest temperature at highest levels of moisture content and excess air.

Figure 6.2 shows the temperature of flue gas versus excess air for wood with a moisture content of 50 percent at three levels of entering combustion air temperature. Although this figure is similar to figure 6.1 in showing the effect of excess air on flue gas temperature, it is clearly shown that changing the combustion air temperature over its expected range does not effect the flue gas as drastically as changing the moisture content (throughout its range) does. For example, increasing the combustion air temperature from 60 F to 600 F increases the flue gas temperature from 2488 F to 2761 F at zero percent excess air, an increase of only 273 F.

Figure 6.3 shows the effect of moisture content on flue gas temperature at three levels of excess air with a combustion air temperature of 600 F. Here again it is

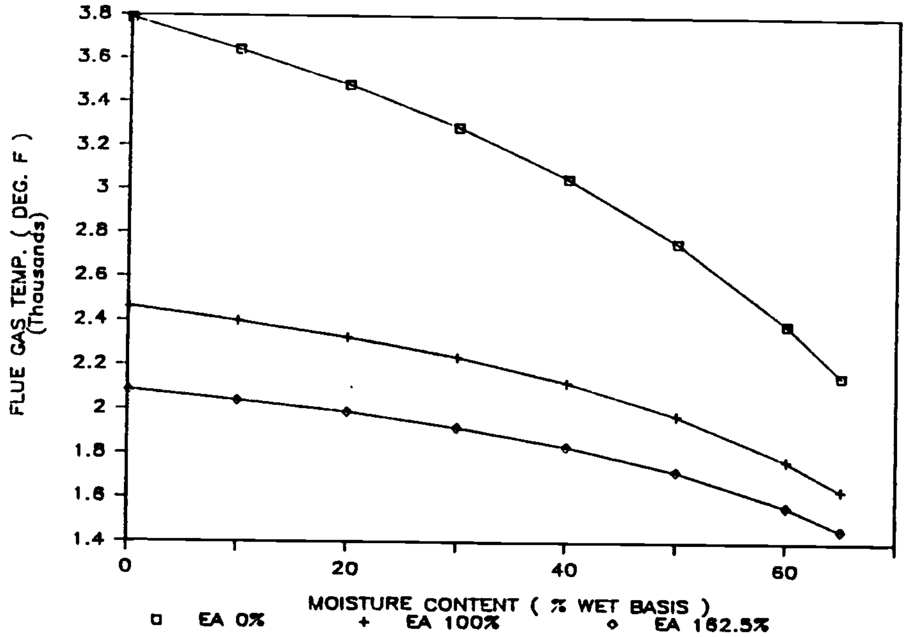

Figure 6.3. Flue gas temperature versus moisture content for excess air levels of 0, 100, and 162.5 percent (combustion air at 600 F).

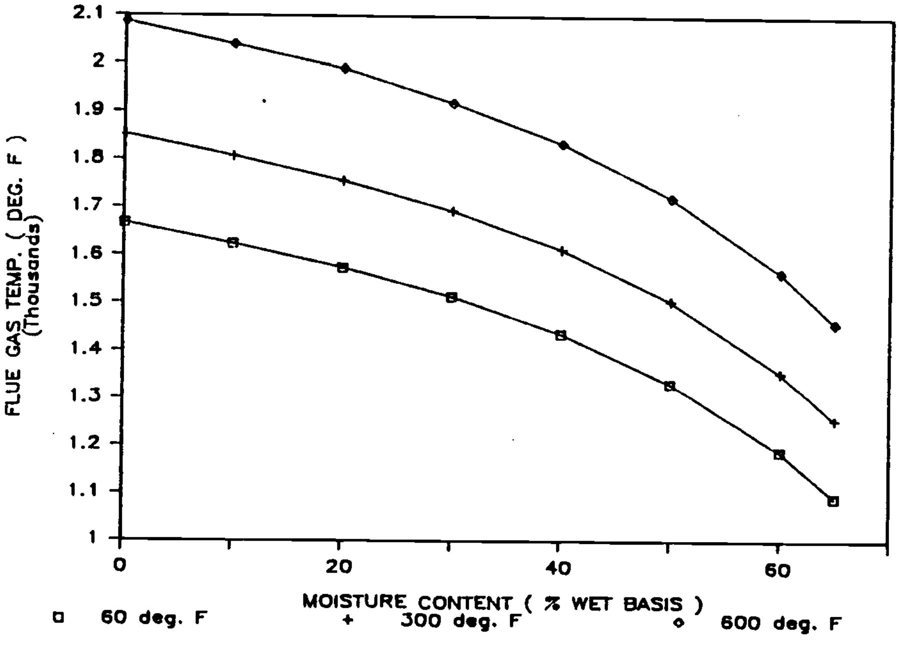

Figure 6.4. Flue gas temperature versus moisture content for combustion air temperatures of 60, 300, and 600 F (162.5 percent excess air).

shown that the flue gas temperature is reduced drastically as the moisture content is increased. For example, the flue gas temperature is reduced from 3785 F for dry wood to 2085 F for wood with a moisture content of 65 percent at zero percent excess air. Comparison of flue gas temperature at different levels of excess air shows that the excess air has a more pronounced effect than moisture content on flue gas temperature.

Figure 6.4 shows the temperature of flue gas as a function of moisture content at three different levels of combustion air temperature with 162.5 percent excess air.

From figures 6.1 through 6.4 it can be concluded that over the indicated range, the temperature of the flue gas is most effected by the amount of excess air used for combustion. The next most important factor is the moisture content of the fuel followed by temperature of the combustion air as the least important factor affecting the temperature of the flue gas. From these graphs one could determine the amount of excess air required for a specific flue gas temperature at a given moisture content and combustion air temperature. It is also worthwhile to note the effectiveness of utilizing the pre-heated air to increase the temperature of the flue gas.

<u>b- First Law Efficiency.</u> In defining the efficiency of a system or process, it is important to note the

effectiveness of the defined efficiency. To point out the importance of this concept several efficiencies (based on the first law and the second law of thermodynamics) have been defined in this work. For the first law analysis the two efficiencies defined are EFF11 and EFF12. EFF11 is given by the following equation:

$$\text{EFF11} = \frac{\text{Qflue} - \text{Qa}}{\text{Qwd}} *100$$

where Qflue is the energy absorbed by the flue gas; Qa is the energy of the combustion air entering the combustor; and Qwd is the energy of the wood entering the combustor. Note that Qflue is the sum of the sensible heat of the flue gas plus the heat of vaporization of water vapor. This value of Qflue is used because, although the heat of vaporization of water is not used in increasing the temperature of the flue gas, if somehow during the overall system process condensation takes place, this amount of heat would be recovered.

The flue gas after utilization is exhausted to the atmosphere. The exhaust gas is usually released at a temperature higher than that of the surroundings; therefore, this exhaust introduces a new loss for the system. For the purpose of calculations it has been assumed that the exhaust leaves the plant (or the combustor) at 350 F.

Therefore, another first law efficiency called EFF12 is defined as:

$$\text{EFF12}= \frac{\text{Qflue} - \text{Qa} - \text{Qe}}{\text{Qwd}} *100$$

where Qe is the energy of the exhaust gas and is calculated the same as Qflue (sensible heat plus the heat of vaporization of water) at 350 F.

Figure 6.5.a and 6.5.b show the first law efficiency as a function of excess air for three values of moisture content with a combustion air temperature of 600 F. As can be seen from these two figures, EFF11 is almost constant over the range of excess air while EFF12 reduces as excess air is increased. EFF11 is about 95.5 over the range of excess air from zero to 200 percent. It also does not change much as moisture content changes. The reason is that this efficiency only takes into account the radiation losses, losses due to unburned carbon, and the losses due to carbon monoxide production, which are assumed to be constant. EFF12, on the other hand, reduces from 80.5 percent to 70.3 percent as excess air is increased from zero to 200 percent. This is because an increased mass flow rate of combustion air causes an increased mass flow rate of exhaust gases which carry the energy out.

Figure 6.6.a and 6.6.b show the first law efficiency

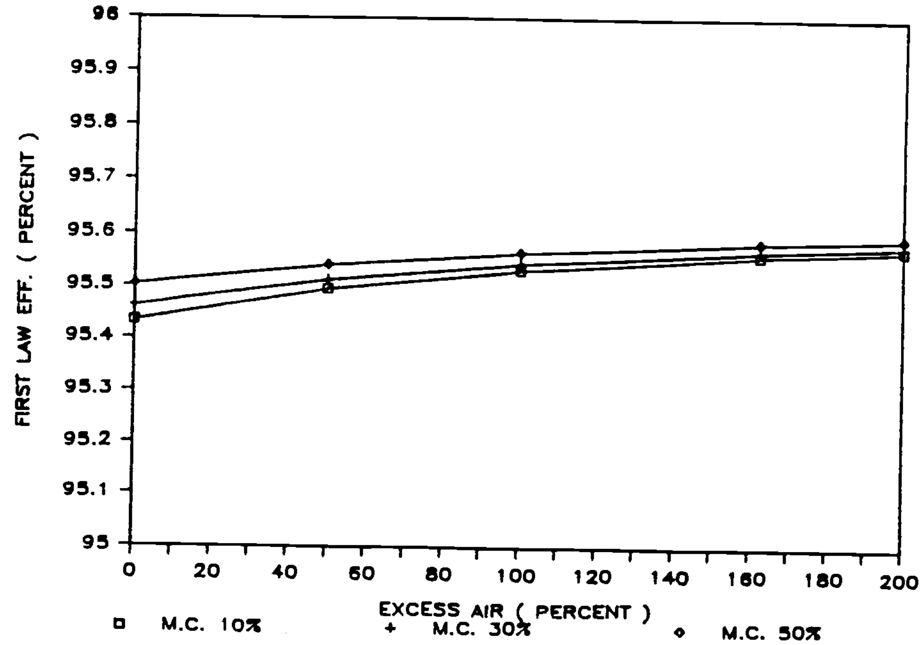

Figure 6.5.a. First law efficiency (EFF11) versus excess air for moisture contents of 10, 30, and 50 percent (combustion air at 600 F).

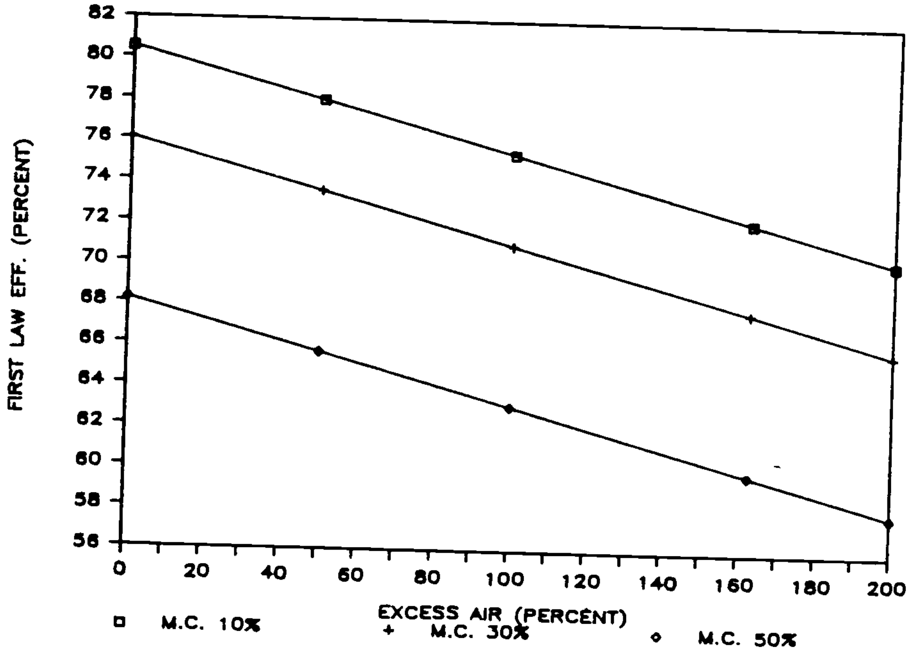

Figure 6.5.b. First law efficiency (EFF12) versus excess air for moisture contents of 10, 30, and 50 percent (combustion air 600 F).

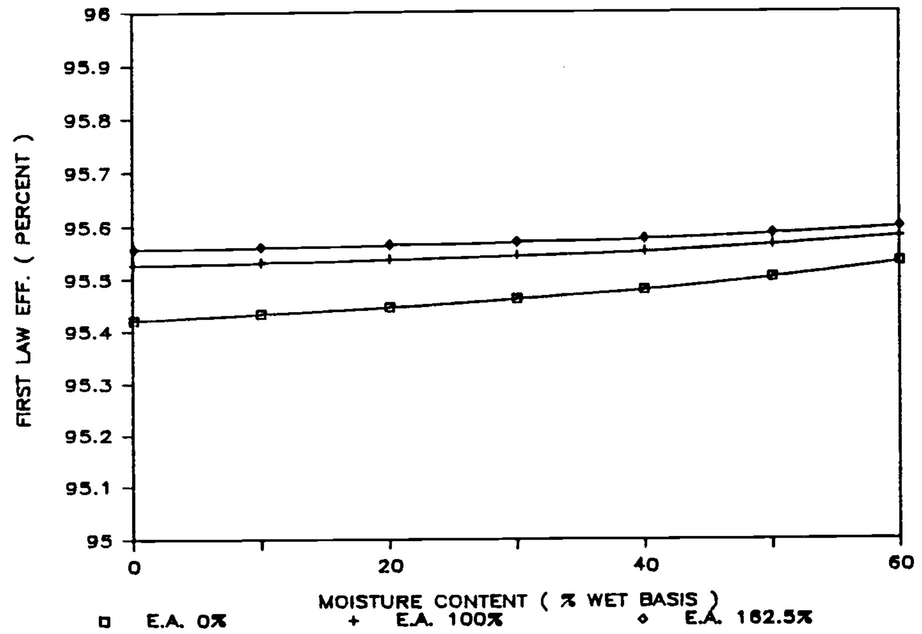

Figure 6.6.a. First law efficiency (EFF11) versus moisture content for excess air levels of 0, 100, 162.5 percent (combustion air at 600 F).

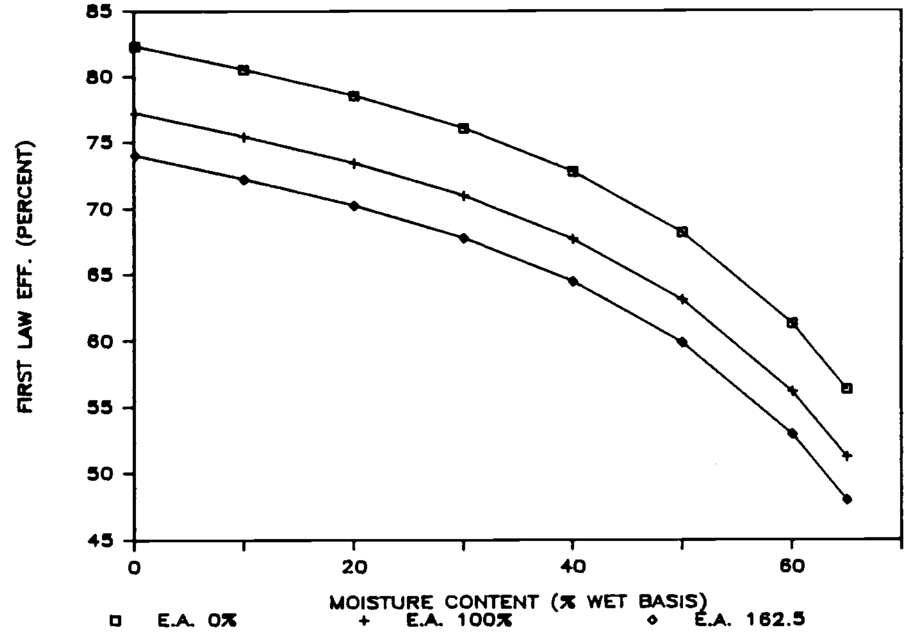

Figure 6.6.b. First law efficiency (EFF12) versus moisture content for excess air levels of 0, 100, 162.5 percent (combustion air at 600 F).

versus wood moisture content for three levels of excess air. Here again, EFF11 remains almost constant over the range of moisture content (about 95.5%), and it does not show much sensitivity to excess air variations. However, EFF12 does show a decline in efficiency due to the increase in moisture content as well as excess air. For example, EFF12 is reduced from 82.3 to 56.3 percent when moisture content is increased from zero to 65 percent for zero percent excess air, and from 74.1 to 48 percent for 162.5 percent excess air (air/fuel ratio of 17). This decrease shows the loss of energy as a result of exhausting the flue gas at elevated temperature.

Figures 6.7.a and 6.7.b show the first law efficiency versus combustion air temperature for three levels of moisture contents. Here both EFF11 and EFF12 remain essentially unchanged over the range of combustion air temperature. It should be noted that EFF11 has a value of about 95.5 percent; whereas, EFF12 is about 72, 68, and 60 percent for the moisture content of 10, 30, and 50 percent, respectively.

From figures 6.5 through 6.7 it can be concluded that EFF11 is not a good indicator of efficiency of the process. EFF12 is the better comparison tool. It is also concluded that based on values of EFF12 over the range of the variables studied the most important criteria for

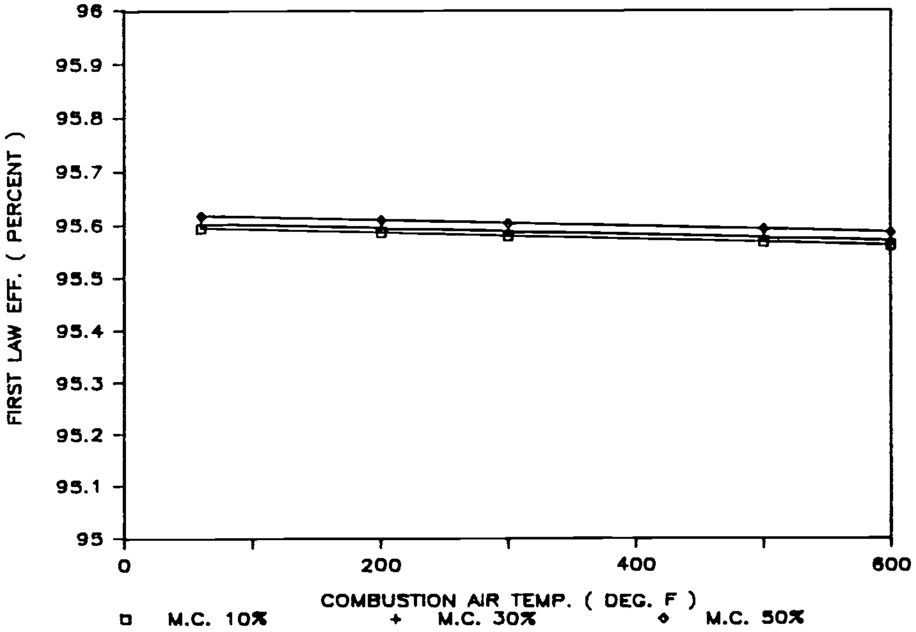

Figure 6.7.a. First law efficiency (EFF11) versus combustion air temperature for moisture contents of 10, 30, and 50 percent (162.5 percent excess air).

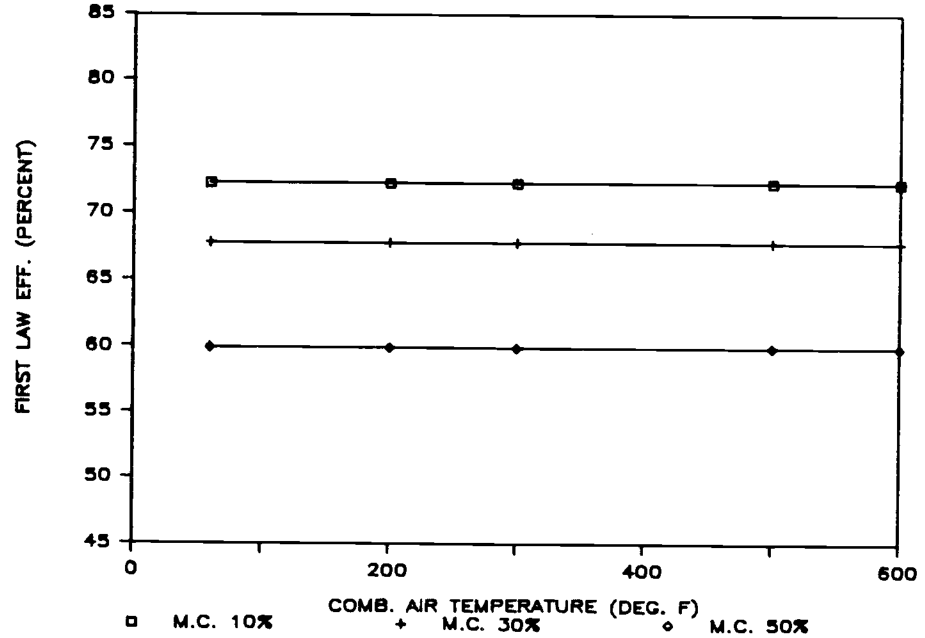

Figure 6.7.b. First law efficiency (EFF12) versus combustion air temperature for moisture contents of 10, 30, and 50 percent (162.5 percent excess air).

efficient combustion is the amount of excess air used for the combustion. Efficiency of the combustion reduces as excess air is increased. Moisture content of the fuel also has a substantial effect on the efficiency of the process. It can also be concluded that the temperature of the combustion air does not affect the first law efficiency as modeled here.

It is clear from the results presented here that EFF11 overestimates the value of the first law efficiency because it does not account for the exhaust losses. EFF12, on the other hand, does take into account the exhaust losses and is a better indicator of the process efficiency when these losses are sure to occur.

<u>c- Second Law Efficiency</u>. There are three second law efficiencies defined in this work; EFF21, EFF22, and EFF23. EFF21 is defined as:

$$EFF21 = \frac{Ac}{Awdin1 + Aa} * 100$$

where Ac is the availability of the flue gas; Awdin1 is the availability of the wet wood based on higher heating value; and Aa is the availability of combustion air.

EFF22 is defined as:

$$EFF22 = \frac{Ac}{Awdin2 + Aa} * 100$$

where Awdin2 is the availability of wet wood based on the empirical equation 5.7.

EFF23 is defined as:

$$EFF23 = \frac{Ac - Ae}{Awdin2 + Aa} * 100$$

where Ae is the availability of exhaust at an assumed temperature of 350 F (availability of flue gas at 350 F).

Figures 6.8.a and 6.8.b show the second law efficiencies versus excess air for different levels of fuel moisture content at a combustion air temperature of of 600 F. In figure 6.8.a, EFF21 is shown only for 50 percent moisture content. Since the value of EFF21 essentially follows the same pattern as EFF22, only one plot of EFF21 is shown here. Note however that using the higher heating value of the fuel as its availability gives a higher value of efficiency than using the value obtained from the empirical equation 5.7. This is because the availability of the fuel is different (higher) from its higher heating value.

Figures 6.8.a and 6.8.b show that the second law efficiency decreases as excess air increases, but this decrease is not as drastic as the first law efficiency (EFF12). Also note that the second law efficiency values are much lower than the first first law efficiency values obtained for the same conditions.

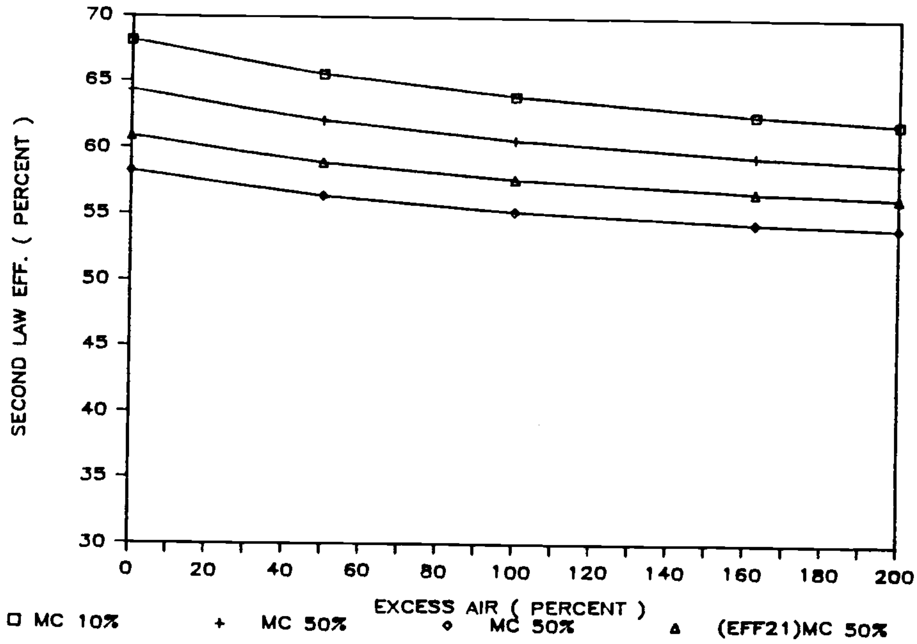

Figure 6.8.a. Second law efficiency (EFF22) versus excess air for moisture contents of 10, 30, and 50 percent (combustion air at 600 F). EFF21 is shown for 50% MC only.

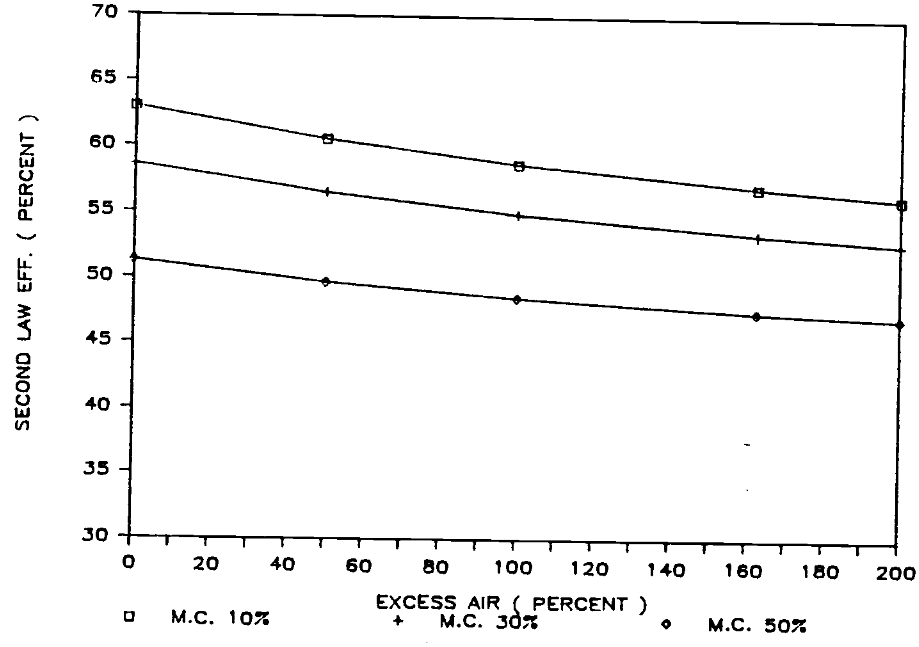

Figure 6.8.b. Second law efficiency (EFF23) versus excess air for moisture contents of 10, 30, and 50 percent (combustion air at 600 F).

Figures 6.9.a and 6.9.b show the second law efficiencies versus moisture content of the fuel for three levels of excess air with a combustion air temperature of 600 F. Both figures show that the second law efficiency reduces as moisture content increases. The interesting point to note is that the second law efficiency values show more sensitivity to moisture content of the fuel than the amount of excess air used for combustion.

Figures 6.10.a and 6.10.b show the effects of combustion air temperature on second law efficiencies for three levels of moisture content for combustion with 162.5 percent excess air. In contrast to the first law efficiencies that show very little variation due to changes of combustion air temperature, second law efficiencies increase substantially as combustion air temperature increases. This can be explained by noting that the irreversibility of mixing for two streams of gases decreases as the temperature difference between the two streams become smaller.

From the study of second law efficiencies over the range of variables considered here one could conclude that the most important factor in improving the second law efficiency of wood combustion is to reduce the moisture content of the fuel. The next important factor is the temperature of the combustion air. The higher the

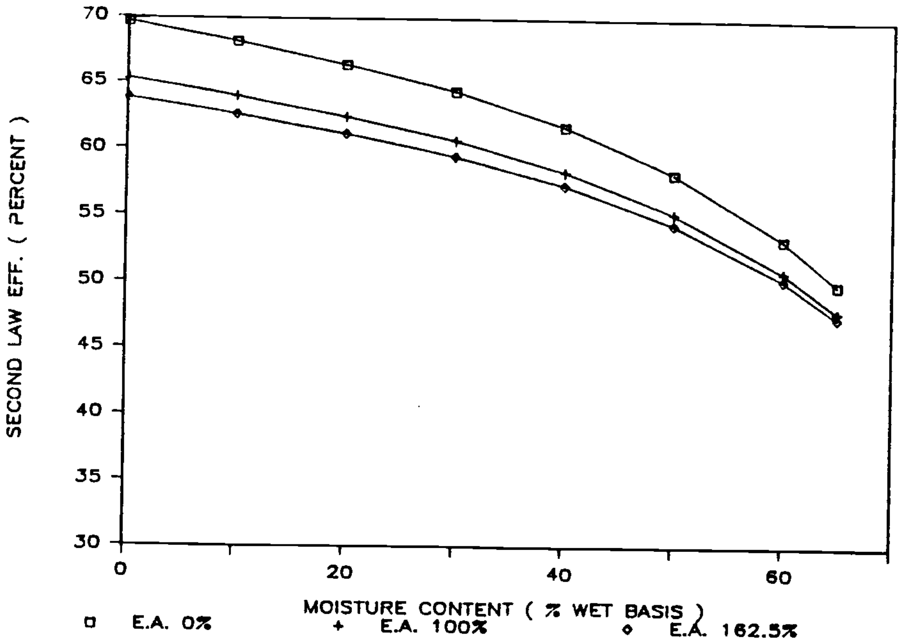

Figure 6.9.a. Second law efficiency (EFF22) versus moisture content for excess air levels of 0, 100, and 162.5 percent (combustion air at 600 F).

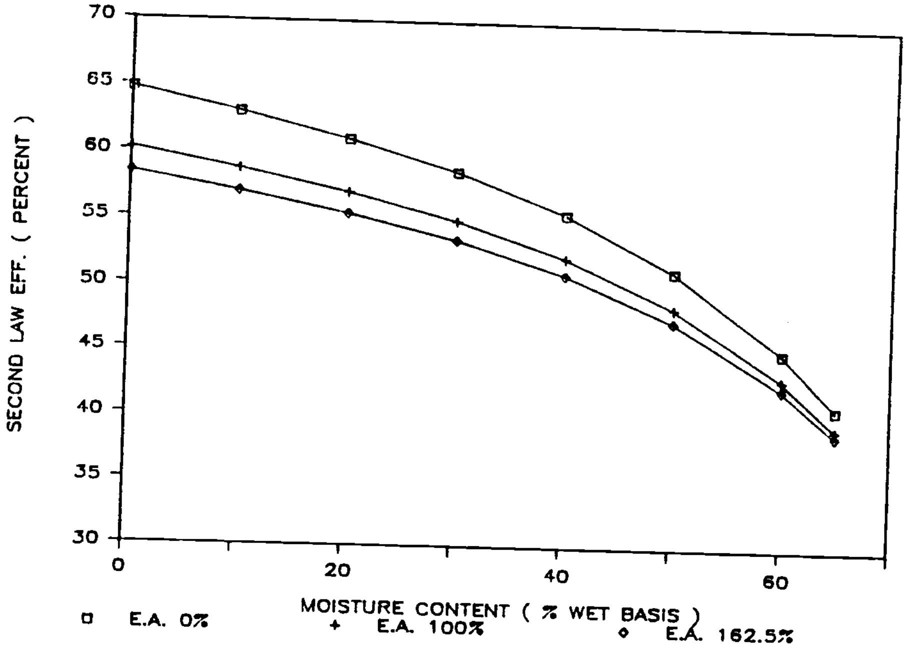

Figure 6.9.b. Second law efficiency (EFF23) versus moisture content for excess air levels of 0, 100, and 162.5 percent (combustion air at 600 F).

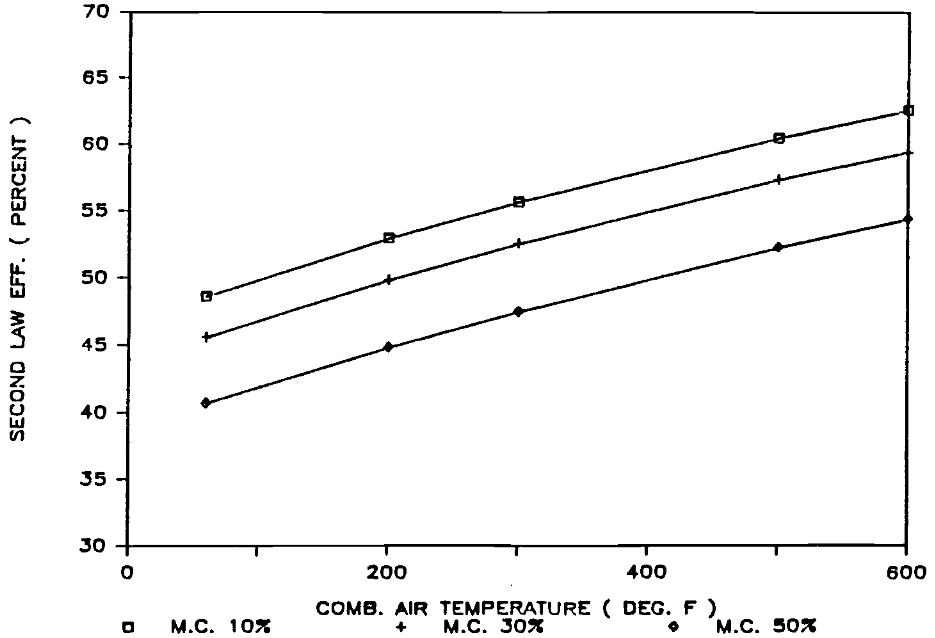

Figure 6.10.a. Second law efficiency (EFF22) versus combustion air temperature for moisture contents of 10, 30, and 50 percent (162.5 percent excess air).

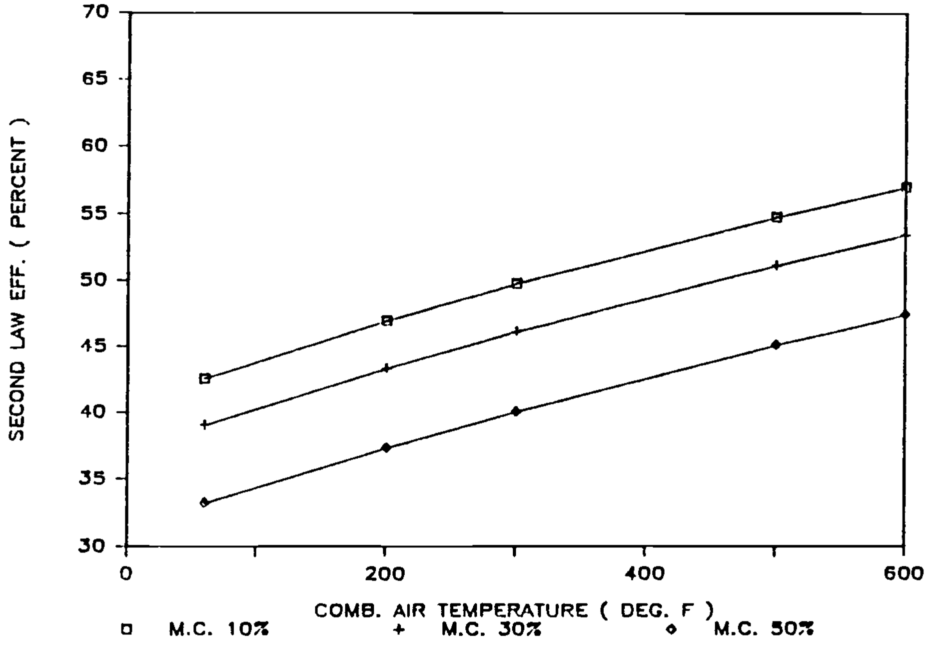

Figure 6.10.b. Second law efficiency (EFF23) versus combustion air temperature for moisture contents of 10, 30, and 50 percent (162.5 percent excess air).

temperature of the combustion air the more efficient the process. It is also concluded that the changes in the amount of excess air is the least important factor in second law efficiency considerations. In short, it can be concluded that the factors that reduce the flue gas temperature reduce the second law efficiency.

Among the three efficiencies (EFF21, EFF22, and EFF23) used in this analysis, EFF23 is the most meaningful efficiency since it is calculated based on true value (or at least is closer to the real value) of the availability of the fuel. It also accounts for the losses introduced by exhausting the flue gas at a temperature higher than that of the surroundings. EFF21 is based on higher heating value of the fuel (which is sometimes used as availability of the fuel) and, therefore, underestimates the availability of the fuel. EFF21 and EFF22 do not account for the exhaust losses of the combustion process and, therefore, overestimate the value of the second law efficiency.

2- Power Plant Performance

Performance of this power plant is evaluated by considering two efficiencies: first, the gross efficiency of the plant based on lower heating value two (LHV2) denoted by EFF12g; and, secondly, the net efficiency of the plant based on higher heating value denoted by EFFHn.

EFF12g is defined as:

$$\text{EFF12g} = \frac{\text{gross power output}}{\text{net energy input}} *100$$

where gross power output is the power generated by the gas and steam turbines minus the work of the compressor. The energy input is the lower heating value two of the fuel times the mass rate of dry fuel.

EFFHn is defined as:

$$\text{EFFHn} = \frac{\text{net power output}}{\text{gross energy input}} *100$$

where net power output is the gross power output minus the auxiliary power required by the components of the power plant; and gross energy input is the mass flow rate of dry wood times its higher heating value.

Figure 6.11 is a plot of gross efficiency of the power plant versus moisture content of the fuel for systems 2, 5, and 7. Gross efficiency of system 7 is higher than the other systems because of the utilization of the trimburner and the resulting higher inlet temperature of the gas turbine. Note that as moisture content of the fuel increases, LHV2 decreases, and, for constant power output, this will lead to an increase in EFF12g. The change in EFF12g over the expected range of moisture content of the fuel (35 to 60 percent wet basis) is less

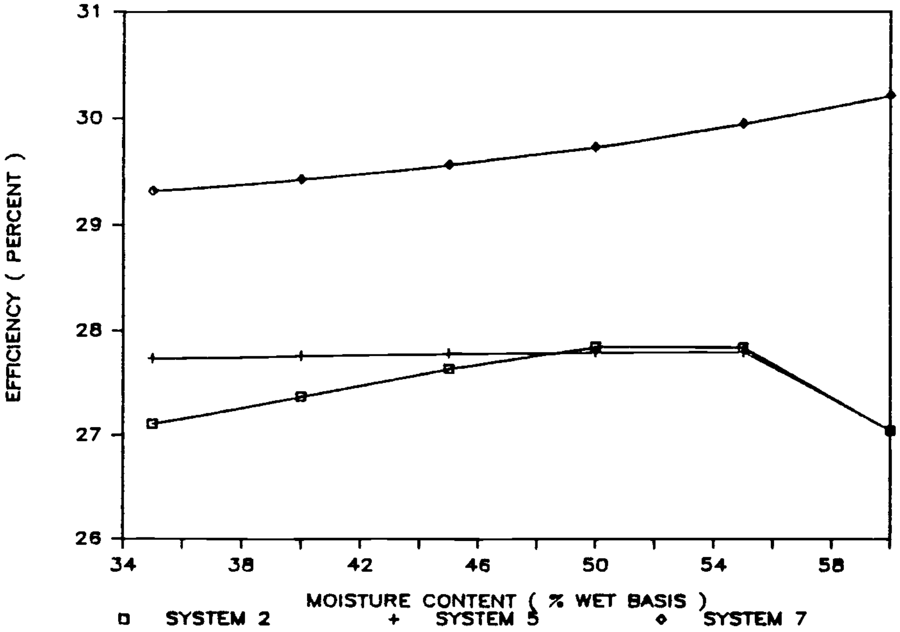

Figure 6.11. Power plant gross efficiency (based on LHV2) versus moisture content for systems 2, 5, and 7.

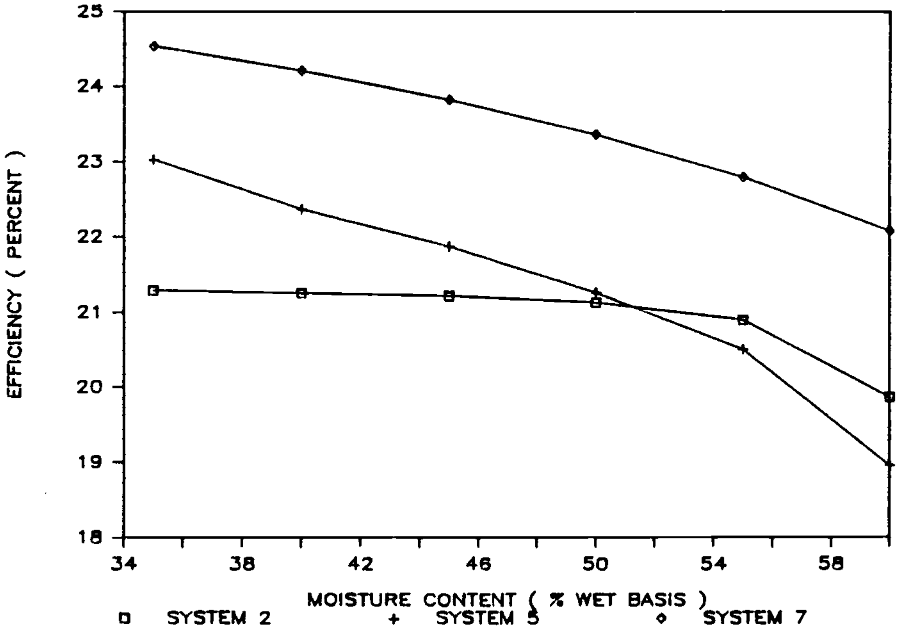

Figure 6.12. Power plant net efficiency (based on HHV) versus moisture content for systems 2, 5, and 7.

than one percent (increasing from 29.3 to 30.2). This increase is due to a higher mass flow rate of gases produced in the combustor as moisture content is increased, as well as, a decrease in LHV2.

System 2 has a constant gross efficiency, since the dryer uses the waste heat exhausting the steam heat recovery system that would normally be exhausted to the atmosphere. This is true up to a moisture content of about 55 percent. At moisture contents higher than 55 percent, additional heat requirements of the dryer (in order to dry the fuel to a specified amount) will reduce the heat available to the steam heat recovery system, and, therefore, reduce the power generated by the steam turbine and plant efficiency.

In system 5 gross efficiency increases as moisture content increases up to about 50 percent. This is again due to the decrease in LHV2. However, at moisture contents of more than 50 percent, the temperature of air entering the gas turbine can no longer be maintained at 1450 F. This decrease will result in a lower power output and, consequently, a lower gross efficiency.

The importance of the EFFl2g is that it takes into account the reduction in net heating value of the fuel (since EFFi2g is based on LHV2).

EFFHn, on the other hand, is calculated based on the net power output of the power plant and the HHV of the fuel. Figure 6.12 shows the plot of net efficiency of the plant versus moisture content of the fuel for systems 2, 5, and 7. In contrast to EFF12g, EFFHn decreases as moisture content of the fuel increases. This is because the auxiliary power requirement of the plant is increased (volume of the gases produced in the combustor increases as moisture content of the fuel decreases) while the HHV remains constant. As with the case of EFF12g, system 7 has a higher efficiency than other systems. System 5 has a higher net efficiency than system 2. This is true for moisture contents of up to 51 percent, after which system 2 shows a higher net efficiency.

From figures 6.11 and 6.12 it can be concluded that EFFHn presents a more meaningful tool for comparison of the results, since it shows the changes in power plant operation rather than changes in fuel heating value. EFF12g, on the other hand, emphasizes the reduction of net heating value of the fuel.

Conclusion

In this section some final conclusions, with regard to combustor performance, power plant performance, and choice of combustion unit are drawn. It should be emphasized again that these conclusions are made based on the thermodynamic model presented in Chapter 5 and do not take into account the chemistry and kinetics of combustion. Note that the basis of recommendation for choosing combustors has been either the information cited in the literature or furnished by manufacturers.

1- Combustor Performance. Based on the results presented in this chapter it is concluded that in order to optimize both the first and second law efficiencies the combustor must be operated at a minimum level of excess air and at the highest combustion air temperature. It is also concluded that the lower the moisture content of the fuel the more efficient the combustion process. It should be pointed out that the conditions indicated above should be considered along with the limitations of each type of combustion unit.

2- Power Plant Performance. From the results presented here, it is concluded that system 7 offers the most attractive alternative among the three systems 2, 5, and 7. It is also important to note that since all types of combustion units use fossil fuel for start up, presence of a trimburner that can support the supplemental fuel used by the combustor seems to be a logical choice. In cases where the moisture content of the fuel is excessively high or the supply of the fuel is short, such a system could be utilized to keep the power plant running at its full capacity. Also, the possibility of using fossil fuel in keeping the combustor operation stable and at a desired temperature should be studied.

Another conclusion is that system 2 (dry fuel combustion system) shows no drastic improvements over system 5 (wet fuel combustion system) that would justify recommending its use.

3- Type of Combustion System. Since one of the objectives of the project has been to utilize the available (off-the-shelf) equipment, the choice of a combustion unit for this power plant is rather limited. This choice includes the Lamb-Cargate wet-cell, fluidized bed combustor, Wellons Cyclo Blast furnace, and the Roemmc burner.

For system 2 the only choice is the Roemmc burner, since none of the other combustion systems are designed for dry fuel. For other systems (5 and 7) because of the importance of control, response to load, automatic ash removal system, and continuous operation of the combustor, only the Lamb-Cargate wet-cell and the fluidized bed combustor will qualify for use in this power plant.

Note that the wet-cell places a restriction on inlet air temperature of the combustor; therefore, the present design of the power plant should be modified. The fluidized bed combustor also places a restriction on exit temperature of the combustor, and if higher temperatures are required, this system cannot be utilized.

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APPENDICES

APPENDIX A

Listing of the Combustor Computer Program

```
$title: 'COMBUSTOR'
$storage:2
$debug
      subroutine BURNER (HHV,radlos,xash,xdirt,xC,xH2,xN2,xO2,
     >                   xH2O,xunbC,xCCO,AF,yCO2a,yH2Oa,yO2a,
     >                   yN2a,yARa,yCOa,Pa,dpburn,Ta,MRa,Tamb,
     >                   yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc,LHV1,
     >                   LHV2,MRwdin,MRbdwd,MRc,Tc,Pc,Qwd,IRRbrn,
     >                   print)
c
c
ccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccc
c                                                                     c
c  This program calculates the temp., mass flow rate, composition,and c
c  pressure of products of the combustion. It also calculates first   c
c  and second law efficiencies, lower heating values of the fuel,     c
c  mass rate of solid discharge, theoretical and actual air-fuel      c
c  ratios, theoretical mass flow rate of air, and heat generated      c
c  in the combustor.                                                  c
c  Input of this program includes higher heating value of the fuel,   c
c  radiation heat loss, mass rate of air to the combustor, mass rate  c
c  of bone-dry fuel, temperature and pressure of the air, pressure    c
c  drop of the combustor, ambient air temperature , dead state temp., c
c  pressure and mole fractions, fuel composition, fraction of         c
c  unburned carbon, and fraction of carbon burned to CO.              c
c                                                                     c
c  METHOD OF CALCULATION:                                             c
c                                                                     c
c  From fuel composition mass flow rates of carbon, hydrogen, oxygen, c
c  nitrogen, ash, dirt, water, carbon burned to CO, and unburned      c
c  carbon are calculated.  It is assumed that all the nitrogen        c
c  remains unchanged, all the hydrogen will form water, and carbon    c
c  will form CO, CO2 or remains unburned as specified.  From these    c
c  mass flow rates, mole rates of the constituents are determined.    c
c  Also mole rates of the constituents in the combustion air is       c
c  calculated.  From mass balance of these streams, final composition c
c  and mass flow rate of flue gas is determined.                      c
c  To determine the flue gas temperature, an energy balance of the    c
c  energy input, energy output, and heat losses is done.  Second law  c
c  is used to calculate the availabilities, irreversibility,  and     c
c  second law efficiencies.                                           c
c                                                                     c
c..NOMENCLATURE:                                                      c
c                                                                     c
c...Substance or Stream Abbreviations:                                c
c                                                                     c
c    a or air......air                                                c
c    AR............argon                                              c
c    ash...........ash in the fuel                                    c
c    b.............biomass (wood)                                     c
```

```
c    c.............combustion products                                 c
c    C or cbn......carbon                                              c
c    CO............carbon monoxide                                     c
c    CO2...........carbone dioxide                                     c
c    dirt..........dirt in the fuel                                    c
c    ds............dead state                                          c
c    e.............exhaust                                             c
c    H2............hydrogen                                            c
c    H2O...........water                                               c
c    N2............nitrogen                                            c
c    O2............oxygen                                              c
c    wd............wood                                                c
c                                                                      c
c                                                                      c
c...Abbreviations:                                                     c
c                                                                      c
c                                                                      c
c    A...........availability (Btu/sec)                                c
c    abs.........absolute (pressure)                                   c
c    AF..........air-fuel ratio (bone-dry wood)                        c
c    bd..........bone-dry                                              c
c    brn.........burner (combustor)                                    c
c    dp..........pressure drop (in. W.C. or  psia)                     c
c    H...........enthalpy (Btu/lb)                                     c
c    in..........into the combustor                                    c
c    IRR.........irreversibility (Btu/sec)                             c
c    ML..........mole rate (lb-mole/sec)                               c
c    MW..........molecular weight (lb/lb-mole)                         c
c    MR..........mass rate (lb/sec)                                    c
c    P...........pressure  (in. W.C. or psia)                          c
c    Q...........heat loss or absorbed (Btu/sec)                       c
c    ref or amb..reference or ambient condition                        c
c    S...........entropy (Btu/lb-deg.R)                                c
c    sld.........solid (discharge from combustor)                      c
c    T...........temperature (deg. F)                                  c
c    unb.........unburned                                              c
c    x...........mass fraction from ultimate analysis of fuel          c
c                per pound of dry clean fuel                           c
c    wdin........wood into the combustor                               c
c    wcf.........wet clean fuel                                        c
c    y***?.......mole fraction of substance *** in ? stream            c
c                                                                      c
c...OTHERS:                                                            c
c                                                                      c
c    EFF1......first law efficiency (percent)                          c
c    EFF21.....second law efficiency based on HHV (percent)            c
c    EFF22.....second law efficiency based on emperical value of       c
c              availability of wood (percent)                          c
c    HHV.......higher heating value,Btu per lb of bone-dry wood        c
c    LHV1......lower heating value (HHV minus heat of vaporization     c
c              of water "formed" in the combustion),dry basis          c
c    LHV2......lower heatong value (LHV1 minu heat of vaporization     c
c              of the moisture of the wet wood),dry basis              c
c    MC........moisture content of wood wet basis (percent)            c
c    radlos....radiation losses (Btu/sec)                              c
```

```
c    EA........excess air (percent)                                    c
c    xCCO......fraction of carbon burned to CO                         c
c    xH2O......moisture content, lb H2O/lb wet, clean fuel (fraction) c
c    xunbC.....fraction of carbon not burned                           c
c                                                                      c
c...EXAMPLES:                                                          c
c                                                                      c
c    Ac.......avilability of flue gas                                  c
c    Hrefa....reference enthalpy of air                                c
c    MLH2b....moles of H2 produced from wood(biomass), per second      c
c    MLO2a....moles of O2 in combustion air, per second                c
c    yCO2c....mole fraction of CO2 in flue gas                         c
c                                                                      c
c                                                                      c
c                                              Abbas Dadkhah-Nikoo     c
c                                                                      c
c                                              1/27/85                 c
c                                              Rev. 7/25/85            c
cccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccccc
c
c
c
c
      implicit real (a-z)
      logical print*2
      common /DS/Tds,Pds,ydsCO2,ydsH2O,ydsO2,ydsN2,ydsAR,ydsCO
      data MWCO2,MWH2O,MWO2,MWN2,MWAR,MWCO,MWC,MWH2/44.01,18.016,
     >     32.0,28.016,39.944,28.01,12.011,2.016/
c
c
c
      eps    = 0.000001
      CpH2O  = 1.0
      Paabs  = 14.696 + 0.03613*Pa
      Pc     = Pa - dpburn
      Pcabs  = 14.696 + 0.03613*Pc
c
c.............calculate air-fuel ratio, mass rate of
c             constituents in fuel, and total mass of
c             biomass fuel to the combustor.(stream b)
c
      AFrat  = MRa/MRbdwd
      MRwcf  = MRbdwd/(1 - xH2O)
      MRH2O  = xH2O*MRwcf
      MRdirt = xdirt*MRbdwd
      MRwdin = MRwcf + MRdirt
      MRash  = xash*MRbdwd
      MRCbn  = xC*MRbdwd
      MRCCO  = xCCO*MRCbn
      MRunbC = xunbC*MRCbn
      MRCCO2 = MRCbn - MRunbC - MRCCO
      MRO2   = xO2*MRbdwd
      MRH2   = xH2*MRbdwd
      MRN2   = xN2*MRbdwd
c
```

```
c..............convert to mole basis ( stream "b" )
c
      MLH2b  = MRH2/MWH2
      MLH2Ob = MRH2O/MWH2O + MLH2b
      MLCb   = MRCbn/MWC
      MLCOb  = MRCCO/MWC
      MLunbC = MRunbC/MWC
      MLO2b  = MRO2/MWO2
      MLN2b  = MRN2/MWN2
      MLCO2b = MRCCO2/MWC
      MLARb  = 0.0
c
c
c..............calculate molecular weight of combustion air
c              and moles of constituents in  combustion air.
c                              ( stream "a" )
c
      MWa    = yARa*MWAR + yCOa*MWCO + yCO2a*MWCO2 +
     >         yH2Oa*MWH2O + yN2a*MWN2 + yO2a*MWO2
      MLARa  = MRa*yARa/MWa
      MLCOa  = MRa*yCOa/MWa
      MLCO2a = MRa*yCO2a/MWa
      MLN2a  = MRa*yN2a/MWa
      MLH2Oa = MRa*yH2Oa/MWa
      MLO2a  = MRa*yO2a/MWa
c
c..............calculate moles of constituents in
c              products of combustion.(stream "c")
c
      MLARc  = MLARa + MLARb
      MLCOc  = MLCOa + MLCOb
      MLCO2c = MLCO2a + MLCO2b
      MLN2c  = MLN2a + MLN2b
      MLH2Oc = MLH2Oa + MLH2Ob
      MLO2c  = MLO2a + MLO2b - (MLCOb/2. + MLCO2b + MLH2b/2.)
c
c..............calculate molefractions of the product
c              of combustion.
c
      sumMLc = MLH2Oc + MLCOc + MLCO2c + MLN2c + MLO2c + MLARc
      yH2Oc  = MLH2Oc/sumMLc
      yCOc   = MLCOc/sumMLc
      yCO2c  = MLCO2c/sumMLc
      yN2c   = MLN2c/sumMLc
      yO2c   = MLO2c/sumMLc
      yARc   = MLARc/sumMLc
c
c..............calculate molecular weight and mass rate
c              of products of combustion. ( MWcp )
c
      MWcp   = yARc*MWAR + yCOc*MWCO + yCO2c*MWCO2 +
     >         yH2Oc*MWH2O + yN2c*MWN2 + yO2c*MWO2
      MRc    = sumMLc*MWcp
c
c..............ENERGY BALANCE
```

```
c
c
c initial guess for Tc
c
      Tc = Ta + 100.0
c
c..............energy into the combustor.
c
c a) Energy of wood
c
      Qwd    = HHV*MRbdwd
c
c b) Energy of air in
c
      Ha     = HGAST(Ta,yCO2a,yH2Oa,yO2a,yN2a,yARa,yCOa)
      Hrefa  = HGAST(Tamb,yCO2a,yH2Oa,yO2a,yN2a,yARa,yCOa)
      Qa     = MRa*( Ha - Hrefa )
c
c..............total energy into the combustor.
c
      Qin    = Qwd + Qa
c
c..............energy losses
c
c a) due to radiation loss
c
      Qrad   = radlos
      Frad   = Qrad/Qin
c
c b) due to unburned carbon
c
      QunbC  = MRunbC*14086.
c
c c) due to co generation
c
      QCCO   = MLCOb*MWCO*4343.6
c
c d) due to formation of H2O from H2 in wood ( at 60 F )
c
      QH2    =MLH2b*MWH2O*1059.
c
c e) due to vaporization of h2o in wood
c
c 1) To vaporize the free water ( at 60 F )
c
       Qfw   = MRH2O*1059.
c
c 2) additional energy for bound water
c
       if ( xH2O .ge. .2308 ) then
          MC = 23.08
       else
          MC = xH2O*100.
       endif
       MRbw  = MC*MRwcf/100.
```

```
        Hbw    = (1./MC)*(4.679415E2*MC - 3.2314115E1*(MC**2) +
     >         1.040786667*(MC**3) + 4.680145E-2*(MC**4) -
     >         6.588278E-3*(MC**5) + 2.569851667E-4*(MC**6) -
     >         3.48937E-6*(MC**7))
        Qbw    = MRbw*Hbw
c
c   total energy to vaporize the water
c
        Qvap   = QH2 + Qfw + Qbw
c
c f) due to heating of the dirt
c
 111    Qdirt  = MRdirt*0.2*(Tc - Tamb)
c
c..............sum of the heat losses
c
c
c   Note that although QH2, Qfw,  and Qw are treated as loss
c   here ( in order to fined Tc ),  however,  these valuse are
c   not lost but absorbed by flue gas.  In other words these values
c   can be recovered if condensation takes place.
c
c
       Qloss = Qrad + QunbC + QCCO + Qvap + Qdirt
c
c..............energy out
c
       Hc     = HGAST(Tc,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
       Hrefc  = HGAST(Tamb,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
       Qc     = MRc*(Hc - Hrefc)
c
c..............check the root
c
       check = ABS((Qin - Qloss - Qc)/Qc)
       if ( check .ge. eps ) then
          Hc = Hrefc + (Qin - Qloss)/ MRc
          Tc = TGASH(Hc,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
          go to 111
       endif
c
c    Total energy transfer to flue gas
c
       Qflue   = Qc + Qvap
c
c..............calculate theo. amont of air
c              and excess air
c
       MLO2th  = MLCb + MLH2b/2. - MLO2b
       MLath   = MLO2th/yO2a
       MRath   = MLath*MWa
       EA      = (MRa - MRath)/MRath*100.0
       thAF    = MRath/MRbdwd
c
c..............calculate lower heating values
c
```

```
          LHV1 = HHV - QH2/MRbdwd
          LHV2 = LHV1 - (Qbw/MRbw + 1059.)*xH2O/(1. - xH2O)
       if ( print ) then
c
c..............secnd law analysis
c
c a)Avail. of wood in
c
c 1) dry wood ( 1: based on HHV, 2: based on impirical formula)
c
        Abdwd1  = MRbdwd*HHV
        Abdwd2  = MRbdwd*((340.124*xC + 5.25*xN2 - 5996.25*xH2 +
     >             1062.45*xO2 - 51.139*xash )*1.7997732 + HHV )
c
c 2) availability of bound water,  based on EMC of 11% (wb)
c
        FDG     = 54.3714 - ( 2.924894E2 - 4.284346E1*MC +
     >             5.039131*(MC**2) - 5.740694E-1*(MC**3) +
     >             3.584556E-2*(MC**4) - 8.335498E-4*(MC**5) -
     >             9.477914E-6*(MC**6) + 1.095668E-6*(MC**7) -
     >             4.493423E-8*(MC**8) + 8.533094E-10*(MC**9))
        Abw     = MRbw*ABS(FDG)
c
c Avail. of wet fuel
c
        Awdin1  = Abdwd1 + Abw
        Awdin2  = Abdwd2 + Abw
c
c b) Avail of air in
c
        Tdsabs = Tds + 459.67
        Sa   = SGASTP(Ta,Paabs,yCO2a,yH2Oa,yO2a,yN2a,yARa,yCOa)
        Sds  = SGASTP(Tds,Pds,ydsCO2,ydsH2O,ydsO2,ydsN2,
     >              ydsAR,ydsCO)
        Hds  = HGAST(Tds,ydsCO2,ydsH2O,ydsO2,ydsN2,ydsAR,ydsCO)


        Aa   = MRa*((Ha - Hds) - Tdsabs*(Sa - Sds))
c
c c)Avail. of flue gas exiting
c
        Hc   = HGAST(Tc,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
        Sc   = SGASTP(Tc,Pcabs,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
        Ac   = MRc*((Hc - Hds) - Tdsabs*(Sc - Sds))
c
c..............calculate the properties at an assumed
c              exhaust temperature of 350
c
        He   = HGAST(350.,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
        Se   = SGASTP(350.,Pcabs,yCO2c,yH2Oc,yO2c,yN2c,yARc,yCOc)
        Ae   = MRc*((He - Hds) - Tdsabs*(Se - Sds))
        Qe   = MRc*(He - Hrefc) + Qvap
c
c..............calculate irrevrsibility of the prosses
c              and  eff.
```

```
      IRR1 = Awdin1 + Aa - Ac
      IRR2 = Awdin2 + Aa - Ac
      EFF21= 100.*Ac/(Awdin1 + Aa)
      EFF22= 100.*Ac/(Awdin2 + Aa)
      EFF23= 100.*(Ac - Ae)/(Awdin2 + Aa)
      EFF1 = 100.*(Qflue - Qa)/Qwd
      EFF12= 100.*(Qflue - Qa - Qe)/Qwd
c
c..............convert to SI, and calc. total mass of solid
c              and total mass in
c
      IRRbrn = IRR2*1.0552
      MRsld = MRash + MRdirt + MRunbc
c
c..............write the results
c
      write(8,10)'-------------------- COMBUSTOR --------------------'
      write(8,10)'                                                   '
      write(8,11)'Theor. Air-Fuel ratio(dry wood)                  :',thAF
      write(8,11)'Theor. mass rate of air, lb/sec                  :',MRath
      write(8,11)'Actual Air-Fuel ratio(dry wood)                  :',AFrat
      write(8,11)'Excess Air, percent                              :',EA
      write(8,10)'                                                   '
      write(8,11)'HHV, Btu/lb (dry basis)                          :',HHV
      write(8,11)'LHV1, Btu/lb (dry basis)                         :',LHV1
      write(8,11)'LHV2, Btu/lb (dry basis)                         :',LHV2
      write(8,10)'                                                   '
      write(8,11)'Mass rate of fuel, lb/sec (bone-dry)             :',MRbdwd
      write(8,11)'Mass rate of fuel, lb/sec (wet dirty)            :',MRwdin
      write(8,11)'Mass rate of water in the fuel, lb/sec           :',MRH2O
      write(8,11)'Mass rate of combustion air, lb/sec              :',MRa
      write(8,11)'Mass rate of solid discharge, lb/sec             :',MRsld
      write(8,11)'Mass rate of flue gas out, lb/sec                :',MRc
      write(8,10)'                                                   '
      write(8,11)'Energy input of fuel, Btu/sec                    :',Qwd
      write(8,11)'Energy input of Comb. air, Btu/sec               :',Qa
      write(8,11)'Energy trans. to flue gas, Btu/sec               :',Qflue
      write(8,10)'                                                   '
      write(8,11)'Heat loss due to unburned C, Btu/sec             :',QunbC
      write(8,11)'Heat loss due to CO gen., Btu/sec                :',QCCO
      write(8,11)'Heat loss due to dirt, Btu/sec                   :',Qdirt
      write(8,11)'Rad. heat loss, Btu/sec                          :',Qrad
      write(8,11)'Energy used for vap. of water, Btu/sec           :',Qvap
      write(8,10)'                                                   '
      write(8,11)'Frac. of unburned carbon                         :',xunbC
      write(8,11)'Frac. radiation loss                             :',Frad
      write(8,11)'Frac. of carbon burned to CO                     :',xCCO
      write(8,10)'                                                   '
      write(8,11)'Temp. of combustion air, deg. F                  :',Ta
      write(8,11)'Temp. of flue gas exiting, deg. F                :',Tc
      write(8,10)'                                                   '
      write(8,11)'Avail. of dry wood (HHV), Btu/sec                :',Abdwd1
      write(8,11)'Avail. of dry wood (emp.), Btu/sec               :',Abdwd2
      write(8,11)'Avail. of bound water, Btu/sec                   :',Abw
      write(8,11)'Avail. of wet wood (HHV), Btu/sec                :',Awdin1
```

```
          write(8,11)'Avail. of wet wood (emp.), Btu/sec       :',Awdin2
          write(8,11)'Avail. of comustion air, Btu/sec         :',Aa
          write(8,11)'Avail. of flue gas, Btu/sec              :',Ac
          write(8,10)'                                          '
          write(8,11)'Irreversibility of comb. (HHV), Btu/sec :',IRR1
          write(8,11)'Irreversibility of comb. (emp.), Btu/sec:',IRR2
          write(8,10)'                                          '
          write(8,11)'First law efficiency, percent            :',EFF1
          write(8,11)'1st law eff. compared to 350 exh.        :',EFF12
          write(8,11)'Sec. law eff. (based on HHV), percent    :',EFF21
          write(8,11)'Sec. law eff. (empirical eq.), percent   :',EFF22
          write(8,11)'2nd law eff. emp. eq. comp. to 350 exh. :',EFF23
c
c
c
 10       format(11x,a)
 11       format(11x,a,f10.4)
        endif
        return
        end
```

APPENDIX B

Computer Program Listing for the Gas Properties Calculation

```
$debug
c       -------- Enthalpy of gas, HGAST(T,yCO2,yH2O,...,yCO) -----
c
c       This function calculates the enthalpy of a gas as a function
c       of temperature and mole fractions.  The equations for Cp are
c       from "Fundamentals of Classical Thermodynamics", G.J. Van Wylen
c       and R.E. Sonntag, pp. 683 - 684.  Cp for argon is assumed to be
c       constant: 5.005 btu/lbmole R.  Maximum error for air was
c       around 0.5 %.
c
        function HGAST (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
        common /DS/Tds,Pds,YdsCO2,YdsH2O,YdsO2,YdsN2,YdsAr,YdsCO
        real Mm,mO2,mN2,mCO2,mH2O,mAr,mCO
        data R /1.9858/
        data a1,a2,a3,a4 /9.3355,-122.56,256.38,-196.08/
        data b1,b2,b3,b4 /8.9465,4.8044E-03,-42.679,56.615/
        data c1,c2,c3,c4 /-.89286,7.2967,-.98074,5.7835E-03/
        data d1,d2,d3,d4 /34.190,-43.868,19.778,-0.88407/
        data e1,e2,e3,e4 /16.526,-0.16841,-47.985,42.246/
        data mO2,mCO2,mN2,mH2O,mAr /32.,44.01,28.016,18.016,39.944/
        data mCO /28.01/
c
        Q =   (T+459.67)/180.
        Qref= (Tds+459.67)/180.
c
        hN2=(a1*(Q-Qref)-a2*2.*(1./sqrt(Q)-1./sqrt(Qref))-a3*(1./Q-
     >          1./Qref)-a4*.5*(1./Q**2-1./Qref**2))*180.
        hO2=(b1*(Q-Qref)+b2*.4*(Q**2.5-Qref**2.5)-b3*2.*(1./sqrt(Q)-
     >          1./sqrt(Qref))-b4*(1./Q-1./Qref))*180.
        hCO2=(c1*(Q-Qref)+c2/1.5*(Q**1.5-Qref**1.5)+c3*.5*(Q**2-Qref**2)
     >          +c4/3.*(Q**3-Qref**3))*180.
        hH2O=(d1*(Q-Qref)+d2/1.25*(Q**1.25-Qref**1.25)+d3/1.5*(Q**1.5-
     >          Qref**1.5)+d4*.5*(Q**2-Qref**2))*180.
        hCO=(e1*(Q-Qref)+e2/1.75*(Q**1.75-Qref**1.75)+e3*2.*(sqrt(Q)-
     >          sqrt(Qref))+e4*4.*(Q**.25-Qref**.25))*180
        hAr=5.005*(Q-Qref)*180.
c
        Mm=yN2*mN2+yO2*mO2+yCO2*mCO2+yH2O*mH2O+yAr*mAr+yCO*mCO
c
        HGAST=(yN2*hN2+yO2*hO2+yCO2*hCO2+yH2O*hH2O+yAr*hAr+yCO*hCO)/Mm
c
        end
c
c       ----- Entropy, SGASTP(T,P,yCO2,...,yCO) -----
c
c       This function calculates the entropy of a gas as a function
c       of temperature, pressure and mole fractions.  The equations for
c       Cp are from "Fundamentals of Classical Thermodynamics",
c       G.J. Van Wylen and R.E. Sonntag, pp. 683 - 684.  Maximum error
c       for air was around 0.5 %.
```

```
c
        function SGASTP (T,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
        common /DS/Tds,Pds,YdsCO2,YdsH2O,YdsO2,YdsN2,YdsAr,YdsCO
        real Mm,mO2,mN2,mCO2,mH2O,mAr,mCO
        data R /1.9858/
        data a1,a2,a3,a4 /9.3355,-122.56,256.38,-196.08/
        data b1,b2,b3,b4 /8.9465,4.8044E-03,-42.679,56.615/
        data c1,c2,c3,c4 /-.89286,7.2967,-.98074,5.7835E-03/
        data d1,d2,d3,d4 /34.190,-43.868,19.778,-0.88407/
        data e1,e2,e3,e4 /16.526,-0.16841,-47.985,42.246/
        data mO2,mCO2,mN2,mH2O,mAr /32.,44.01,28.016,18.016,39.944/
        data mCO /28.01/
c
        Q =   (T+459.67)/180.
        Qref= (Tds+459.67)/180.
        sO2 =0.
        sN2 =0.
        sCO2=0.
        sH2O=0.
        sCO =0.
        sAr =0.
c
        if (yO2 .ne. 0.0) then
                sO2=b1*alog(Q/Qref)+b2/1.5*(Q**1.5-Qref**1.5)-b3/1.5*
     >          (Q**(-1.5)-Qref**(-1.5))-b4*.5*(Q**(-2)-Qref**(-2))
     >          -R*alog(yO2*P/ydsO2/Pds)
        endif
        if (yN2 .ne. 0.0) then
                sN2=a1*alog(Q/Qref)-a2/1.5*(Q**(-1.5)-Qref**(-1.5))-
     >          a3*.5*(Q**(-2)-Qref**(-2))-a4/3.*(Q**(-3)-Qref**(-3))
     >          -R*alog(yN2*P/ydsN2/Pds)
        endif
        if (yCO2 .ne. 0.0) then
                sCO2=c1*alog(Q/Qref)+c2*2.*(sqrt(Q)-sqrt(Qref))+
     >          c3*(Q-Qref)+c4*.5*(Q**2-Qref**2)
     >          -R*alog(yCO2*P/ydsCO2/Pds)
        endif
        if (yH2O .ne. 0.0) then
                sH2O=d1*alog(Q/Qref)+d2*4.*(Q**.25-Qref**.25)+
     >          d3*2*(sqrt(Q)-sqrt(Qref))+d4*(Q-Qref)
     >          -R*alog(yH2O*P/ydsH2O/Pds)
        endif
        if (yCO  .ne. 0.0) then
                sCO=e1*alog(Q/Qref)+e2/.75*(Q**.75-Qref**.75)-e3/.5*(1.
     >          /sqrt(Q)-1./sqrt(Qref))-e4/.75*(Q**(-.75)-Qref**(-.75))
     >          -R*alog(yCO*P/ydsCO/Pds)
        endif
        if (yAr  .ne. 0.0) then
                sAr=5.005*alog(Q/Qref)-R*alog(yAr*P/ydsAr/Pds)
        endif
c
c
        Mm=yN2*mN2+yO2*mO2+yCO2*mCO2+yH2O*mH2O+yAr*mAr+yCO*mCO
c
```

```
        SGASTP=(yN2*sN2+yO2*sO2+yCO2*sCO2+yH2O*sH2O+yAr*sAr+yCO*sCO)/Mm
c
        end
c
c       ------- Temperature of gas, TGASH (H,yCO2,...) ------
c
c               This program calculates the temperature of gas as a
c       function of the enthalpy, Tref and mole fractions.  It uses
c       the function HGAST and iterates.  3-5 iterations are needed
c       for air.  For low temperatures, it only takes 2-3.
c
        function TGASH(H,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
        common /DS/Tds,Pds,YdsCO2,YdsH2O,YdsO2,YdsN2,YdsAr,YdsCO
        T1=Tds
        H1=0.
        T2=Tds+50.
        H2=HGAST(T2,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
c...    Iteration:
c
        DO 10 I=1,20
                T=T2-(H2-H)*(T2-T1)/(H2-H1)
                IF(ABS(T/T2-1.).LT.0.0005)GO TO 20
                T1=T2
                H1=H2
                T2=T
                H2=HGAST(T2,yCO2,yH2O,yO2,yN2,yAr,yCO)
10      CONTINUE
20      TGASH=T
c
        END
c
c       ---- Temperature of gas, f(S,Pds,yCO2,...,yCO) ----
c
c               This program calculates the temperature of gas as a
c       function of the entropy, P, Tds, Pds and mole fractions.  It
c       uses the function SGASTP and iterates.  4-6 iterations are
c       needed for air.  For low temperatures, it only takes 2-4.
c
        FUNCTION TGASS(S,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
        common /DS/Tds,Pds,YdsCO2,YdsH2O,YdsO2,YdsN2,YdsAr,YdsCO
        T1=Tds
        S1=SGASTP(T1,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
        T2=Tds+50.
        S2=SGASTP(T2,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
c...    Iteration:
c
        DO 10 I=1,20
                T=T2-(S2-S)*(T2-T1)/(S2-S1)
                IF(ABS(T/T2-1.).LT.0.0005)GO TO 20
                T1=T2
                S1=S2
```

```
                 T2=T
                 S2=SGASTP(T2,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
10          CONTINUE
20          TGASS=T
c
            END
c
c           ----------------------- Mole fractions -----------------------
c
c                This function calculates the mole fractions after
c           mixing two gas streams. The mole fractions of the mixing gases
c           and mass flows are input to the program.  It then returns the
c           mass flow, mole fractions and molar mass of the new mixture.
c           ynO2,ynN2,... are number of moles pr. lb of "dry" gas.
c
            subroutine MOLFRC (MRi1,yi1CO2,yi1H2O,yi1O2,yi1N2,yi1Ar,yi1CO,
     >           Mi1,MRi2,yi2CO2,yi2H2O,yi2O2,yi2N2,yi2Ar,yi2CO,Mi2,
     >           MRo,yoCO2,yoH2O,yoO2,yoN2,yoAr,yoCO,Mo)
            real MRi1,MRi2,MRo,mO2,mN2,mCO2,mH2O,mAr,mCO,MOLi1,MOLi2,MOLo
            real Mi1,Mi2,Mo
            data mO2,mCO2,mN2,mH2O,mAr /32.,44.01,28.016,18.016,39.944/
            data mCO /28.01/
c
c...        molar mass of inlet streams:
c
            Mi1=yi1O2*mO2+yi1N2*mN2+yi1CO2*mCO2+yi1Ar*mAr+yi1H2O*mH2O+
     >           yi1CO*mCO
            Mi2=yi2O2*mO2+yi2N2*mN2+yi2CO2*mCO2+yi2Ar*mAr+yi2H2O*mH2O+
     >           yi2CO*mCO
c
c...        number of moles (pr.sec):
c
            MOLi1=MRi1/Mi1
            MOLi2=MRi2/Mi2
            MOLo=MOLi1+MOLi2
c
            yoCO2=(yi1CO2*MOLi1+yi2CO2*MOLi2)/MOLo
            yoH2O=(yi1H2O*MOLi1+yi2H2O*MOLi2)/MOLo
            yoO2=(yi1O2*MOLi1+yi2O2*MOLi2)/MOLo
            yoN2=(yi1N2*MOLi1+yi2N2*MOLi2)/MOLo
            yoAr=(yi1Ar*MOLi1+yi2Ar*MOLi2)/MOLo
            yoCO=(yi1CO*MOLi1+yi2CO*MOLi2)/MOLo
c
            MRo=MRi1+MRi2
            Mo=yoO2*mO2+yoN2*mN2+yoCO2*mCO2+yoAr*mAr+yoH2O*mH2O+yoCO*mCO
c
            end
c
c           ------------ Viscocity, MUGAST (T,yCO2,...,yAr) -------------
c
c                This function calculates the dynamic viscosity of
c           gas.  It uses semiempirical formula recommended by
c           Frank M. White, "Viscous Fluid Flow", pp. 25-38.
c
c                y1  = yCO2      y2  = yH2O
```

```
c               y3  = yO2         y4  = yN2
c               y5  = yAr         y6  = yCO
c
      real function MUGAST (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
      real mu(6),y(6),M(6),T0(6),S(6),mu0(6)
      data M /44.01,18.016,32.,28.016,39.944,28.01/
      data T0 /491.6,750.,491.6,491.6,491.6,491.6/
      data S /400.,1550.,250.,192.,260.,245./
      data mu0 /.1370,.1703,.1919,.1663,.2125,.1657/
c
      y(1)= yCO2
      y(2)= yH2O
      y(3)= yO2
      y(4)= yN2
      y(5)= yAr
      y(6)= yCO
      Tabs=T+459.67
c
      do 10 i=1,6
              mu(i)=mu0(i)*(Tabs/T0(i))**1.5*(T0(i)+S(i))/(Tabs+S(i))
10    continue
c
      temp=0.
      do 20 i=1,6
              temp1=0.
              do 30 j=1,6
                      temp2=1.+sqrt(mu(i)/mu(j))*(M(j)/M(i))**.25
                      temp3=sqrt(8.+8.*M(i)/M(j))
                      temp1=temp1+temp2**2/temp3*y(j)
30            continue
              temp=temp+mu(i)*y(i)/temp1
20    continue
c
      MUGAST=temp*1.E-3/14.882
c
      end
c
c     ------------ Conductivity, KGAST (T,yCO2,...,yAr) -------------
c
c             This function calculates the thermal conductivity of
c     gas.  It uses semiempirical formula recommended by
c     Frank M. White, "Viscous Fluid Flow", pp.30 - 36.  It is not
c     accurate for high temperatures. (approx 4-5 % for T > 1000. F)
c
c             y1  = yCO2        y2  = yH2O
c             y3  = yO2         y4  = yN2
c             y5  = yAr         y6  = yCO
c
      real function KGAST  (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
      real k(6),y(6),M(6),T0(6),S(6),k0(6)
      data M /44.01,18.016,32.,28.016,39.944,28.01/
      data T0 /491.6,491.6,491.6,491.6,491.6,491.6/
      data S /4000.,2300.,400.,300.,270.,320./
```

```
        data k0 /.008407,.01036,.01419,.0140,.009444,.01342/
c
        y(1)= yCO2
        y(2)= yH2O
        y(3)= yO2
        y(4)= yN2
        y(5)= yAr
        y(6)= yCO
        Tabs=T+459.67
c
        do 10 i=1,6
                k(i)=k0(i)*(Tabs/T0(i))**1.5*(T0(i)+S(i))/(Tabs+S(i))
10      continue
c
        temp=0.
        do 20 i=1,6
                templ=0.
                do 30 j=1,6
                        temp2=1.+sqrt(k(i)/k(j))*(M(j)/M(i))**.25
                        temp3=sqrt(8.+8.*M(i)/M(j))
                        templ=templ+temp2**2/temp3*y(j)
30              continue
                temp=temp+k(i)*y(i)/templ
20      continue
c
        KGAST=temp
c
        end
c
c       ----------- Density, f(T,P,yCO2,yH2O,yO2,yN2,yAr,yCO) ----------
c
c       This function calculates the density according to the ideal
c       gas law. Temperature in F, pressure in psia and density in
c       lbm/cu.ft
c
        function ROGAS (T,P,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
        real Mm,mO2,mN2,mCO2,mH2O,mAr,mCO
        data mO2,mCO2,mN2,mH2O,mAr /32.,44.01,28.016,18.016,39.944/
        data mCO /28.01/
c
        Mm=yN2*mN2+yO2*mO2+yCO2*mCO2+yH2O*mH2O+yAr*mAr+yCO*mCO
        ROGAS =Mm/10.73/(T+459.67)*P
c
        end
c
c       -------- Prandtl number, f(T,yCO2,yH2O,yO2,yN2,yAr,yCO) -------
c
c       This function calculate the Prandtl number of a gas as a
c       function of temperature.  But since the thermal conductivity
c       is not accurate for high temperatures, the Prandtl number is
c       assumed constant.  That is just as accurate.
c                                       Sigurdur Brynjolfsson, 07-24-83
c
        function PrGAST (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
```

```
c
c       real k,mu,KGAST,MUGAST
c
c       CP=HGAST(T+.5,T-.5,yCO2,yH2O,yO2,yN2,yAr,yCO)
c       mu=MUGAST (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
c       k =KGAST (T,yCO2,yH2O,yO2,yN2,yAr,yCO)
c
c       PrGAST=Cp*mu/k * 3600.
        PrGAST=0.69
c
        end
```

APPENDIX C

Typical Input of the Combustor Computer Program

Typical Output of the Combustor Computer Program

Typical Input of the Computer Program

```
Tds   =59.0
Pds   =14.696
ydsCO2=0.0003
ydsH2O=0.0111
ydsO2 =0.2072
ydsN2 =0.7722
ydsAR =0.0092
ydsCO =0.000001
xash  =0.008
xdirt =0.035
xC    =0.523
xH2   =0.063
xN2   =0.001
xO2   =0.405
xCCO  =0.01
dpburn=8.0
yCO2in=0.000296
yH2Oin=0.011129
yO2in =0.20717
yN2in =0.77221
yARin =0.009196
yCOin =0.000001
Pa    =26.0
Tamb  =60.
radlos=390.
xunbC =0.02
xCCO  =0.01
HHV   =8800.
MRa   =37.57
Ta    =600.0
xH2O  =0.5
MRbdwd=2.21
```

Typical Output of the Combustor Program

```
-------------------- COMBUSTOR --------------------

Theor. Air-Fuel ratio(dry wood)               :     6.4760
Theor. mass rate of air, lb/sec               :    14.3119
Actual Air-Fuel ratio(dry wood)               :    17.0000
Excess Air, percent                           :   162.5097

HHV, Btu/lb (dry basis)                       :  8800.0000
LHV1, Btu/lb (dry basis)                      :  8203.7830
LHV2, Btu/lb (dry basis)                      :  7006.7130

Mass rate of fuel, lb/sec (bone-dry)          :     2.2100
Mass rate of fuel, lb/sec (wet dirty)         :     4.4974
Mass rate of water in the fuel, lb/sec        :     2.2100
Mass rate of combustion air, lb/sec           :    37.5700
Mass rate of solid discharge, lb/sec          :      .1181
Mass rate of flue gas out, lb/sec             :    41.9491

Energy input of fuel, Btu/sec                 :19448.0000
Energy input of Comb. air, Btu/sec            : 4981.7530
Energy trans. to flue gas, Btu/sec            :23571.3300

Heat loss due to unburned C, Btu/sec          :  325.6204
Heat loss due to CO gen., Btu/sec             :  117.0787
Heat loss due to dirt, Btu/sec                :   25.7060
Rad. heat loss, Btu/sec                       :  390.0000
Energy used for vap. of water, Btu/sec        : 3798.8800

Frac. of unburned carbon                      :      .0200
Frac. radiation loss                          :      .0160
Frac. of carbon burned to CO                  :      .0100

Temp. of combustion air, deg. F               :  600.0000
Temp. of flue gas exiting, deg. F             : 1721.6670

Avail. of dry wood (HHV), Btu/sec             :19448.0000
Avail. of dry wood (emp.), Btu/sec            :20362.8600
Avail. of bound water, Btu/sec                :    55.4834
Avail. of wet wood (HHV), Btu/sec             :19503.4800
Avail. of wet wood (emp.), Btu/sec            :20418.3500
Avail. of comustion air, Btu/sec              : 1663.9420
Avail. of flue gas, Btu/sec                   :12015.3500

Irreversibility of comb. (HHV), Btu/sec       : 9152.0760
Irreversibility of comb. (emp.), Btu/sec      :10066.9400

First law efficiency, percent                 :    95.5861
1st law eff. compared to 350 exh.             :    59.8442
Sec. law eff. ( based on HHV), percent        :    56.7634
Sec. law eff. (empirical eq.), percent        :    54.4117
2nd law eff. emp. eq. comp. to 350 exh.       :    47.4584
```

APPENDIX D

Calculation of Availability for the Bound Water

APPENDIX D

Calculation of Availability for the Bound Water

Availability of the bound water is calculated from the change in Gibbs free energy of the bound water. The change in Gibbs free energy of bound water at an equilibrium moisture content of 11 percent (wet basis) is equal to:

$$FDG = 54.3714 - \{2.924894*10^{2} - 4.284346*10^{1}*MC + 5.039131*(MC^{2}) - 5.740694*10^{-1}*(MC^{3}) + 3.584556*10^{-6}*(MC^{4}) - 8.335498*10^{-4}*(MC^{5}) - 9.477914*10^{-6}*(MC^{6}) + 1.095668*10^{-6}*(MC^{7}) - 4.493423*10^{-8}*(MC^{8}) + 8.533094*10^{-10}*(Mc^{9})\}$$

where FDG is the change in Gibbs free energy of the bound water (Btu/lb), and MC is the percent moisture content of the wood (wet basis). MC is equal to the fiber saturation point (if the moisture content of the wood is higher than FSP), or it is equal to the moisture content of the wood (if the moisture content of the wood is less than FSP.

The availability of the bound water is then calculated as:

$$Abw = MRbw* |(FDG)|$$