# THE THERMAL PERFORMANCE STUDY OF BIOMASS FIELD- ERECTED WATER TUBE BOILERS USING ANALYTICAL MODEL

**SIVABALAN TANAPALA**

**FACULTY OF ENGINEERING**
**UNIVERSITY OF MALAYA**
**KUALA LUMPUR**

**2013**

## UNIVERSITI MALAYA

## ORIGINAL LITERARY WORK DECLARATION

Name of Candidate: SIVABALAN A/L TANAPALA

Registration/Matric No: KGH090021

Name of Degree: MASTER OF ENGINEERING (MECHANICAL)

Title of Project Paper/Research Report/Dissertation/Thesis ("this Work"):
**THE THERMAL PERFORMANCE STUDY OF BIOMASS FIELD- ERECTED WATER TUBE BOILERS USING ANALYTICAL MODEL**

Field of Study: HEAT TRANSFER

I do solemnly and sincerely declare that:

(1) I am the sole author/writer of this Work;
(2) This Work is original;
(3) Any use of any work in which copyright exists was done by way of fair dealing and for permitted purposes and any excerpt or extract from, or reference to or reproduction of any copyright work has been disclosed expressly and sufficiently and the title of the Work and its authorship have been acknowledged in this Work;
(4) I do not have any actual knowledge nor ought I reasonably to know that the making of this work constitutes an infringement of any copyright work;
(5) I hereby assign all and every rights in the copyright to this Work to the University of Malaya ("UM"), who henceforth shall be owner of the copyright in this Work and that any reproduction or use in any form or by any means whatsoever is prohibited without the written consent of UM having been first had and obtained;
(6) I am fully aware that if in the course of making this Work I have infringed any copyright whether intentionally or otherwise, I may be subject to legal action or any other action as may be determined by UM.

Candidate's Signature Date

Subscribed and solemnly declared before,

Witness's Signature Date
Name:
Designation:

# THE THERMAL PERFORMANCE STUDY OF BIOMASS FIELD-ERECTED WATER TUBE BOILERS USING ANALYTICAL MODEL

**SIVABALAN TANAPALA**

**RESEARCH REPORT SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF MASTER ENGINEERING**

**FACULTY OF ENGINEERING**
**UNIVERSITY OF MALAYA**
**KUALA LUMPUR**

**2013**

# ABSTRACT

Renewable Energy is one of the major contributors in fulfilling the world's energy demand. Biomass is one of the renewable energy sources which are currently being exploited by the community. Palm oil industries use the waste from processed palm fruits, shell and empty fruit bunch as fuel to generate steam in order to cook the fresh fruits and generate power for the whole plant. Tropical climate in Malaysia provides the best platform for the palm trees to grow and maintain Malaysia's ranking as second largest palm oil producer in the world. Stoker firing water tube boilers are used widely in the mills as it is the best method that converts the chemical energy in the fuel through combustion into mechanical energy which runs the turbine to generate electricity. A comprehensive review have been done through this paper on the existing design, fuel, heat transfer, heat losses and CFD studies of biomass boiler. The heat transfer and heat losses in the boiler due to biomass combustion have been analysed and studied thoroughly in the literatures. The present design of boiler used in the tropical countries is based on the empirical data from western countries due to lack of tropics data's. Higher temperature, humidity and wind velocity of tropical climate impact on the boiler and its component efficiency were studied. Other than that the effect on fuel demand and the heat transfer in the components were also studied. An actual running unit in Casanare, Colombia which is in tropical zone were selected and simulated for the study. The cost impact and the payback period were determined for the best and worst climate condition that happens in the tropics.

# ABSTRAK

Tenaga boleh ganti merupakan salah satu alternatif yang boleh digunakan untuk memenuhi keperluan tenega di sisi masyarakat dunia. Industri sawit merupakan salah satu contoh industri yang menggunakan hampas kelapa sawit yang merupakan salah satu sumber tenaga bagi tujuan memasak buah sawit dan menjana kuasa untuk keseluruhan kilang. Cuaca Khatulistiwa yang sememangnya sesuai untuk penanaman sawit menjadikan Malaysia sebagai pengeluar sawit kedua terbesar di dunia selepas Indonesia. Dandang merupakan salah satu komponen mekanikal yang digunakan secara meluas di industry sawit untuk mengubah tenaga kimia yang terkandung di dalam hampas kelapa sawit kepada tenaga mekanikal untuk menjana kuasa. Rumusan mendalam telah dibuat bagi tujuan mengenalpasti rekabentuk sedia ada dandang, bahan api, kadar kehilangan haba serta analisa sedia ada dinamik bendalir dandang. Kadar kehilangan haba dandang merupakan aspek penting yang ditekankan di dalam rumusan. Rekabentuk dandang yang sedia ada adalah berdasarkan data kajian yang diperolehi daripada negara bermusim dan tiada data yang diperolehi daripada kawasan tropika. Impak kawasan tropika yang mempunyai suhu yang panas dan lembap sepanjang tahun terhadap tahap efisien dikaji. Selain daripada itu kesannya terhadap pengunaan bahan bakar turut dikaji. Sebuah dandang berkapasiti 35 tan/jam bertempat di Casanare,Colombia telah diambil sebagai model untuk kajian ini. Impak cuaca terhadap tempoh bayaran balik bagi dandang dibuat dengan membandingkan penjimatan kos bagi keadaan terburuk dan terbaik.

# ACKNOWLEDGEMENT

I would like to thank god for giving an opportunity for me to further studies in Master's level and maintain my confidence throughout the project until the completion of this project. I would also like to express my gratitude and appreciation to my supervisor Associate Professor.Ir.Dr.Yau Yat Huang for his consistent encouragement, advice and invaluable guidance throughout this project.

A special thanks to my bosses Mr. Jose Lorenzo and Mr. Okuma from Okutech Sdn. Bhd. for their guidance throughout this project and for allowing me to use their boiler data's. The guidance and knowledge that they had given will be always remembered and used in future. I extend my sincere thanks Mr. Jairo Prada from Manuelita Palm oil mill for providing the plant data and necessary information required.

I'm also deeply indebted to my family members in Sungai Petani, Kedah and Tampin, Negeri Sembilan for their love, sacrifice, motivation and support given throughout these studies. My sincere thanks goes to my parents Mr.Tanapala and Mrs.Letchimi who keep on motivating me by asking when you are going to finish your Master's degree.

I would like to express my appreciation to my beloved wife Santha for her patience, love, and support which have been crucial throughout the completion of this project. Finally, thank you to those who have contributed directly or indirectly towards the success of this research project.

# CONTENTS

## LIST OF FIGURES:

## LIST OF TABLES:

## NOMENCLATURES

| Nomenclature | Description | Unit |
|---|---|---|
| A | Area | $m^2$ |
| $C_P$ | Specific heat | kj/kg.K |
| GCV | Gross Calorific Value | kj/kg |
| H | Enthalphy | kj/kg |
| h | Heat transfer coefficient | $W/(m^2K)$ |
| HHV | High Heating Value | kj/kg |
| Kg | Kilogram | kg |
| LCV | Low Calorific Value | kj/kg |
| LHV | Low Heating Value | kj/kg |
| M | Meter | m |
| M | Mass flow rate | kg/s |
| Q | Heat transfer rate | kW |
| S | Seconds | s |
| T | Ton | t |
| U | Overall Heat Transfer Coefficient | $W/(m^2K)$ |
| V | Volume | $m^3$ |
| Wt | Weight | kg |
| Ω | Ratio of moisture/ratio of dry air | kg/kg |
| A/F | Air Fuel | - |
| ASME | American Society of Mechanical Engineers | - |
| BS | British Standard | - |
| C | Carbon | - |
| CFD | Computational Fluid Dynamic | - |
| Cl | Chlorine | - |
| CO | Carbon Monoxide | - |
| CO2 | Carbon Dioxide | - |
| EA | Excess Air | - |
| Exp | Experimental | - |
| F | Fibre | - |
| FC | Fixed Carbon | - |
| H | Hydrogen | - |
| Hr | Hour | - |
| L | Heat Loss | - |
| LMTD | Log Mean Temperature Difference | - |
| N | Nitrogen | - |
| Nox | Nitrogen Oxide | - |
| O | Oxygen | - |
| PTC | Power Test Code | - |
| S | Sulphur | - |
| S | Shell | - |

| | | |
|---|---|---|
| $SO_2$ | Sulphur Dioxide | - |
| T | Temperature | - |
| VM | Volatile Matter | - |
| $m^2$ | Square meter | - |
| °C | Degree Celcius | - |
| °F | Degree Fahrenheit | - |
| $\eta$ | Efficiency | - |

## List of Appendices:

# CHAPTER 1: INTRODUCTION

## 1.0 Background of Studies

Steam Generator or better known as "Boiler" is part of daily life where it will be used for different purposes such as Production, Palm oil mills, Oil and Gas, Power Plants, Oleo Chemical Plants and others. Boilers can be categorized into a few types which are as follows Fire tube, Water Tube, Combination Boiler, Hot water Boiler, Thermal Oil Heater and others. These Boilers are specified according to industries, capacity, cost and availability of space. Boilers fire using organic and non-organic materials such as coal, biomass, rubbish, oil and gas to generate hot pressurized steam above the atmospheric pressure. Boiler converts the chemical energy in the fuel via combustion into thermal energy, which will be used to boil water in the steam drum continuously until steam produced. A good boiler design should fulfil the thermodynamics, heat transfer and environmental requirements in order to save cost and prevent pollution.

The demand for electricity has become higher as the industries blooming particularly in tropical country like Malaysia. The growing number of oil palm related industries and power plants also affected the demand for electricity but the location of these industries has limited the access of electricity supplies. Boilers or Steam generators have given an alternative solution in order to tackle this kind of situation but fuel has become a restriction since there is a limitation on the availability and the high rising cost. Biomass boilers preferred nowadays especially in the power generation industries since the availability, cost and environmental effect are better compared to the coal, oil and gas fired boilers. The main concern of Biomass boiler is to give the same efficiency as the fossil fuel fired boilers because of the heating value is lower. The efficiency of boiler will enable it to convert the chemical energy from combustion into heat energy to generate steam. The efficiency of biomass boilers can be maintained or improved by minimizing the factors that affect the performance such as heat losses in the equipment.

The impact of tropical climate towards the boiler and its component efficiency has yet been studied. Study on tropical climate impact which is known for high ambient temperature and humidity towards boiler efficiency will become a novel approach which can be used to improve boiler designs in the tropical region.

# CHAPTER 2: RESEARCH OBJECTIVES

The objective of this research is will be mainly focusing on optimizing the design of a field erected water tube boiler based on the tropical climate. In order to achieve this there is few aspects need to be clarified such as:

i) Ambient temperature and relative humidity effect on boiler efficiency.

Tropical climate has high temperature and humidity for the whole year and the impact of this factor will be studied. The heat loss due to climate effect will be determined by using the Power Test Code (PTC 4.1).

ii) Ambient temperature and relative humidity effect on fuel consumption

Reduction in efficiency causes the fuel consumption to increase and this will directly affect the cost. Tropical climate impact on the fuel consumption will be studied and analyzed for different ambient temperature and humidity.

iii) Ambient temperature and relative humidity effect on Boiler Heat Transfer

Radiation and convection are the main heat transfer mechanism in the boiler while conduction plays a minor role. High temperature and humidity in tropical increases the moisture content in the air where higher sensible heat is found. The impact of the moisture content in the air and flue gas towards the efficiency of boiler components will be studied and discussed. Furthermore the effect of those parameters towards the heat transfer surfaces will be discussed.

iv) Cost analysis and payback period

The payback period for the boiler will include the Boiler cost, installation, commissioning and labour cost for a period of 15 years. Surplus fibre and shell from fuel saving normally sold to other boiler companies, industries that is producing mattresses and agriculture farms or used to produce biogas. The payback period will be calculated based on the selling value of fuel and the number of additional days for production results from the fuel saving. The impact of humidity and ambient temperature will be studied for these different conditions.

# CHAPTER 3: LITERATURE REVIEW

## 3.0 General Design of a Biomass Boiler

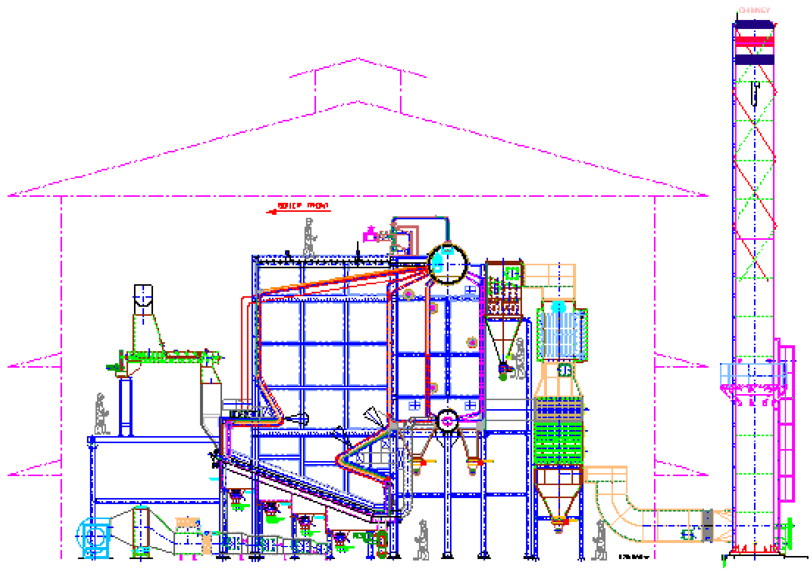


**Figure 3.1: Okutech Bi-Drum boiler**

## 3.1 Systems in Boiler

Boiler is a complete system that comprises of air, fuel, water and control system which enable it to operate at its best efficiency.

### 3.1.1 Air & Draft System

Air system in the boiler comprises of forced draft air, induced air and combustion air supply or better known as the secondary air supply. Forced draft air is normally preheated in order to eliminate the moisture content in the air and to dry out the fuel in the furnace. The forced draft air is supplied through under the grate and normally creates positive draft in the furnace [1]. Induced draft fan brings out the combustion products or flue gas through the stack and creates negative pressure in the furnace in order to prevent back fire [1]. The flue gas will pass through the super heater, convection bank, economizer and air preheater before it is taken out through the stack. The secondary air provides additional combustion air required in order to make sure almost stoichiometric combustion achieved. The secondary air normally supplied on top of the flame which will cause turbulence effect to take place and result into better combustion[1]. Introduction of secondary air is an important breakthrough in boiler combustion engineering [2-4].

### 3.1.2 Combustion System

Chemical energy in a biomass converted into heat energy by using few methods such as direct firing, gasification, co-firing and others. The easiest method is by using direct firing where the biomass material will be burned in the combustion chamber or furnace [5-7]. The heat from biomass combustion will be exploited to produce steam in a boiler. Since direct firing is an inefficient way of converting energy, a more advanced approach known as biomass gasification can be used. This method employs a partial combustion process where it converts the fuel into a combustible gas. These gases can be used to replace natural gas even though it has lower energy content. Biomass gasification promises high efficiency and offers the best option for future of biomass-based power generation as it is still under development [8-10]. Co-firing of coal and biomass can also be considered as another way of increasing the efficiency of biomass fuel.

Grate firing is a favourite choice used to convert chemical energy in biomass into heat through direct firing [11]. A spreader stoker system will throw the fuel uniformly on the grate .Fines will ignite and burn in suspension while the bigger particle will drop in the thin fast burning bed [3, 12]. This will cause the fuel to be evenly distributed across the active grate area. Grate firing widely used to burn coal or solid wastes because the advantage of this method is simple construction, easy handling and flexible but the disadvantage of this firing method is low thermal efficiency compared to other methods [13].

### 3.1.3 Feed Water System.

The feed water system comprises of makeup water for the boiler, chemical treatment system, deaerator and economizer. Make up water for the boiler need to undergo some treatment before it can be supplied to the boiler which is important to prevent erosion and cavitation in the drum and tubes[1]. Deaerator removes oxygen which is an important agent for corrosion from the water supply. The water temperature will increase during this process before being supplied to the boiler.

## 3.2 Components in a Biomass Grate Fired Boiler.

### 3.2.1 Furnace.

Furnace is the main component in a biomass boiler because this is where the fuel is burned and combustion takes place[14]. The wall of furnace consists of water and steam cooling carbon steel or low alloy steel in order to maintain the temperature within an acceptable limit. The tubes were connected at the top and bottom by headers or manifolds. Current furnace design has implemented a membrane wall type where it helps to reserve the thermal energy needed compared to the spaced tube furnace which have been used as the main furnace construction for many years [15]. The membrane wall furnace provides a cooler furnace which will protect the cast iron used in the grate construction from being damaged and prevent leakage by giving a tight gas enclosure. Furnace contributes to the highest exergy destruction rate in a boiler where 19,270.8 kJ/s of energy have been destroyed while 10,320 kJ/s of exergy destructed. Furthermore, energy loss in the heat exchanging equipment was 22.5% but exergy loss is about 52% where combustion gases carries away 9.2% of heat [16].

### 3.2.2 Grate

A biomass boiler requires grate for a uniform combustion where the fuel will be thrown evenly on top of it. Other than that the air has to be supplied uniformly through the grates to release the energy under optimum condition. A grate design that is highly resistant to air flow is desirable to achieve even air distribution across the surface and even combustion conditions. Combustion grates existing today are from the continuous ash discharge type and classified as Pin Hole or fixed grate, Vibrating grate, Travelling Grate and Reciprocating grate. The type of grate will vary based on the type of fuel used to provide a better combustion and efficiency other than the cost. Vibrating Grate provides a better combustion platform compared to the fixed grate and Travelling grate because it provides an intermittent vibration which helps to distribute the fuel evenly for complete combustion[17].Reciprocating Grate is divided into four zones which are Moist fuel inlet, Fuel drying and ignition, Combustion and finally de-ashing. The ignition of the moist fuel starts from the flame and furnace wall radiation which is transported against the airflow [12]. The grate fired biomass boilers have its advantage to control emissions due to incomplete combustion or NOx by increasing the fuel residence time in the combustion zone [3]. *Table 3.1* shows the difference between grates used for biomass combustion.

| Type of grate | The major features |
|---|---|
| Stationary sloping grate | The grate does not move. The fuel burns as it slides down the slope under gravity. The degree of sloping is an important characteristic of this kind of boiler.<br>Disadvantages :<br>1) Difficult control of the combustion process.<br>2) Risk of Avalanching of the fuel. |
| Travelling Grate | The fuel is fed on one side of the grate and is burned when the grate transports it to the ash pit. Compared to stationary sloping grate, it has improved control and better carbon burnout efficiency (due to small layer of fuel on the grate). |
| Reciprocating grate | The grate tumbles and transports fuel by reciprocating (forward or reverse) movements of the grate rods as combustion proceeds. Finally the solids are transported to the ash pit at the end of the grate. Carbon burnout is further improved due to better mixing. |
| Vibrating grate | The grate has a kind of shaking movement that spreads the fuel evenly. This type of grate has less moving parts than other moveable grates (and thus lower maintenance and higher reliability). Carbon burnout efficiency is also further improved. |

**Table 3.1: Different type of grates and their characteristics.[3]**

### 3.2.3 Drums

Steam drum is one of the main components in a boiler where the water is boiled before supplied to the process. The minimum water inlet temperature is at ambient temperature and boiled until it reaches saturated temperature. The feed water will be heated to an elevated temperature in order to reduce the temperature gap between the saturated temperature and the incoming water temperature from deaerator or economizer minimizing the amount of energy consumed by the boiler [18]. The selection of material and thickness of the steam drum is crucial because it has to withstand high pressure and temperature. Code such as ASME and BS are crucial and widely used in calculating the drum thickness and material selection. Mud drum is used as a container for mud and sludge in the feed water which is supplied to the boiler. The mud is collected in the drum during natural circulation that happens when the convection bank is heated by the combustion gas. The water in the tubes boils and turned into saturated steam when the tubes are heated by the combustion gas causing the pressure to drop and the steam to rise back to the steam drum [1, 18].

### 3.2.4 Super heater

Super heater is a bank of tubes located at the exit of flue gas from the furnace which is known as the radiation area. The saturated steam will pass these banks and the temperature will increase due to convective the heat transfer process[1]. The dry superheated steam will be sent to the turbine for power generation and pressure reduced before sent to the sterilizer.

### 3.2.5 Convection Bank

Convection bank is where the water is circulated by using natural circulation from the steam drum and mud drum. The flue gas that exit the furnace will pass the convection bank to heat up the water contained tubes and further reduced the temperature of the flue gas. Convection bank can be categorized into two types which is one pass and three pass Convection Bank.

### 3.2.6 Economizer

Economizer is used in the boiler to heat up the incoming feed water to a certain temperature. The heated water will be further boiled in the steam drum until it reaches saturated temperature. The use of economizer is preferred because it helps to save the fuel consumed.

### 3.2.7 Air Preheater

Air preheater is used to heat up the incoming combustion air in order to remove the moisture in the air. Air preheater consists of tubes where the flue gas flows and opening for the combustion air. The air preheater used the theory of cross flow heat exchanging equipment where the air as the cold fluid outside the tube is heated by the hot combustion gas in the tubes.

## 3.3 Boiler Water Circulation

Water-tube boilers can be further differentiated to the method of water circulation which is natural circulation boilers and forced circulation boilers. In natural circulation or thermal circulation the water will be heated and expands causing the density of the water decreases as it changes phase into steam. The gravity will force water in the drum to flow downwards and the steam water mixture to flow upwards[1]. Natural circulation can be classified into two type which is free or acceleration type. There are four main factors that affect the circulation rate of natural circulation which are the height of the boiler, Operating pressure, Heat Input and free flow areas of the component. Forced circulation is created by adding a pump to circulate the water and steam mixture rather than depending on the density difference.

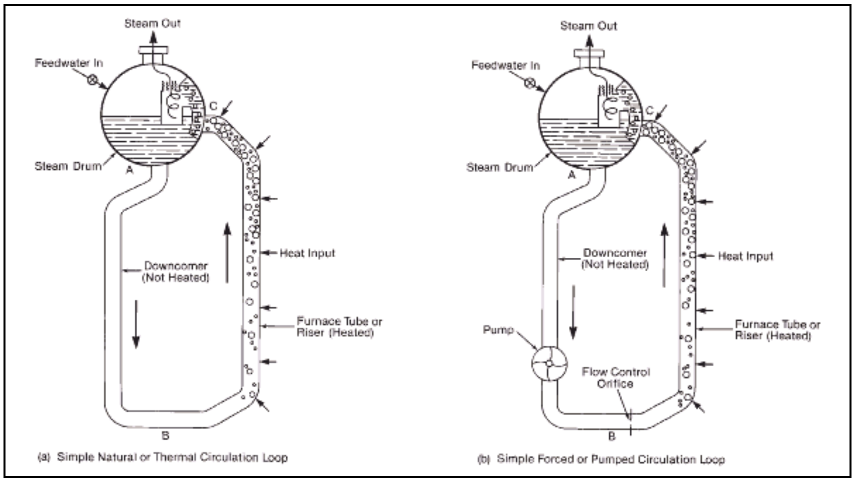



**Figure 3.2: Water Circulation in Water Tube Boiler; (a) Simple Natural Circulation Loop, (b) Simple Forced or Pumped Circulation Loop.[18]**

## 3.4 Biomass as Boiler's Alternative Fuel

Biomass is one of the oldest renewable resources after the sun, hydro and wind power which is obtained from live or dead organisms. Biomass is based on carbon and mixtures of organic molecules such as hydrogen, oxygen, nitrogen and also other atoms. During growth, biomass recycled carbon dioxide by absorbing it from the environment and emits it again during combustion which indirectly helps to avoid the greenhouse effect [19, 20]. Biomass fuels can be converted into various forms such as liquid, solid and gas with the help of conversion processes that involves physical, chemical and biological factors[20]. There are five groups of biomass material that is used for energy generation which is virgin wood from forestry or wood processing industries, energy crops, agricultural, food and industrial wastes[19, 21, 22].

### 3.4.1 Ultimate Analysis of Biomass Fuels.

Ultimate analysis helps to identify the content of Carbon, Hydrogen, Nitrogen and sulphur in biomass fuels in terms of percentage. Furthermore the fuel properties will be used to determine the Calorific value by calculating the percentage C, H, O and the environmental impact of biomass.

| Fuel | C | H | O | N | S | Cl | Ref |
|---|---|---|---|---|---|---|---|
| | % | % | % | % | % | % | |
| Lignite | 65.20 | 4.50 | 17.50 | 1.30 | 4.10 | 0.4 | [20] |
| Spruce Wood | 51.40 | 6.10 | 41.20 | 0.30 | 0.0 | 0.10 | [23] |
| Hazelnut shell | 50.80 | 5.60 | 41.10 | 1.0 | 0.0 | 0.20 | [23-25] |
| Corn cob | 49.00 | 5.40 | 44.20 | 0.40 | 0.0 | 0.20 | [23] |
| Corn stover | 49.40 | 5.60 | 42.50 | 0.60 | 0.10 | 0.30 | [23] |
| Tobacco Stalk | 49.30 | 5.60 | 42.80 | 0.70 | 0.0 | 0.20 | [23] |
| Tobacco leaf | 41.20 | 4.90 | 33.90 | 0.90 | 0.0 | 0.30 | [23] |
| Almond shell | 47.90 | 6.00 | 41.70 | 1.10 | 0.06 | 0.10 | [23] |
| Sawdust | 46.90 | 5.20 | 37.80 | 0.10 | 0.04 | - | [23, 26] |
| Rice husk | 47.80 | 5.10 | 38.90 | 0.10 | - | - | [23] |
| Bagasse | 44.80 | 5.40 | 39.60 | 0.40 | 0.01 | - | [23, 27] |
| Palm Kernels | 51.00 | 6.50 | 39.50 | 2.70 | 0.27 | 0.21 | [23] |
| Pistachio Shell | 48.79 | 5.91 | 43.41 | - | - | - | [28] |
| Cereals | 46.50 | 6.10 | 42.00 | 1.20 | 0.10 | 0.20 | [29] |
| Switch grass | 42.04 | 4.97 | 35.44 | 0.77 | 0.18 | - | [30, 31] |
| Rice Straw | 38.45 | 5.28 | - | 0.88 | - | - | [32] |
| Poplar | 48.40 | 5.90 | 39.60 | 0.40 | 0.01 | - | [27] |
| Alfafa stalk | 45.40 | 5.80 | 36.50 | 2.10 | 0.09 | - | [31] |

**Table 3.2: Ultimate analysis of different types of biomass fuels (wt% dry basis).**

### 3.4.2 Proximate Analysis of biomass fuels.

Proximate analysis is one of the methods used to identify the percentage of volatile matter, fixed carbon and ash contents to study the combustion phenomenon of biomass. High ash contents in biomass fuels causes ignition and combustion problems while high amount of carbon and particulates increases the heating value of biomass fuel [33]. Fouling and slagging problem happens due to the low melting point of the ash.

| Fuel | C | H | O | N | S | Cl | Ref |
|---|---|---|---|---|---|---|---|
| | % | % | % | % | % | % | |
| Palm Stem | 47.50 | 5.90 | 42.50 | 0.28 | 0.13 | 0.18 | [34] |
| Palm Branch | 45.60 | 5.60 | 39.30 | 0.19 | 0.16 | 1.33 | [34] |
| Palm Fibre | 52.20 | 7.10 | 28.00 | 0.70 | 0.07 | 0.06 | [34] |
| Palm Shell | 51.50 | 5.70 | 37.70 | 0.36 | 0.03 | 0.05 | [34] |
| Coffee Husks | 49.40 | 6.10 | 41.20 | 0.81 | 0.07 | 0.03 | [34] |
| Masasi CNS | 56.00 | 6.90 | 34.70 | 0.44 | 0.05 | 0.03 | [34] |
| Olam CNS | 56.90 | 7.00 | 33.60 | 0.45 | 0.04 | 0.03 | [34] |
| Rice Husks | 35.60 | 4.50 | 33.40 | 0.19 | 0.02 | 0.08 | [34] |
| Rice Bran | 37.80 | 5.00 | 35.40 | 0.55 | 0.05 | 0.09 | [34] |
| Bagasse | 48.10 | 5.90 | 42.40 | 0.15 | 0.02 | 0.07 | [34] |
| Jatropha Seeds | 56.60 | 7.50 | 27.40 | 3.16 | 0.17 | 0.12 | [34] |
| Mango Stem | 48.00 | 5.80 | 41.50 | 0.13 | $<0.012$ | 0.03 | [34] |

**Table 3.3: Ultimate analysis of different types of Tropical biomass fuels (wt% dry basis)**

| Fuel | FC | VM | ASH | Ref |
|---|---|---|---|---|
| | % | % | % | |
| Spruce Wood | 29.30 | 70.20 | 1.50 | [23] |
| Hazelnut shell | 28.30 | 69.30 | 1.40 | [23-25] |
| Corn cob | 11.50 | 87.40 | 1.10 | [23] |
| Corn stover | 10.90 | 84.00 | 5.10 | [23] |
| Almond shell | 20.71 | 76.00 | 3.29 | [23] |
| Sawdust | 15.00 | 82.20 | 2.80 | [23, 26] |
| Rice husk | 16.95 | 61.81 | 21.24 | [23] |
| Bagasse | 11.95 | 85.61 | 2.44 | [23, 27] |
| Switch grass | 14.34 | 76.69 | 8.97 | [30, 31] |
| Rice Straw | 15.86 | 65.47 | 18.67 | [32] |
| Alfafa stalk | 15.81 | 78.92 | 5.27 | [31] |

**Table 3.4: Proximate analysis of different types of biomass fuels (wt% dry basis)**

### 3.4.3 Heating Value

Heating Value is the energy content available in the biomass fuel which will be converted during combustion for steam production [21, 35]. Higher Heating Value (HHV) is known as heat release from combustion of a unit fuel mass whether the product of combustion will be in forms of ash ,gaseous Carbon dioxide ($CO_2$),Nitrogen (N),Sulphur Dioxide ($SO_2$) and liquid Vapour. Lower Heating Value (LHV) is calculated by using HHV where all the water in the combustion product remains as vapour[21]. Table 3.5 shows calorific value for different fiber and shell mixture at 7 different mills in Malaysia. The fuel and ash properties of wood and agricultural residues are shown in table 3.6.

| Mill no. | Proportional weight (%) | Calorific value (MJ $kg^{-1}$) |
|---|---|---|
| 1 | F 64 | 7368 |
| | S 36 | 6943 = 14.311 |
| 2 | F 60 | 6076 |
| | S 40 | 7714 = 13.790 |
| 3 | F 67 | 7713 |
| | S 33 | 6364 = 14.077 |
| 4 | F 70 | 8058 |
| | S 30 | 5786 = 13.844 |
| 5 | F 60 | 6076 |
| | S 40 | 7714 = 13.790 |
| 6 | F 50 | 5756 |
| | S 50 | 9202 = 14.958 |
| 7 | F 50 | 5756 |
| | S 50 | 9292 = 14.958 |

F—Fibre, S—Shell. Calorific value of Fiber = 19.188 MJ $kg^{-1}$, Shell = 21.430 MJ $kg^{-1}$, Moisture content ($x$) in: Fiber = 40%, Shell = 10%.

**Table 3.5: Calorific Value of Biomass waste [35].**

| Fuel type | Hog fuel | Furniture waste | Hybrid poplar | Forest residue | Switchgrass | Rice straw | Almond hulls | Wheat straw Hi-alkali |
|---|---|---|---|---|---|---|---|---|
| Fuel ash (%) | 1.0 | 3.61 | 2.70 | 3.97 | 8.97 | 18.67 | 6.13 | 9.55 |
| Chlorine (%) | | < 0.01 | 0.04 | 0.04 | 0.19 | 0.58 | 0.02 | 1.79 |
| HHV (MJ $kg^{-1}$) | 20.95 | 20.15 | 18.95 | 20.18 | 18.08 | 18.91 | 18.90 | 16.78 |
| *Ash composition* | | | | | | | | |
| $SiO_2$ | 35.18 | 57.62 | 0.88 | 17.78 | 65.18 | 74.67 | 9.28 | 37.06 |
| $Al_2O_3$ | 2.31 | 12.23 | 0.31 | 3.55 | 4.51 | 1.04 | 2.09 | 2.66 |
| $TiO_2$ | 0.01 | 0.50 | 0.16 | 0.50 | 0.24 | 0.09 | 0.05 | 0.17 |
| $Fe_2O_3$ | 4.41 | 5.63 | 0.57 | 1.58 | 2.03 | 0.85 | 0.76 | 0.84 |
| CaO | 25.37 | 13.89 | 44.4 | 45.46 | 5.60 | 3.01 | 8.07 | 4.91 |
| MgO | 7.62 | 3.28 | 4.32 | 7.48 | 3.00 | 1.75 | 3.31 | 2.55 |
| $Na_2O$ | 5.64 | 2.36 | 0.23 | 2.13 | 0.58 | 0.96 | 0.87 | 9.74 |
| $K_2O$ | 9.26 | 3.77 | 20.08 | 8.52 | 11.60 | 12.30 | 52.90 | 21.70 |
| $SO_3$ | 3.03 | 1.00 | 3.95 | 2.78 | 0.44 | 1.24 | 0.34 | 4.44 |
| $P_2O_5$ | 5.68 | 0.50 | 0.15 | 0.44 | 4.50 | 1.41 | 5.10 | 2.04 |
| $CO_2$ | | | 19.52 | | | | 20.12 | |
| Undet. | 1.58 | − 0.78 | 5.43 | 9.78 | 2.32 | 13.89 | − 2.89 | 14.32 |
| Total (%) | 100.00 | 100.00 | 100.00 | 100.00 | 100.00 | 100.00 | 100.00 | 100.00 |
| Alkali ($Na_2O + K_2O$, kg $GJ^{-1}$) | 0.07 | 0.10 | 0.16 | 0.20 | 0.56 | 1.22 | 1.62 | 1.66 |

**Table 3.6 : Fuel and ash properties of wood and agricultural residues[36].**

### 3.4.4 Emission from Biomass Combustion.

Emission from biomass combustion such as Nitrogen Oxide (NOx), Sulphur Oxide (SOx) and particulate matter is higher compared to oil and gas fired boilers. Eight biomass fuel pellets such as apple pomace (Malus domestica), reed canary grass (Phalaris arundinacea), pectin waste from citrus shells (Citrus reticulata), sunflower husk (Helianthus annuus), peat, wood and two types of wheat straw pellets (Triticum aestivum) have been tested under standard laboratory condition while DIN plus wood pellet tested in real life condition [37]. A 40 kW multi-fuel domestic pellet boiler under standard laboratory conditions another two 35 kW boilers in real life conditions were used for this testing purposes. The study shows that in normal condition the NOx emission higher compared to laboratory conditions but CO and particle emissions were lower.

A semi industrial boiler was used to compare the emission and combustion efficiency of various vegetable oils and petro diesel [38]. The effects of oil energy rate and the air-fuel ratio on combustion efficiency and emission were analysed to determine the outcome of replacing petro diesel with the product of vegetable oils. Biodiesel fired boiler performance is found to be similar with petrodiesel at higher energy consumption and lower air-fuel ratio. Increase in the combustion air had caused the biodiesel combustion efficiency to drop. There is no difference found in CO emission at the fuel complete combustion pressure specified [38].

Comparisons have been carried out for different air flow rate effect towards the combustion efficiency and combustion gas emission at different energy level of biodiesel and mixture of biodiesel-diesel [39]. Furthermore from the studies made it is found that Biodiesel is more efficient compared to diesel at lower energy level where the emission rate is lower compared to diesel except for NOx emission.

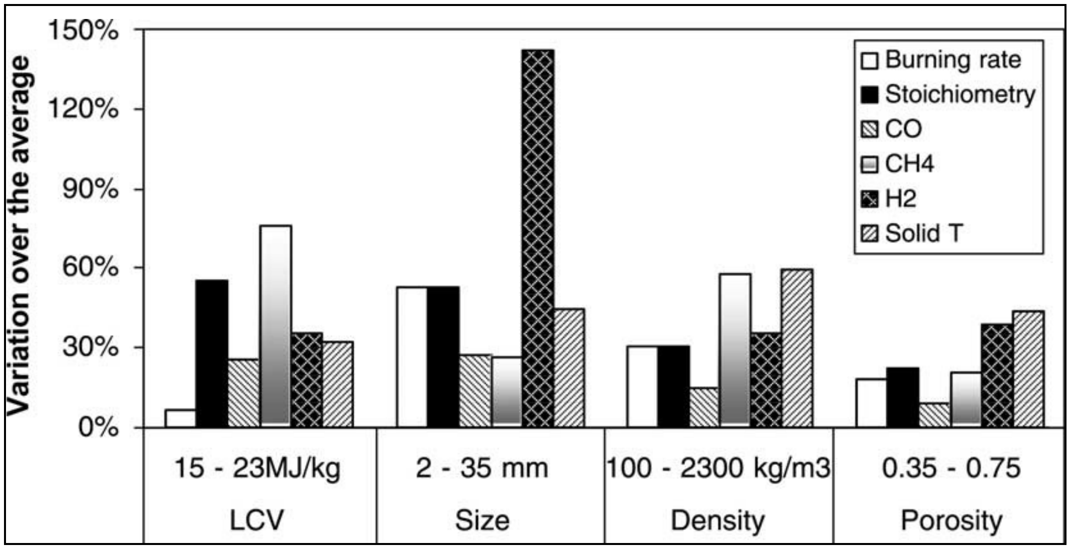



**Figure 3.3:Summary of the effects of different fuel properties[40].**

## 3.5 Tropical Climate Characteristics

Tropical climate is a climate where the mean temperature for the whole year is maintained above 18 °C (64 °F). Tropical climate remains persistent throughout the year and the seasonal variations are mainly dominated by precipitation or relative humidity [40]. Tropical climate can be further divided into few types such as Tropical rainforest climate, Tropical monsoon climate and Tropical wet and dry climate or known as savannah climate. The climate types are only differentiated by the precipitation that happens in the climate zones. Relative humidity in tropical zones ranges from 77% to 88% [40]. Relative Humidity plays a major part in the design of low-temperature systems because it controls the dew point temperature[41].

## 3.6. Heat Transfer in Boilers

### 3.6.1 Mode of Heat transfer in Boiler.

There are three basic modes of heat transfer that took place in a boiler which is conduction, convection and radiation. The efficiency of a boiler is how it transferred maximum amount of heat from combustion that took place in a furnace to the equipment and minimize the heat loss. Conduction normally took place in the wall of the boiler equipment's such as the drums, furnace, super heater, economizer, air preheater and others. Conduction process in boiler normally transfers heat from high temperature to low temperature. Convection process took place as the heat from flue gas is transferred to the equipment's and when the heat is taken out during heat loss. Radiation in a boiler mainly took place in a furnace where fuel is burnt and the heat is absorbed by the furnace wall. The heat transfer in the boiler equipment is reduced due to deposits which were formed during combustion of biomass.

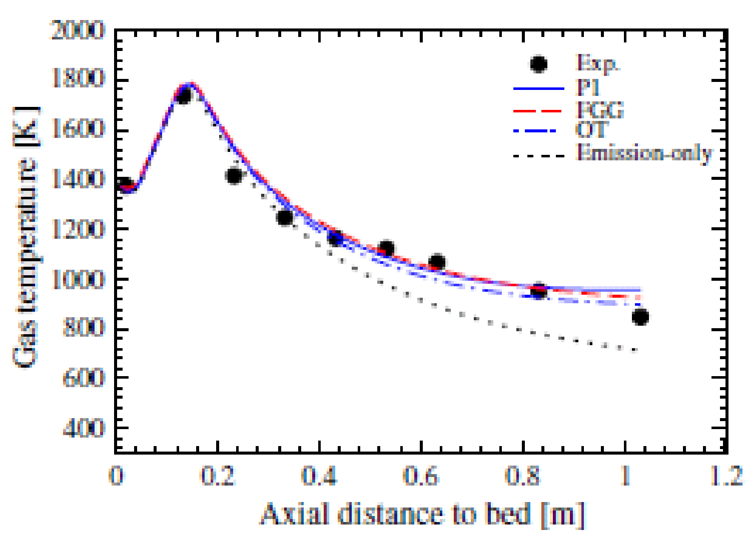



**Figure 3.4a : Mean temperature of the combustion gas along the axis of the furnace, calculated using different radiative transfer models [42].**

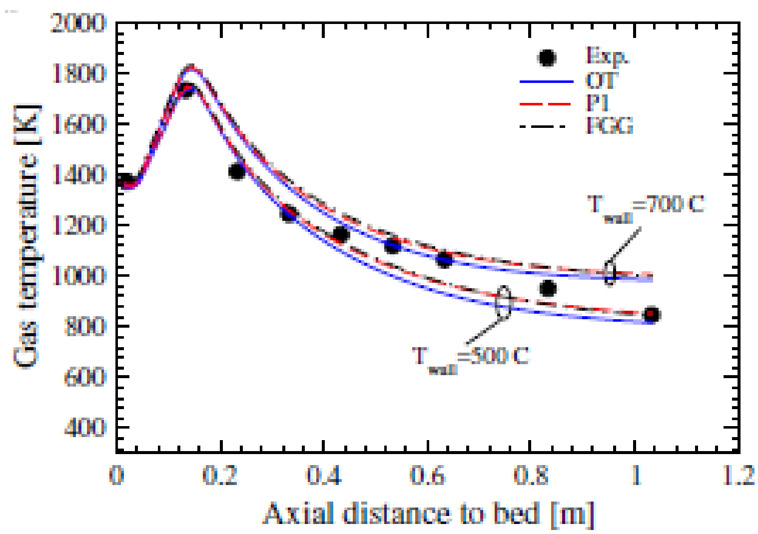



**Figure 3.4b: Comparison of Mean gas temperature along the axis of the furnace calculated using different radiative transfer models for different wall temperature[42].**

## 3.7 Boiler Efficiency

Efficiency of a steam generator can be defined as the percentage of heat input that is utilised effectively in order to maximize heat transfer for steam generation by reducing the heat losses in the boiler[16]. ASME Power Test Code, PTC 4.1 proposed that the boiler efficiency can be calculated by using two methods which are known as the direct method and indirect method[43].

### 3.7.1 Direct Method

The direct method compares the energy gain by the water when it converts into steam during combustion with the energy content of the fuel. Direct method makes the plant operator job easier to evaluate the efficiency of the boiler because it needs only few parameters to help the computation. Direct method has its disadvantages because it didn't give any clue to the operator on why the efficiency is low. The disadvantage of this method is it doesn't help the operators to determine the other losses that should be considered at different efficiency level.

### 3.7.2 Indirect Method

The heat balance efficiency measurement method or indirect method considers the heat losses that occur in the boiler [44-47]. Indirect method efficiency can be obtained by subtracting the loss percentage of various losses that happens in a boiler from 100%. The major losses which occur in a boiler such as follows [1, 44-47]:

1. Dry Flue Gas Loss
2. Moisture in fuel
3. Hydrogen in fuel
4. Moisture in Air
5. Unburnt Gas Loss due to Carbon Monoxide
6. Specific Heat Loss from Bottom Ash and Fly Ash
7. Radiation and Unaccounted Loss
8. Radiation to Furnace Bottom
9. Heat Credit due to Mill, Primary Air Fan, Forced Draught Fan, Circulating Water Pumps

#### 3.7.2.1 Heat loss due to moisture in the fuel

Greater amounts of energy are required to burn fuel with large amount of moisture where it leaves as superheated vapour. The moisture will be brought to boiling point by the sensible heat which occurs due to the heat loss[45]. Moisture in the biomass fuel exists due to the environment factor and contribution from the process. One of the major factors which contribute to the amount of moisture in the palm waste is due to sterilization process where the palm fruit is cooked by using saturated steam.

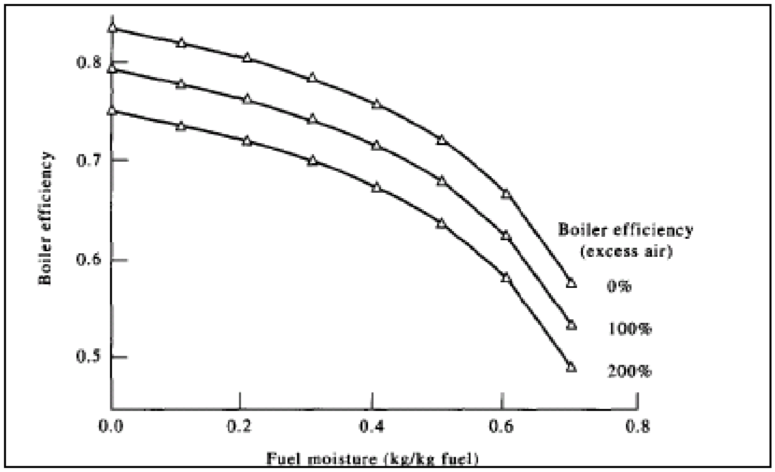



**Figure 3.5: Boiler Efficiency as a function of fuel moisture content [46].**

### 3.7.2.2 Heat loss due to Combustion of Hydrogen

Calorific value plays an important role in combustion of fuel where the gases and moisture are taken up the stack. Heat loss occurs during combustion of hydrogen where water is formed and converted into steam. Heat is carried out due to the latent heat content of the water[45].

### 3.7.2.3 Heat loss due to moisture in the air

Relative humidity of air can greatly affect the performance of a boiler where moisture in incoming air will be superheated as it passes through the boiler[45, 46]. In order to remove moisture in the incoming air, it will be pre-heated by using a heat exchanger or air pre-heater.

#### 3.7.2.4 Heat loss due to dry flue gas

Heat Losses through the chimney affected by few factors such as scale formation in the tube, high flue gas velocity in the boiler, excess air, inlet air temperature and final gas temperature. Combustion gas acid dew point temperature achieved if the flue gas exit temperature is too low where it may lead into acid deposits. The acid dew point is the temperature where sulphuric acid deposits begin to form where moisture absorbs the sulphur from the gas and starts to degrade the metal.

#### 3.7.2.5 Heat loss due to soot-blower

Soot-blower loss happens during cleaning of ashes on the boiler components where a certain percentage of steam is supplied to the soot blower. The steam will be supplied intermittently to the soot-blower to minimize heat loss that happens in a boiler.

#### 3.7.2.6 Heat loss due to blow down

Blow down process normally takes place during high water level where the water is taken out to maintain the water level in the drum. Water within the system is replaced with treated water once the water level dropped and thus dilutes any chemicals in the water.

#### 3.7.2.7 Heat loss due to incomplete combustion

Incomplete combustion happens during low loads, especially during night time operation or when there is insufficient air supply which causes a high percentage of ash in the fuel. Product of combustion from incomplete combustion such as Carbon Monoxide (CO),$H_2$ and various other hydrocarbon reacts with oxygen and releases more energy[45].

#### 3.7.2.8 Heat loss due to combustible in ash

Combustible in ash occurred when the amount of fuel supplied is too much or known as rich combustion which caused unburned fuel to happen. Hot unburned ash that mixed up in the fuel carries away the heat during de-ashing process and the loss is not more than 2% [46].

#### 3.7.2.9 Miscellaneous factors

There are several other losses that occur in the boiler such as radiation, leaks and others. These losses occur due to insufficient cladding or insulation around header drums, piping and other components.

## 3.8 Numerical modelling of combustion in boiler

CFD studies have been carried out to study the circulating fluidized bed boiler operation and it is usually used to simulate the problems in an operating boiler at site. The impact of ash deposits towards the boiler efficiency were studied thoroughly and compared with the actual running unit. The ash deposits reduced the heat transfer rate by creating a layer on the equipment and affect the efficiency of the boiler[48].

CFD studies used to study superheater tube failure in the boiler due to increased temperature, decreased hardness values and scale build-up on the inner surface of the tube. The inner scale creates an insulation layer where it blocks the steam from cooling the tube which will cause the tube to overheat and fail. The scale generation effect towards the tube surface temperature and hardness of the tube material were studied based on service hours. Furthermore, life of the tubes estimated by the cumulative creep damage method which is later modelled by using ANSYS[49]

Different models such as a three-dimensional geometry, k–e gas turbulence model, Eulerian–Lagrangian approach, particles-to-turbulence interaction, diffusion model of particle dispersion, six-flux method for radiation modelling and pulverized coal combustion model based on the global particle kinetics and experimentally obtained kinetic parameters can be studied by using CFD. The models predicted the impact of those parameters towards furnace flue gas temperature and the furnace wall radiation which is later verified with actual running unit [50]. Zone method was used to predict the radiative heat fluxes on the furnace wall. Minimum heat flux obtained in the corner of the wall or near the exit while maximum value found in the central region where the directed heat flux area is vast. Even though the zone method is accurate in determining radiative heat transfer but it can’t be applied to all types of furnace due to the complex geometry of real furnaces [51].

A CFD model or numerical model can be used in order to study the biomass grate combustion in a specific type of boiler. Analytical equations representing the combustion on the travelling grate and freeboard area normally carried out separately due to limitations of the CFD software such as ANSYS. The grey box model used to study the oxygen concentration while the black box model for steam generation in the heat recovery system [9]. Travelling grate modelled by separate zones in order to study the combustion process at each stage. The model is linearized and reduced from 46 states to 17 states to facilitate a real-time implementation [9]. Grate combustion normally consists of two ordinary differential equations. The equation represents the water content in the grate water evaporation zone and in the dry biomass located at thermal decomposition zone [12].

Poor mixing of bulk air flow and secondary air in the furnace is the main factor leading to the incomplete combustion. A CFD model developed to study the air flow in the boiler where different condition can be studied and validated with the actual site data. The gas phase temperature above the grate is higher because it was influenced by the bed model. The heat transfer rate in the superheater is found to be higher than 100% due to the boundary condition set lower than actual [52].

# CHAPTER 4: METHODOLOGY

## 4.0 Overview

This chapter will provide a brief explanation about the methods that will be used throughout this project. A biomass boiler selected and the site data's such as Boiler Size, Steam Flow, Temperature, Pressure, Fuel data, and the heat transfer area will be used to model the boiler. Data such as Fuel consumption, Air flow, and Flue gas flow will be determined theoretically due to lack of measurement devices in the palm oil mill boiler. On the other hand theoretical estimation will be helpful in order to study the changes that happen in the boiler due to the climate changes and its effect towards the boiler performance. The collected information will be used to model the boiler as close as possible as the real running unit before carrying out required studies. It is important to model the boiler as close as possible in order to make sure the result will be almost accurate with the actual condition. Ambient Temperature and Humidity level will be the main factors studied in order to determine the impact of these parameters on boiler and its components efficiency and performance. The calculations involved in this study will be explained subsequently as follows;

a) Combustion Calculation of Fuel

b) Theoretical Input Parameters

c) Boiler Efficiency Calculation as per ASME PTC 4.1

d) Energy Efficiency of Boiler components.

e) Effectiveness of boiler components heat transfer surfaces

f) Fuel Saving Analysis & Payback Period Analysis

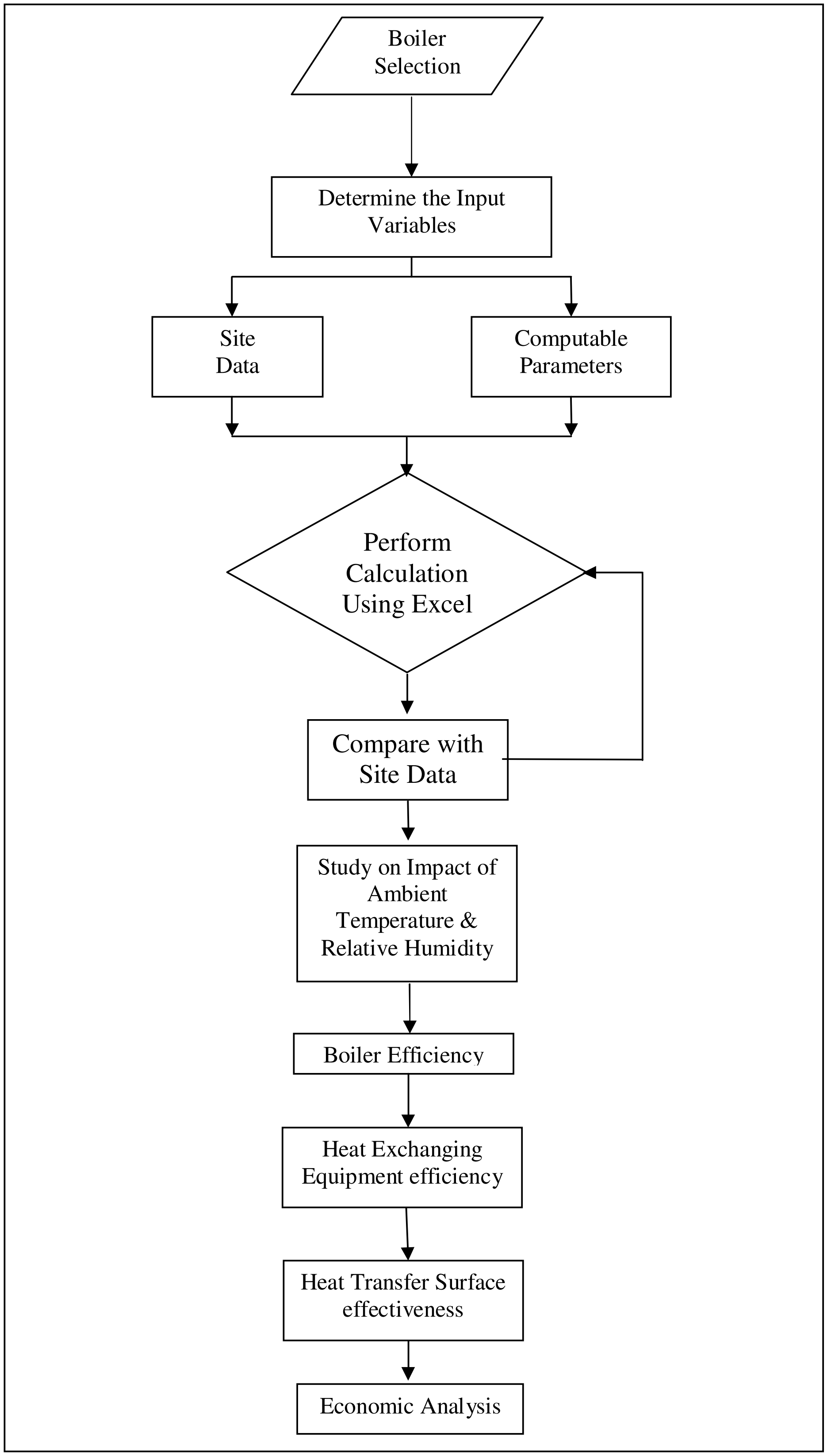



**Figure 4.1: Methodology Charts of Studies**

## 4.1 Combustion Calculation of Fuel

i) Fuel Ultimate Analysis

The ultimate analysis method will be used to find the composition of the average fuel mixture of Palm fibre and shell in weight percentage of carbon, hydrogen, oxygen, sulphur, nitrogen and the calorific value of the received biomass fuel. The analysis is done based on dry basis or moisture free basis.

| **FUEL** | | **Palm Fiber** | **Palm Shell** |
|---|---|---|---|
| **CONTENT** | **UNIT** | **75** | **25** |
| C | Wt.% | 47.20 | 52.40 |
| H2 | Wt.% | 6.00 | 6.30 |
| O2 | Wt.% | 36.70 | 37.30 |
| S | Wt.% | 0.30 | 0.20 |
| N2 | Wt.% | 1.40 | 0.60 |
| ASH | Wt.% | 8.40 | 3.20 |
| H20 | Wt.% | 35.00 | 15.00 |
| GCV | kcal/kg | 4,586 | 5,122 |

**Table 4.1: Fuel Properties of Palm waste used in the Boiler**

Sample Calculation (Dry Basis)

%wt Carbon : Palm Fiber

=(1-Moist Content in Fuel%)* %wt Carbon (1)

= (1 – 35/100)*47.20 %

= 30.28 %

ii) Fuel Mixture Combustion Calculation

Combustion calculation was carried out by calculating the weight of required Oxygen for complete combustion. The required weight of Oxygen will be determined by using the molecular weight of the substances involved.

$C + O_2 = CO_2$

$S + O_2 = SO_2$

$H_2 + 0.5O_2 = H_2O$

Weight of Oxygen Required in Reaction with Carbon.C

$$= \frac{M_{O_2}}{M_C} \text{ x \%wt } C_{(\text{Average})} \qquad (2)$$

$$= \frac{2x16}{12} \text{ x } 34.15\%$$

$= 0.91$ kg/kg $_{\text{d.a}}$

## 4.2 Theoritical Input Value

| Fuel Content | Unit | Average Analysis |
|---|---|---|
| C | Wt.% | 48.50 |
| H2 | Wt.% | 6.08 |
| O2 | Wt.% | 36.85 |
| S | Wt.% | 0.28 |
| N2 | Wt.% | 1.20 |
| ASH | Wt.% | 7.10 |

**Table 4.2 Average analysis of fuel used in the boiler**

i) Stoichiometric air required

Weight of air required (Stoichiometric)

$$= \frac{Oxygen\ \mathrm{Re}quired\ in\ Combustion}{\%\ Oxygen\ in\ air} \tag{3}$$

= 4.33 kg/kg

ii) Total Dry air (Lean Combustion)

Total Dry Air

$$= \frac{Oxygen\ \mathrm{Re}quired\ in\ Combustion}{\%\ Oxygen\ in\ air}\ x\ (1 + EA) \tag{4}$$

Where; EA = Excess Air

iii) Total Wet air (With Moisture in air)

Total Wet Air;

$$= Total\ Dry\ air\ (Lean)\ x\ (1 + \omega) \tag{5}$$

Where; $\omega$ = weight of moisture in air/ weight of dry air

iv) Wet Flue Gas formed

= Excess Oxygen in Combustion air + ( % N2 in air + moist in air) (6)
*Combustion Air + Total Product of Combustion

v) Dry Flue Gas formed

= Total Wet Flue Gas Formed – (H20 in Combustion Product) (7)
-(H2 in Combustion Product)- (Moist in air)^2

### 4.3 Boiler Efficiency Calculation (ASME PTC 4.1)

ASME PTC 4.1 suggested two methods to analyze the boiler efficiency known as direct method and indirect method. The direct method is the simplest way to estimate the boiler efficiency but yet it is not the accurate way. The heat loss method is more accurate compared to the direct method and this method will be chosen for the study. Indirect method estimates the efficiency by considering various losses in the boiler such as Heat loss due to dry flue gas, Heat loss due to moisture in the fuel, Heat loss due to combustion of Hydrogen, Heat loss due to moisture in the air, Radiation heat loss, Unburned Fuel loss and other unaccounted losses. The losses that take place in the boiler are shown in figure 4.2.

Heat Losses;

L1 = Heat Loss due to dry Flue Gas

L2 = Heat Loss due to Combustion of Hydrogen

L3 = Heat loss due to Moisture in Fuel

L4 = Heat Loss due to moisture in air

L5 = Heat Loss due to Carbon Monoxide

L6 = Heat Loss due to Surface radiation, Convection & other unaccounted losses

L7 = Heat Losses due to fly ash losses

L8 = Heat Losses due to bottom ash losses

$$\text{Boiler Efficiency} = 100\% - (L1 + L2 + L3 + L4 + L5 + L6 + L7 + L8)\ \% \qquad (8)$$

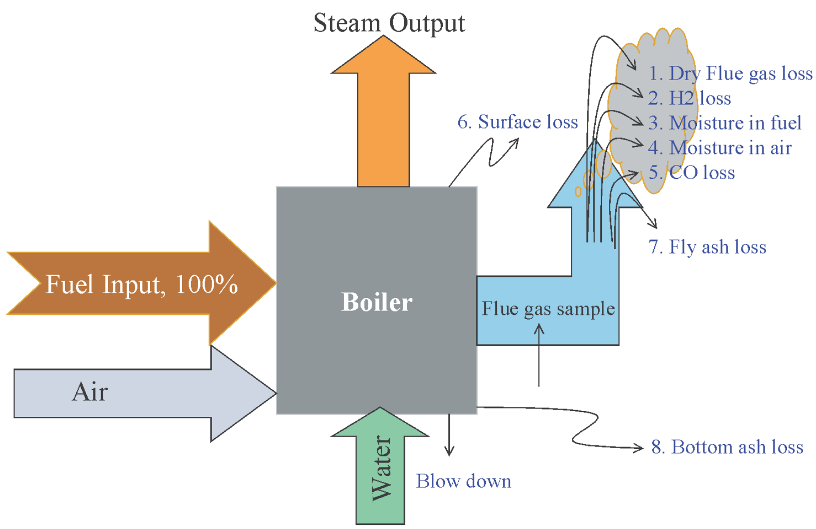



**Figure 4.2: Boiler efficiency by using indirect method[53]**

### 4.3.1 Heat Loss due to dry flue gas.

This is the major contributor compared to other losses in the boiler where the N2 enters the boiler as part of combustion and leaves at an elevated temperature causing energy loss. The heat loss can be calculated by using the following equation;

$$L1 = \frac{\dot{m} \; x \; C_P \; x \; (T_g - T_a)}{LHV} \; x \, 100 \qquad (9)$$

### 4.3.2 Heat loss due to combustion of hydrogen in fuel

Latent heat loss occurs when the water formed during hydrogen combustion carried away the heat due to the water latent heat content.

$$L2 = \frac{9 \; x \; H_2 \; x \; (h_l + C_P (T_g - T_a))}{LHV} \; x \, 100 \qquad (10)$$

#### 4.3.3 Heat loss due to moisture in fuel

The moisture in the fuel will be boiled as superheated vapour during combustion causing energy loss due to sensible heat content in the water vapour.

$$L3 = \frac{M\ x\,(h_l + C_P(T_g - T_a))}{LHV}\, x\, 100 \qquad (11)$$

#### 4.3.4 Heat loss due to moisture in air

Moisture content in the air known as humidity will be superheated as it passes the boiler and the sensible heat content in the water causes energy loss to occur. The humidity in combustion air can be obtained by using Psychometric Chart.

$$L4 = \frac{AAS\ x\ \omega + C_P(T_g - T_a)}{LHV}\, x\, 100 \qquad (12)$$

#### 4.3.5 Heat loss due to incomplete combustion

$$L5 = \frac{\%CO\ x\ C}{\%CO + \%CO2}\ x\ \frac{5744}{LHV}\, x\, 100 \qquad (13)$$

#### 4.3.6 Surface heat loss

Surface heat loss such as radiation and convection are the main factor reduces boiler efficiency. Convection heat loss has been neglected in this study due to the boiler location which is in a closed boiler house where only minimal air flow can be found.

Radiation heat loss estimated by using ABMA radiation heat loss graph based on the boiler heat output.

## 4.4 Heat Transfer in Boiler and Equipments

### 4.4.1 Heat Transfer in Furnace (Combustion Chamber)[18]

The steps of calculating Heat transfer in the furnace are as follows:

i) Boiler Output = $\dot{m}_s \ x \ (h_s - h_{fw})$ (14)

ii) $$Heat\ Input\ \mathrm{Re}quired, Q_{in} = \frac{Boiler\ Output}{Boiler\ Efficiency} \quad (15)$$

iii) $$Heat\ Credit, Q_{cr} = \dot{m}_{flue}\ x\ C_{P,flue}\ (T_{b,exit} - T_{AH,exit}) \quad (16)$$

iv) $$Net\ Heat\ Input, Q_{net} = F_{net}\ x\ Q_{in} + Q_{cr} \quad (17)$$

v) $$Heat\ \mathrm{Re}lease\ Rate, Q_{HRR} = \frac{Q_{net}}{A_s} \quad (18)$$

vi) Heat Available In Flue Gas, $Q_{available}$

$$= \dot{m}_{flue}\ x\ C_{p,fue}\ x\ (T_{FEGT} - T_{amb}) \quad (19)$$

vii) Heat Absorbed by furnace, Q $_{furnace}$

$$= Q_{net} - Q_{available} \tag{20}$$

### 4.4.2 Heat Transfer in Screen[18]

i) Heat Available In Flue Gas, Q $_{flue}$

$$= \dot{m}_{flue}\ x\ \ (C_{p,fue} T_{FEGT} - C_{p,fue} T_{f1}) \tag{21}$$

ii) Heat Absorbed by screen, Q $_{furnace}$

$$= U\ \ x\ A_{proj}\ \ x\ LMTD \tag{22}$$

iii) Heat Balance

Q $_{flue}$ = Q $_{screen}$

$$\dot{m}_{flue}\ x\ \ (C_{p,fue} T_{FEGT} - C_{p,fue} T_{f1}) = U\ \ x\ A_{proj}\ \ x\ LMTD \tag{23}$$

iv) Percentage of Heat Transferred

$$= \frac{Heat\,Transferred\ in\ Screen}{Boiler\ Output}\ x\,100\% \tag{24}$$

### 4.4.3 Heat Transfer in Super heater[18]

i) Steam Side

$$= \dot{m}_s\ \ x\,(h_s - h_{sat}) \tag{25}$$

ii) Flue Gas Side

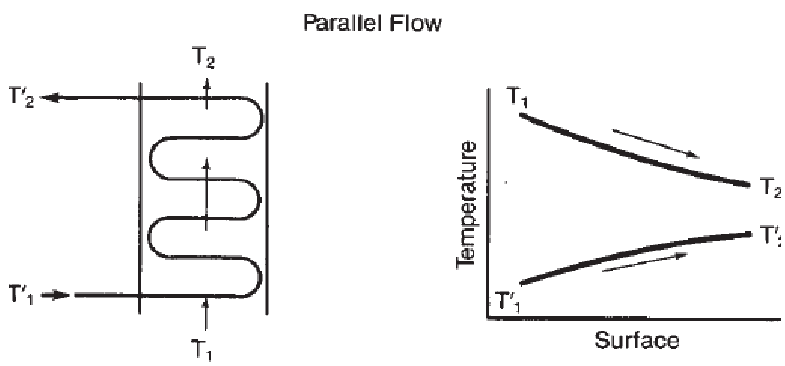
$$= \dot{m}_{flue}\ x\ (C_p T_{f1} - C_p T_{f2}) \tag{26}$$

iii) Heat Absorbed by Super heater, $Q_{SH}$

$$= U\ x\ A_{proj}\ x\ LMTD \tag{27}$$

iv) Percentage of Heat Transferred

$$= \frac{Heat\ Transferred\ in\ Superheater}{Boiler\ Output}\ x\ 100\% \tag{28}$$




$$\triangle T_{LMTD} = \frac{(T_1 - T'_1) - (T_2 - T'_2)}{\ell n \left( \frac{T_1 - T'_1}{T_2 - T'_2} \right)}\ ;\ F = 1.0$$

**Figure 4.3: Parallel flow heat exchanging[18]**

### 4.4.4 Heat Transfer in Boiler Bank[18]

i) Flue Gas Side= $\dot{m}_{flue}\ x\ (C_p T_{f2} - C_p T_{f3})$ (29)

ii) Heat Absorbed by Banks, $Q_{Bank}$

$$= U \ x \ A_{proj} \ x \ LMTD \tag{30}$$

iii) Percentage of Heat Transferred

$$= \frac{Heat\ Transferred\ in\ Banks}{Boiler\ Output} x\,100\% \tag{31}$$

### 4.4.5 Heat Transfer in Economizer[18]

i) Flue Gas Side

$$= \dot{m}_{flue} \ x \ (C_p T_{f3} - C_p T_{f4}) \tag{32}$$

ii) Water Side

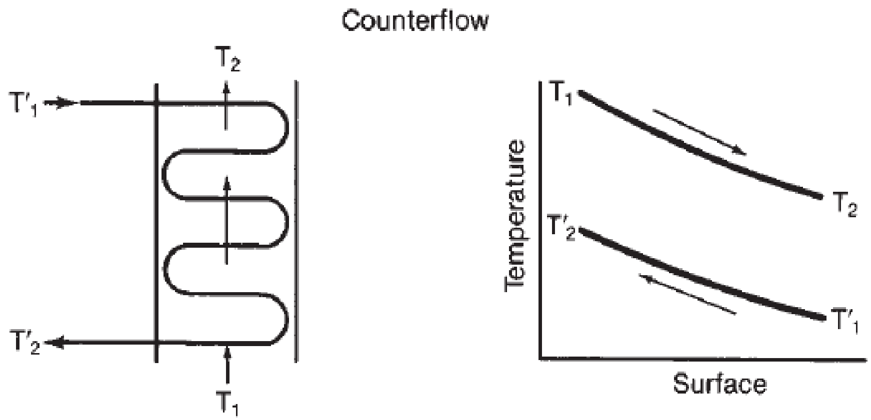
$$= \dot{m}_{fw} \ x \ (h_{hw} - h_{fw}) \tag{33}$$

iii) Heat Absorbed by Economizer , $Q_{Eco}$

$$= U \ x \ A_{proj} \ x \ LMTD \tag{34}$$

iv) Percentage of Heat Transferred

$$= \frac{Heat\ Transferred\ in\ Economizer}{Boiler\ Output} x\,100\% \tag{35}$$




$$\triangle T_{LMTD} = \frac{(T_1 - T'_2) - (T_2 - T'_1)}{\ell n \left( \frac{T_1 - T'_2}{T_2 - T'_1} \right)} \text{; } F = 1.0$$

**Figure 4.4: Counter flow heat exchanging[18]**

## 4.4.6 Heat Transfer in Air Preheater[18]

i) Flue Gas Side

$$= \dot{m}_{flue} \ x \ (C_p T_{f4} - C_p T_{f5}) \qquad (36)$$

ii) Air Side

$$= \dot{m}_{c.a} \ x \ (C_p T_{a,o} - C_p T_{a,i}) \qquad (37)$$

iii) Heat Absorbed by Air Preheater, $Q_{SH}$

$$= U \ x \ A_{proj} \ x \ LMTD \qquad (38)$$

iv) Percentage of Heat Transferred

$$= \frac{Heat\ Transferred\ in\ Air\ \mathrm{Pr}eheater}{Boiler\ Output} \ x\, 100\% \qquad (39)$$

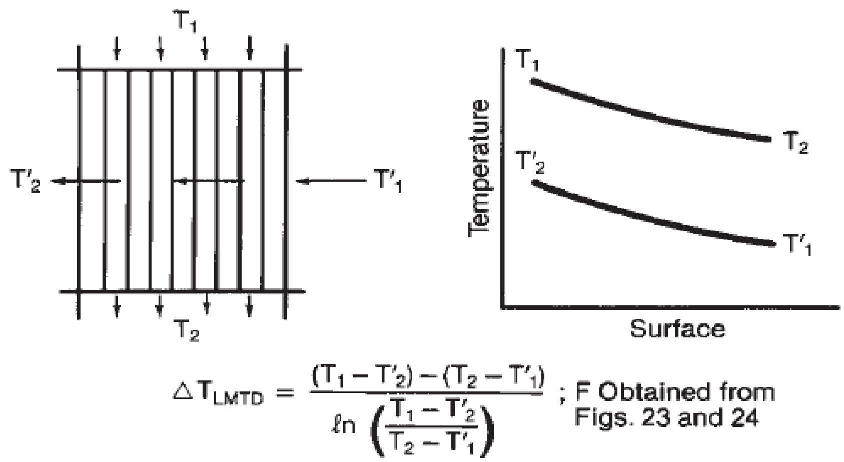



**Figure 4.5: Cross flow heat exchanging.[18]**

## 4.5 Heat Transfer Efficiency of Boiler Components.

The Heat transfer efficiency of boiler components surfaces obtained by using the expression as follow:

$$Q_{efficiency} = \frac{Heat\ transfer\ on\ Boiler\ Components}{Boiler\ Output}\ x\,100\% \qquad (40)$$

Boilers can be analyzed by 1$^{st}$ law analysis in order to determine the efficiency. The analysis can be separated into two major components which is the combustor or furnace and heat exchanging equipments. The energy efficiency can be determined by the following expression:

$$\eta = \frac{Energy\ in\ product\ outputs}{Energy\ input} \qquad (41)$$

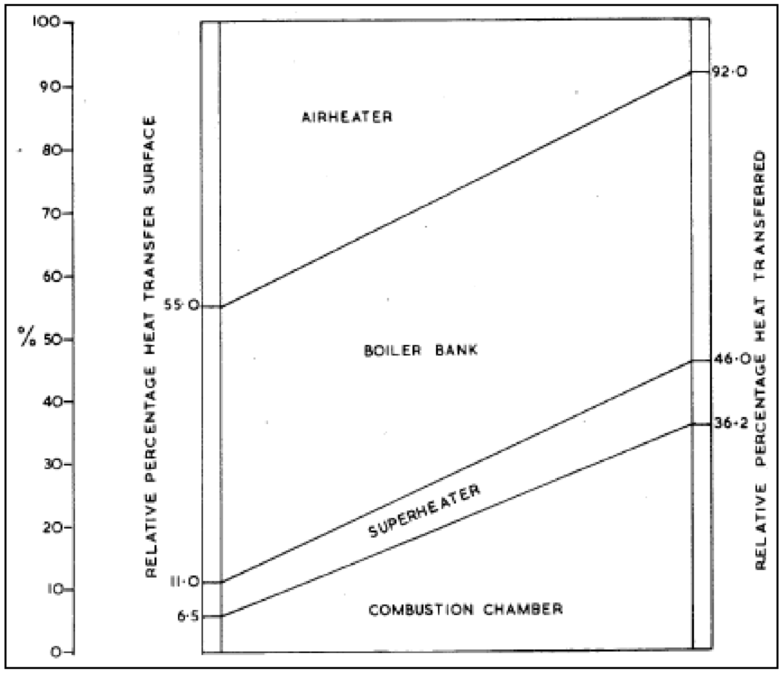



**Figure 4.6: Effectiveness of Heat Transfer Surfaces[1]**

## 4.6 Cost and Payback Period Analysis

The palm waste cost is considered as free because it's coming from the production line and not bought from outside plants .The payback period can be estimated by calculating the annual fuel saving that can be achieved and the selling price of the fuel. The income from the surplus fuel sold can be used to estimate the cost saving and payback period. Other than that the additional fuel that mills have can be used to run the production for extra hours or after shutdown where dry fuel is needed. The additional production hours can be used to produce more oil which is considered gain for the mill. The payback period can be calculated as follows:

$$Cost = Boiler\ \cos t + Installation + Commisioning + Labour \cos t \tag{42}$$

$$Annual\ Fuel\ usage = Consumed\ Palm\ waste\ Cost \tag{43}$$

$$\Pr oduction\, rate = Palm\, oil\ \ produced\ x\ \ price\ \ per\, tonne \tag{44}$$

$$Payback\ \ period = \frac{Cost + Annual\ \ Fuel\ \ Usage}{Annual\ \ Fuel\ \ Saving} \tag{45}$$

# CHAPTER 5: BACKGROUND OF STUDIES

## 5.1 Introduction.

Boiler design is mainly based on the empirical data of seasonal countries which have a lower ambient temperature and humidity. The study on the impacts of hot and humid tropical climate towards the boiler performance and heat transfer have not been carried out before this so there are less data available regarding this topic. A 35,000 kg/hr 3-pass palm oil mill water tube boiler located in Colombia firing on palm waste was chosen for the study. This study will be beneficial for them in order to understand their boiler performance and save their operational cost. Microsoft excel will be used to model the boiler where related equations used obtained from Babcock & Wilcox Steam book and ASME PTC4.1. The result will be verified later with the data collected from the site before different conditions are simulated.

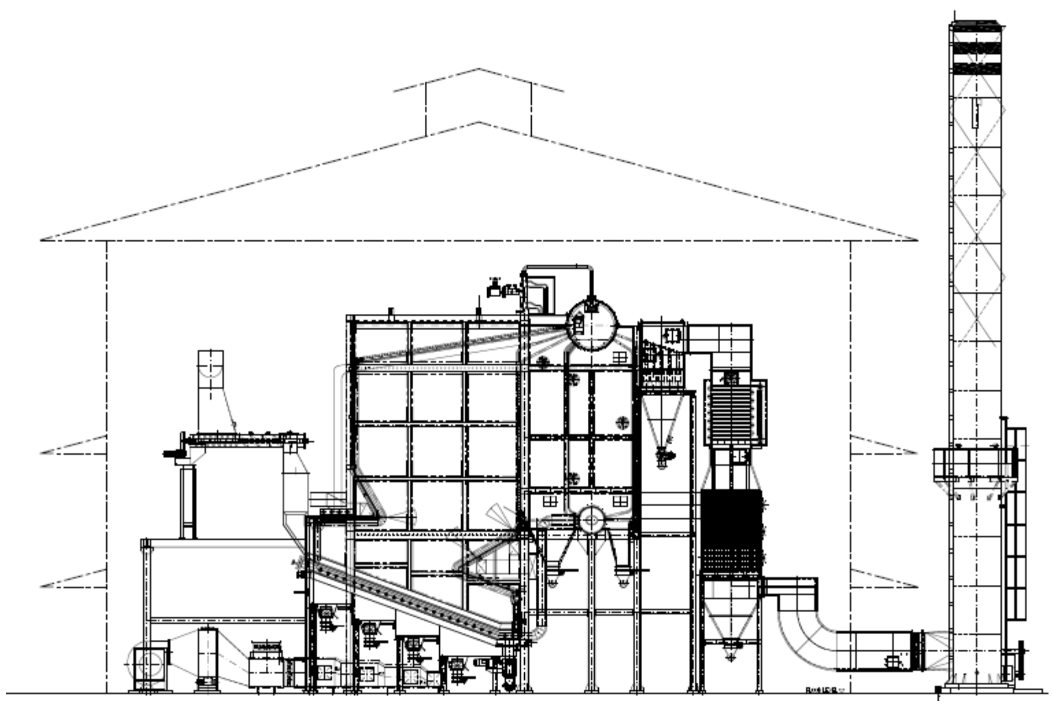


**Figure 5.1: Manuelita palm oil mill boiler.**

## 5.1 Site Condition.

The boiler site is located at the Manuelita Palm oil mill, Orocue Municipality Casanare, Colombia (4° 47’ 39.01”N, 71° 20’ 24” W) and has been operating for about 5 years. The boiler site experience hot and humid conditions as other tropical countries which make it suitable for the study. Environmental impacts on the boiler performance such as ambient temperature and Humidity will be studied thoroughly. A detailed study on the boiler component heat transfer efficiency and the component efficiency will be studied. The site conditions are as shown below:

Ambient Temperature: 77° F ~88° F

Humidity Range: 50% ~ 95%

## 5.2 Boiler Specification.

The boiler is a 35 ton/hr 3-pass water tube boiler firing on palm fiber and shell with high moisture content. The boiler specifications are listed in table 5.1 as follows:

| | |
|---|---|
| Ambient Temperature | : 27°C |
| Relative Humidity | : 85% |
| Wind Speed | : 1.0 m/s |
| Boiler Capacity | : 35000 kg/hr |
| Pressure | : 41 Barg |
| Temperature | : 370°C |
| Feedwater Temperature | : 105°C |

**Table 5.1: Boiler operating and site condition**

## 5.3 Fuel Specification

The boiler fuel specification normally given by the mill owners in the early stage of designing because the fuel properties may vary slightly from one place to another. The fuel specifications normally depend on the analysis done towards the palm waste. The fuel used in this boiler is a mixed type fuel which encompasses of 75% Palm Fiber and 25% Palm Shell combination while the moisture content is 35% and 15% respectively. The fuel average fuel properties will be determined through the ultimate analysis. The fuel specifications used in Manuelita are as shown in table 5.2:

| CONTENT | UNIT | Palm Fiber | Palm Shell |
|---|---|---|---|
| C | Wt.% | 47.20 | 52.40 |
| H2 | Wt.% | 6.00 | 6.30 |
| O2 | Wt.% | 36.70 | 37.30 |
| S | Wt.% | 0.30 | 0.20 |
| N2 | Wt.% | 1.40 | 0.60 |
| ASH | Wt.% | 8.40 | 3.20 |
| TOTAL | Wt.% | 100.00 | 100.00 |
| GCV | kcal/kg | 4,586 | 5,122 |

**Table 5.2: Boiler fuel Specification as given by Manuelita plant.**

## 5.4 Boiler equipments

### 5.4.1 Grate

The selected boiler is fitted with a 4-zone water-cooled reciprocating grate for the entire length of the grate system. The grate consists of 4 zones which is drying, de-volatilization, char-burning and de-ashing. This type of zone arrangement efficiently burns almost all types of fuels with average moisture content. The zones are controlled individually by using a hydraulic pusher which operates according to the burning time. The burning time varies depending on the blend ratio of fuel and moisture in the fuel mixture.

### 5.4.2 Furnace

Majority of the heat generated by combustion will be absorbed by the furnace wall through radiation while convection plays a minor part in the furnace. A bare water cooled furnace is used in order to bring down the furnace exit temperature lower than the ash fusion temperature. The dimension of the furnace supplied is as follows:

| | |
|---|---|
| Furnace Depth | : 6000 mm |
| Furnace Width | : 4650 mm |
| Furnace Height | : 10900 mm |
| Furnace Convective Surface | : 135.47 $m^2$ |
| Screen Convective Surface | : 31.18 m2 |
| Screen Flue gas flow area | : 5.18 m2 |

**Table 5.3: Furnace dimension and heating surface available.**

### 5.4.3 Super heater

A non-drainable super heater was used in the boiler in order to generate the required amount of steam with high temperature and pressure. A double loop design was used in order to maximize the heating surface of the super heater. Heat will be transferred to the super heater through convection from the hot flue gas to the saturated steam in the super heater tube. Saturated steam from the steam drum will enter the super heater and superheated by the hot combustion gas through convection. The super heater basically is a heat exchanging equipment and in this study we are using a parallel flow design. The convective area and flow area in the superheater area are as follows:

| | |
|---|---|
| Convective heating surface | 194.9 $m^2$ |
| Flue gas flow area | 7.02 $m^2$ |
| Steam flow area | 0.06 $m^2$ |

**Table 5.4: Heating Surface available in Super heater.**

### 5.4.4 Convection Banks

Convection bank in the site uses 3 pass arrangements in order to maximize the heating surface which will enhance the circulation rate in the tubes. The banks were arranged in three pass cross flow arrangement which reduces the outlet temperature of the flue gas. The details of the banks are as follows:

| | |
|---|---|
| Convective heating surface | 796.29 $m^2$ |
| Flue gas flow area | 4.304 $m^2$ |

**Table 5.5: Heating Surface available in Convection Bank**

### 5.4.5 Heat Recovery System

A bare tube Economizer and Air preheater were installed in the boiler system in order to increase the efficiency of the boiler by minimizing the stack temperature. Hot flue gases from the boiler exit were used to heat up the feed water from the deaerator into the steam drum through convection. The heated water will reduce the temperature difference with the saturated temperature and minimize the energy required.

Economizer design uses the counter flow concept where the inlet of the hot gas will be in the opposite direction with the inlet of the feed water. Heat absorption rate from the hot flue gas is better by using a counter flow heat exchanging design and it helps to reduce the stack temperature. Air preheater uses a cross flow design where the hot flue gas will be flowing into the tube and the hot air will be flowing outside of the tube. Cross flow concept was the best of all heat exchanging method because it helps to absorb heat from the hot gas efficiently before supplying it into the combustion chamber. Hot combustion air will be used to dry out the wet fuel so that the combustion will be more efficient. Convective area and flow area in the heat exchanging equipment are as follows:

| | |
|---|---|
| Economizer | |
| Convective heating surface | 138.44 $m^2$ |
| Flue gas flow area | 1.63 $m^2$ |
| Air Preheater | |
| Convective heating surface | 701.55 $m^2$ |
| Flue gas flow area | 2.35 $m^2$ |
| Air flow area | 2.96 $m^2$ |

**Table 5.6: Heating Surface available in Economizer and Air preheater.**

## 5.5 Test Condition

The test will be carried out for two major climate impacts which are the ambient temperature and relative humidity. Wind speed is considered negligible as the boiler is located in the boiler house and the wind speed convective heat transfer has minimum effect compared to radiation heat loss. The ABMA radiation chart was used to determine the heat loss to the surroundings based on boiler thermal output. The temperature range of the studies is set between 20°C and 35°C as the tropical climate ambient temperature is higher than 18°C. The relative humidity selected ranges from 50% to 100% based on the climate data of Malaysia and Colombia which is located in Tropical zones.  The simulated conditions are summarized in table 5.7:

| Temperature | Relative Humidity (%) | | | | | |
|---|---|---|---|---|---|---|
| °C | 50 | 60 | 70 | 80 | 90 | 100 |
| 20 | √ | √ | √ | √ | √ | √ |
| 25 | √ | √ | √ | √ | √ | √ |
| 30 | √ | √ | √ | √ | √ | √ |
| 35 | √ | √ | √ | √ | √ | √ |

**Table 5.7: Boiler simulation conditions**

## 5.6 Boiler costing

Costing for the boiler considers boiler cost which includes the valves, mountings, pumps, fans and controls. Other than that, other factors which have been considered are installation, commissioning, maintenance and labor cost . The maintenance and labor cost were considered for the maximum design life of a boiler which is about 15 years. The electrical cost was not considered as the electrical supply comes from the energy generated by boiler superheated steam which is used to run the turbine.

The investment cost of the boiler is as shown in table 5.8 below:

| No. | Item | RM |
|---|---|---|
| | | |
| 1 | Boiler | 3,850,000.00 |
| 2 | Installation + Commissioning | 316,800.00 |
| 3 | Maintenance cost (15years) | 100,000.00 |
| 4 | Labor cost (3 person for 15 years) | 810,000.00 |
| 5 | Fuel Consumption Cost | 0.00 |
| | (24hrs x 6days x 52 week) | |
| | Total | 5,076,800.00 |

**Table 5.8: Investment cost of Boiler**


**Figure 5.2: Reciprocating grate internal view**

**Figure 5.3: Reciprocating grate during Combustion**

# CHAPTER 6: RESULT AND DISCUSSION

The boiler components heating surface and free flow area information is listed in the table 6.1.

| Component | Convective | Flow Area | | |
|---|---|---|---|---|
| | Heating Surface | Flue | Steam | Air |
| | $m^2$ | $m^2$ | $m^2$ | $m^2$ |
| Furnace | 135.47 | | | |
| Screen | 31.18 | 5.18 | | |
| Superheater | 197.51 | 7.02 | 0.06 | |
| Cavity | 27.94 | | | |
| Boiler Bank | 835.08 | 4.30 | | |
| Economizer | 128.50 | 1.68 | | |
| Air Preheater | 701.55 | 2.35 | | 2.96 |

**Table 6.1: Boiler Components heating surface and free flow area.**

## 6.1 Effect Relative Humidity towards the Boiler Overall Efficiency

Figure 6.1 shows the impact of moisture in the air at different ambient temperature towards the boiler overall efficiency. The lowest boiler efficiency is 74.74% obtained at 20 °C and 100% humidity while the highest efficiency is 75.62% obtained at 35 °C and 50% humidity. The efficiency reduces as the moisture content in the air increases from 50% to 100% because the amount of heat carried away is higher. The water vapor in the air will be heated up while passing through the boiler from ambient temperature to the flue gas temperature while leaving the boiler. The moisture carried out in the form of sensible heat where energy will be used to heat up the water which leads into a higher loss percentage. The efficiency rises significantly when the ambient temperature increases while maintaining the humidity level. The efficiency increase ranges from 0.66% to 0.71% when the temperature increased from 20°C to

35°C. Higher ambient temperature reduces the energy required to heat up the moisture in the air and reduces the heat loss.

| T℃ | Relative Humidity | | | | | |
|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 90 | 100 |
| 20 | 74.91 | 74.87 | 74.84 | 74.79 | 74.77 | 74.74 |
| 25 | 75.16 | 75.12 | 75.07 | 75.03 | 74.99 | 74.95 |
| 30 | 75.41 | 75.35 | 75.29 | 75.24 | 75.18 | 75.12 |
| 35 | 75.62 | 75.55 | 75.50 | 75.48 | 75.45 | 75.42 |

**Table 6.2: Boiler efficiency at different ambient temperature and humidity.**

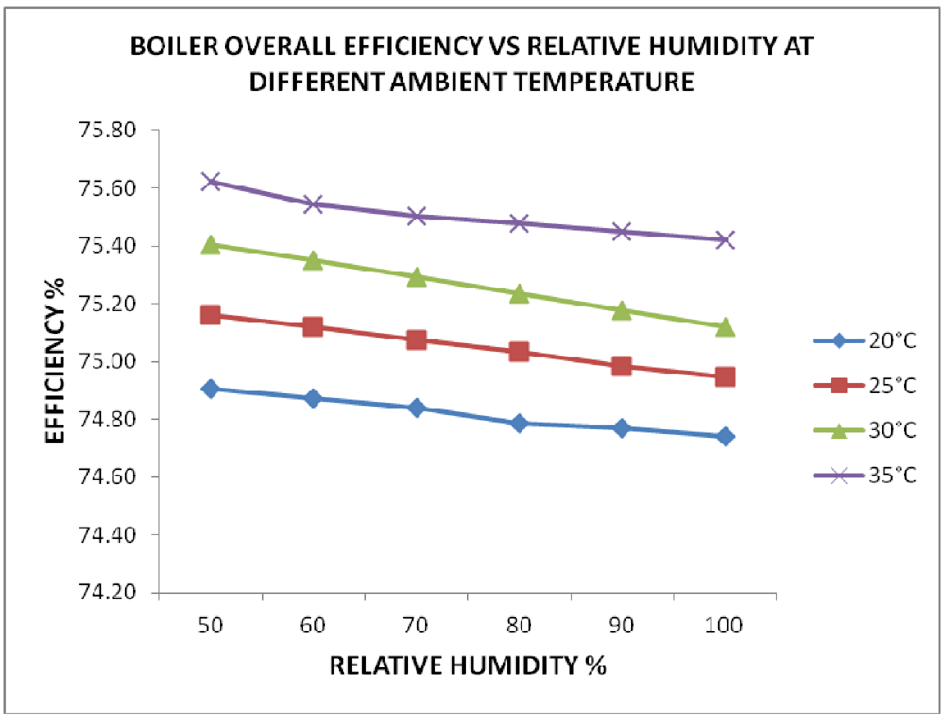



**Figure 6.1: Boiler Overall Efficiency vs Relative Humidity at different ambient temperature**

## 6.2 Effect Relative Humidity towards the Boiler Fuel Consumption

Figure 6.2 shows the fuel consumption for the specific boiler increases with increasing of moisture content in the air but reduces with increasing ambient temperature. The maximum fuel consumption is 9078.16 kg/hr while the minimum fuel consumption is 8972.57 kg/hr. The fuel demand increases as the moisture content goes higher in order to compensate the sensible heat taken out by the water vapor. The fuel demand goes lower with increasing ambient temperature because less energy needed to compensate the energy loss due to moisture in the air. The fuel demand increases from minimum 81.87kg/hr to 85.47 kg/hr when the ambient temperature decreases from 35°C to 20°C. Higher ambient temperature closes the temperature gap between the combustion temperature and the surrounding temperature causing less energy required.

| T℃ | Relative Humidity | | | | | |
|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 90 | 100 |
| 20 | 9058.04 | 9062.27 | 9066.24 | 9072.59 | 9074.45 | 9078.16 |
| 25 | 9027.29 | 9032.36 | 9037.95 | 9043.04 | 9048.64 | 9053.48 |
| 30 | 8998.22 | 9004.79 | 9011.86 | 9018.45 | 9025.53 | 9032.38 |
| 35 | 8972.57 | 8981.44 | 8986.46 | 8989.62 | 8993.01 | 8996.29 |

**Table 6.3: Boiler fuel consumption at different ambient temperature and humidity.**

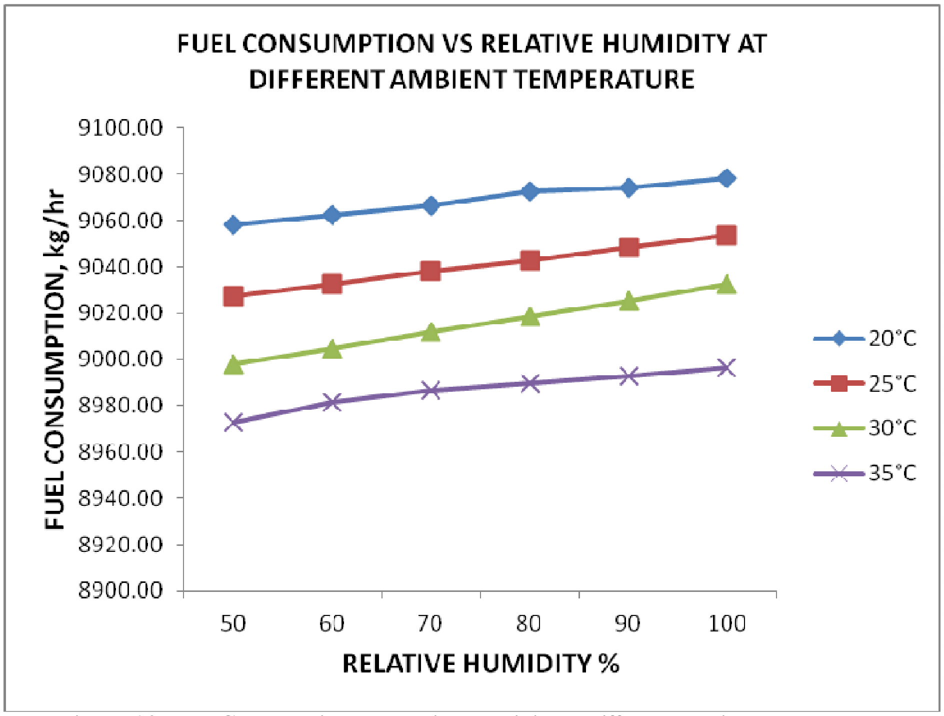



**Figure 6.2: Fuel Consumption vs Relative Humidity at different ambient temperature**

## 6.3 Effect Relative Humidity towards the Furnace Exit Gas Temperature (FEGT)

Furnace exit gas temperature (FEGT) increases along with relative humidity for a specific ambient temperature. From the data shown in table 6.4 it is noticed that the minimum FEGT 943.83°C occurs at the highest ambient temperature which is 35°C and 50% humidity while the maximum FEGT is 947.60°C at 20°C and 100% humidity. FEGT depends on the heat release rate or heat flux in the furnace and it increases as the net heat input required by the furnace increased due to lower efficiency of the boiler. Increase of relative humidity causes the heat to be transferred out through the moisture sensible heat capacity where the heated moisture is taken out together with the flue gas. Heat gain by the moisture in the flue gas increase the temperature and raises the FEGT temperature.

Furthermore from the graph it can be noticed that the FEGT is lower when the ambient temperature was increased from 20°C to 35°C where an increment of 2.77°C to 2.94°C is noticed for 50% and 100% relative humidity. Higher ambient temperature increases the boiler efficiency and it reduces the net heat input required for the boiler to generate the required steam capacity. Reduction in the net heat input may cause the heat release rate or heat flux required to be lower and this may cause the FEGT to drop. Furnace exit gas temperature need to be lower than the ash fusion temperature to prevent clinkering which might happen at the furnace exit zone. Clinkering happens when FEGT is higher than the ash fusion temperature or known as ash melting temperature. High temperature will cause the ash to evaporate into the flue gas and solidified when it cools down on the way out of the boiler along the flue gas path which led into clinkering at the super heater and screen areas.

| T℃ | Relative Humidity | | | | | |
|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 90 | 100 |
| 20 | 946.77 | 946.94 | 947.11 | 947.37 | 947.44 | 947.60 |
| 25 | 945.70 | 945.91 | 946.14 | 946.35 | 946.58 | 946.78 |
| 30 | 944.69 | 944.97 | 945.26 | 945.54 | 945.83 | 946.12 |
| 35 | 943.83 | 944.21 | 944.42 | 944.55 | 944.69 | 944.83 |

**Table 6.4: Furnace Exit Gas Temperature (FEGT) at different ambient temperature and humidity**

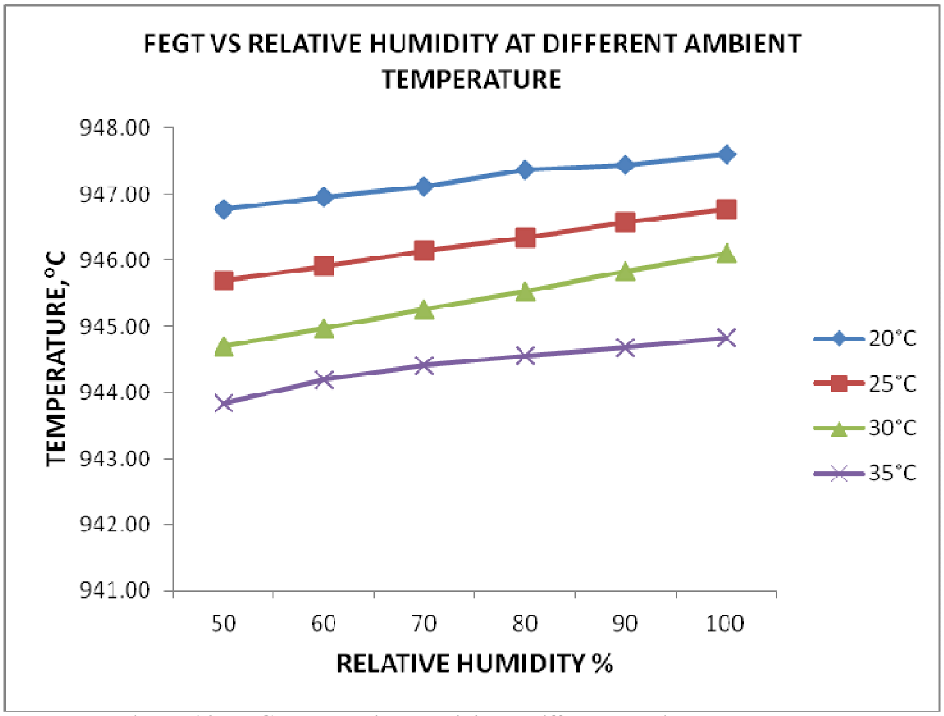



**Figure 6.3: FEGT vs Relative Humidity at different ambient temperature**

## 6.4 Effect Relative Humidity towards the Combustion Temperature

Highest fuel combustion which is 1518.2°C occurs at 25°C and 50% relative humidity and it reduces about 7.4°C as the relative humidity increases to 100%. The moisture particles in the air will absorb the heat at the combustion zone and reduces temperature. From the graph shown in figure 6.4 it is noticed that the combustion temperature decreases as the ambient temperature increases. The combustion temperature decreases in the range of 7.40°C to 13.50°C when the humidity increases from 50% to 100% along with the ambient temperature increment from 20°C to 35°C. Higher ambient temperature reduces the energy input required and increased the flue gas mass flow which may reduce the enthalpy of combustion which led into lower combustion temperature.

| T℃ | Relative Humidity | | | | | |
|---|---|---|---|---|---|---|
| | 50 | 60 | 70 | 80 | 90 | 100 |
| 20 | 1453.48 | 1451.99 | 1450.59 | 1448.36 | 1447.71 | 1446.41 |
| 25 | 1452.35 | 1450.48 | 1448.44 | 1446.59 | 1444.56 | 1442.81 |
| 30 | 1450.47 | 1447.97 | 1445.30 | 1442.82 | 1440.16 | 1437.60 |
| 35 | 1447.09 | 1443.60 | 1441.63 | 1440.40 | 1439.08 | 1437.80 |

**Table 6.5: Combustion Temperature at different ambient temperature and humidity.**

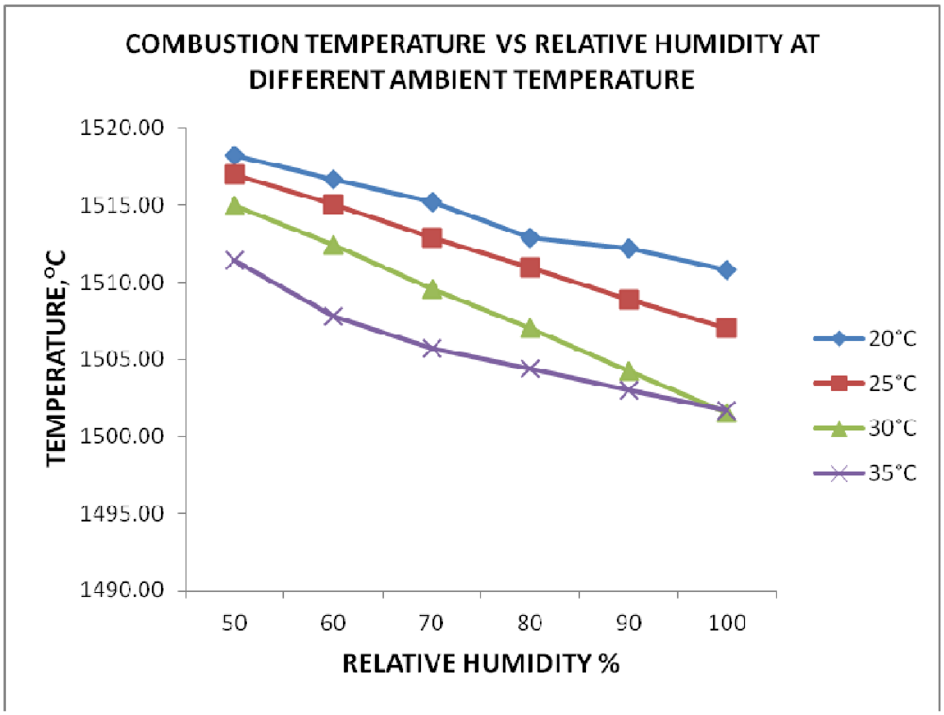



**Figure 6.4: Combustion Temperature vs Relative Humidity at different ambient temperature**

## 6.5 Effect Relative Humidity towards the Heat Exchanging Equipment Efficiency

### 6.5.1 Superheater

Figure 6.5 shows the efficiency of superheater drops averagely 0.51% to 0.69% as the ambient temperature increases from 20°C to 35°C while the relative humidity increases from 50% to 100%. The maximum efficiency for the superheater is 92.68% while the minimum efficiency calculated is 85.91%. Superheater uses the principle of parallel flow where the flue gas flows along with the steam in the superheater tube. The steam will be heated from saturated temperature to the required superheated steam temperature where the heat duty can be calculated for the required steam flow.

Increase in ambient temperature reduces the FEGT of the flue gas and the heat will be absorbed by the superheater heating surface in order to heat up the water. The Log Mean Temperature Difference of the superheater increases as the ambient temperature increases which can be used to calculate the outlet temperature of the combustion gas. Flue gas with higher outlet temperature reduces the temperature difference between the flue gas inlet temperatures. Reduction in temperature difference will reduce the flue gas heat duty that leads into higher efficiency of the equipment. Furthermore Flue mass flow reduces as the temperature increases and this will minimize the heat losses which happen during heat exchanging.

Relative Humidity effect on the super heater efficiency can be seen in figure 6.5 where the efficiency drops as the humidity increases. Rise in humidity will increase the mass flow of the flue gas and the heat loss in the equipment. Apart from that increase in moisture content raises FEGT and reduces the flue gas outlet temperature which causes higher heat duty needed and drops in efficiency.

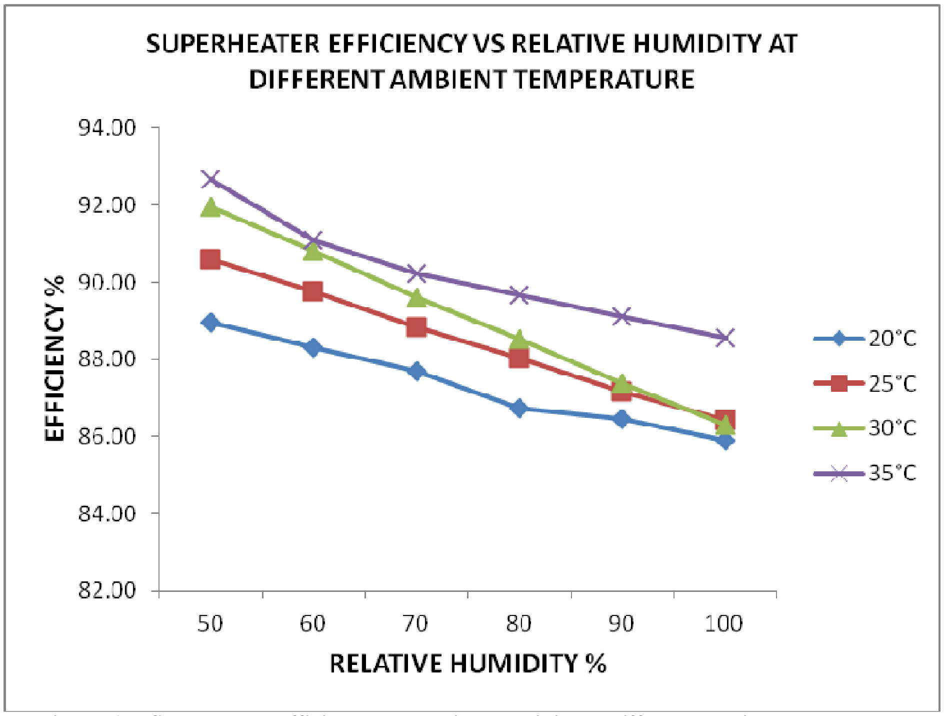


**Figure 6.5: Superheater Efficiency vs Relative Humidity at different ambient temperature**

### 6.5.2 Economizer

Economizer size was designed as a counter flow heat exchanging equipment in order to give a better efficiency. The flue gas outlet was maintained at 240°C where the flue gas inlet varies based on the flue gas temperature from the convection bank 3$^{rd}$ pass outlet. The water inlet temperature from the deaerator will be maintained at 105°C and the outlet temperature will be varied based on the flue gas heat duty. The maximum efficiency of the economizer is about 99.15% while the minimum efficiency is 98.10%. The economizer efficiency drops approximately about 0.8% as the relative humidity increases from 50% to 100% but there is slightly uncertainty in the efficiency when the ambient temperature was increased. Increase in moisture content in the air will cause more heat loss in the equipment during heat exchanging. The LMTD and heat transfer coefficient of economizer were maintained almost the same when the ambient temperature was increased, which means the heat duty will remain almost the same for the same heat exchanging surface area available. This differs compared to moisture in the air where LMTD remains the same but the overall heat transfer coefficient increases causing the flue gas heat duty to increase. The efficiency of economizer drops as the humidity increases because the ratio between the cold fluid and hot fluid drops due to the increase of the hot flue gas heat duty.

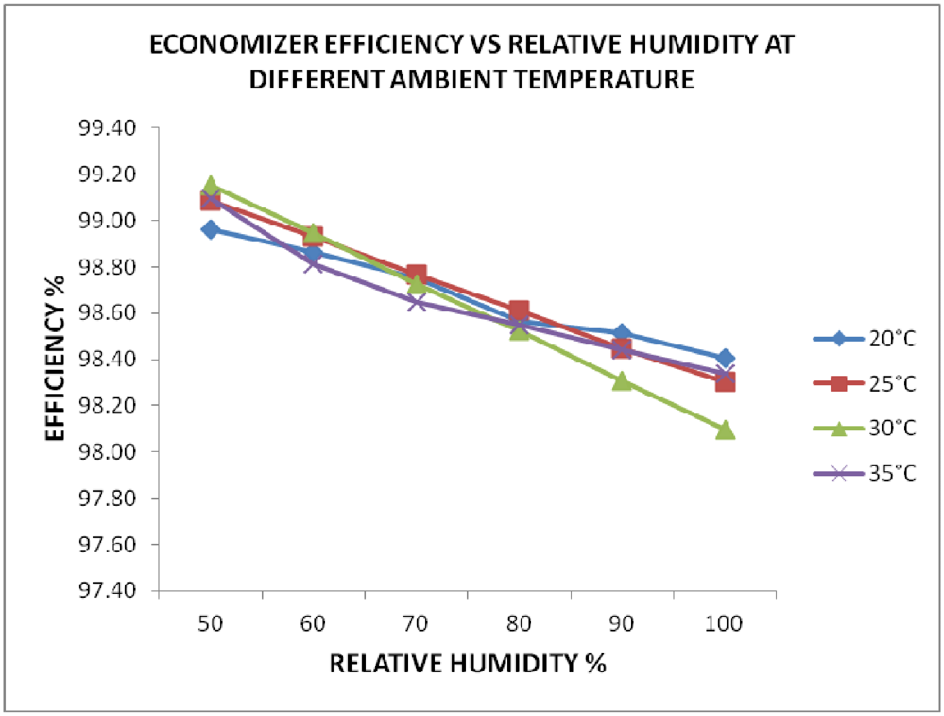



**Figure 6.6: Economizer Efficiency vs Relative Humidity at different ambient temperature**

### 6.5.3 Air Preheater

Air preheater designed as cross flow heat exchanging device and it acts as the best heat exchanging method. The efficiency of air preheater increases as the ambient temperature increases but the efficiency remains almost the same as the humidity ratio increases. From figure 6.7 it is noticed that the highest efficiency is 102.57% and the minimum efficiency is 94.97%. It is noticed that the efficiency drops about 7.60% as the temperature goes from 35°C to 20°C and it remains same as the relative humidity increase from 50% to 100%. The efficiency went more than 100% when the temperature is higher than 30°C because the heat duty of combustion air is higher compared to the flue gas heat duty. The flue gas outlet temperature was maintained at 170°C and the heat duty was maintained for the hot flue gas. The combustion air heat duty increases as the ambient temperature drops. Higher hot fluid heat duty reduces the efficiency of the equipment as the flue gas heat duty has remained the same.

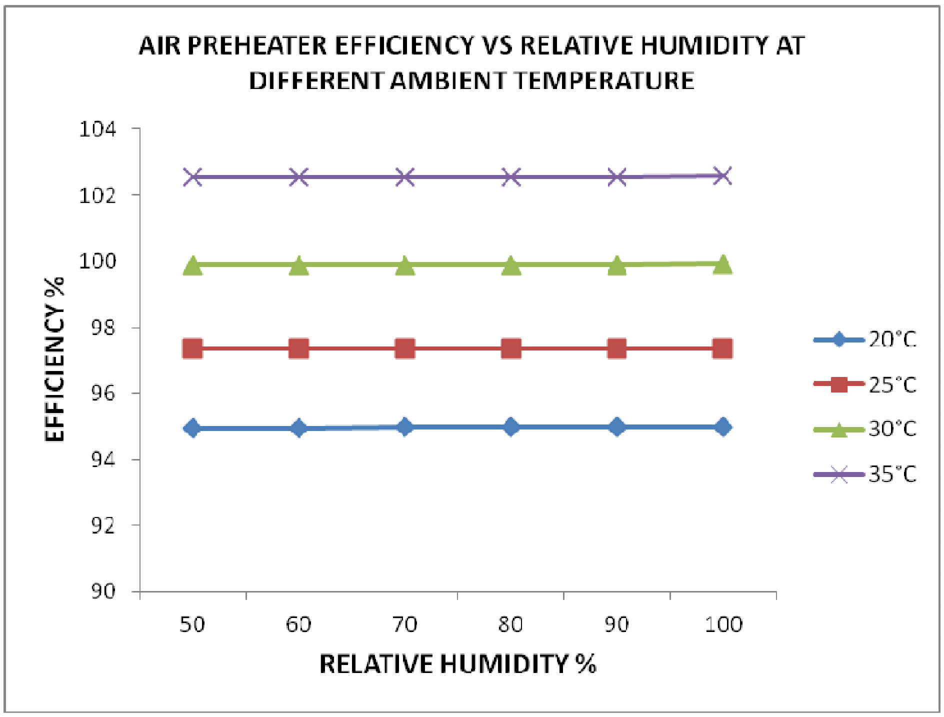



**Figure 6.7: Air Preheater Efficiency vs Relative Humidity at different ambient temperature**

## 6.6 Effectiveness of Boiler Heat Transfer surfaces

Table 6.6 shows the effective heating surfaces percentage available in Manuelita 35Ton boiler and the percentage of relative heat transferred to the surface at 27°C and 85% humidity. From the table it is shown that the largest heat transfer surface is the convection bank area which holds 40.59% of the total area. The minimum area available is the cavity where it holds 1.36% of the total area available. The largest heat transferred occurs in the convection bank and the minimum heat transferred is in the cavity area. Furnace has the second highest amount of heat transferred which is about 33.41% because the combustion heat is absorbed by the water cooled walls. The heat transferred in the superheater which is about 11.76% plays an important role in order to supply steam flow with the desired pressure and temperature. Other than that heat exchanging equipments such as the economizer and air preheater have 6.25% and 34.10% of heat transfer area respectively in order to exchange 6.31% and 5.73% of the total boiler thermal output.

Figure 6.9 to 6.12 shows the fluctuation of total heat transferred percentage of the given heat transfer surface at different ambient temperature and humidity. The heat transfer percentage increases as the humidity increases for all the equipments except for the furnace where the heat transfer percentage reduces averagely about 0.5%. The heat transfer rate reduction is due to the less heat input and low flue gas exit temperature when the moisture content in the air increases. Boiler cavity seems to have the lowest heat transfer where the fluctuation is about 0.01% of the total heat transfer for different ambient condition and humidity.

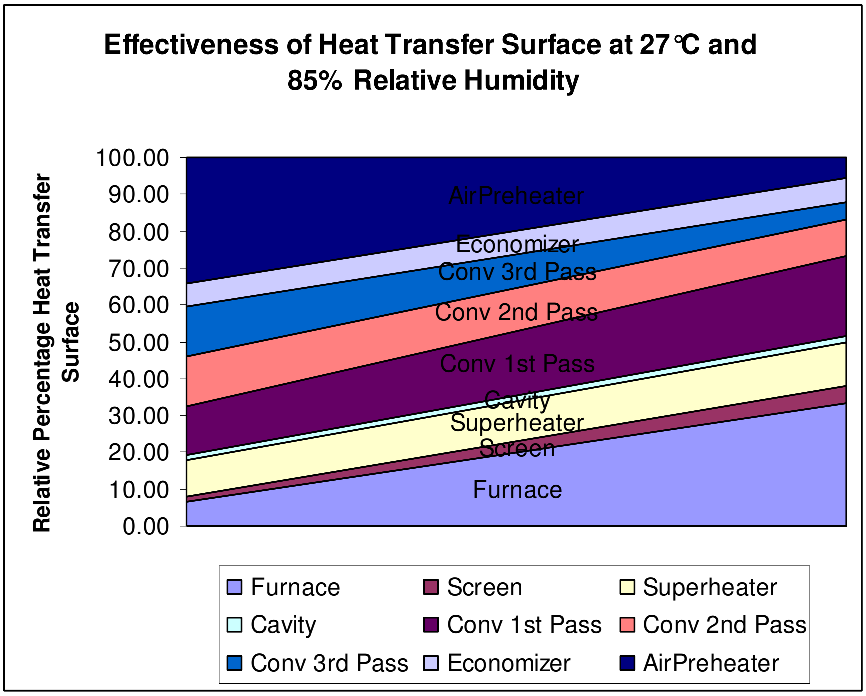



**Figure 6.8: Effectiveness of Boiler Heat Transfer surfaces.**

| Boiler Heat Transfer Surface | Relative Heat Transfer surface | Relative Heat Transferred (%) |
|---|---|---|
| Furnace | 6.59 | 33.41 |
| Screen | 1.52 | 4.82 |
| Superheater | 9.60 | 11.76 |
| Cavity | 1.36 | 1.61 |
| Conv 1st Pass | 13.53 | 21.50 |
| Conv 2nd Pass | 13.53 | 9.94 |
| Conv 3rd Pass | 13.53 | 4.92 |
| Economizer | 6.25 | 6.31 |
| AirPreheater | 34.10 | 5.73 |

**Table 6.6: Effectiveness of Heat Transfer surfaces of Boiler Heat transfer components at 27°C and 85% humidity.**

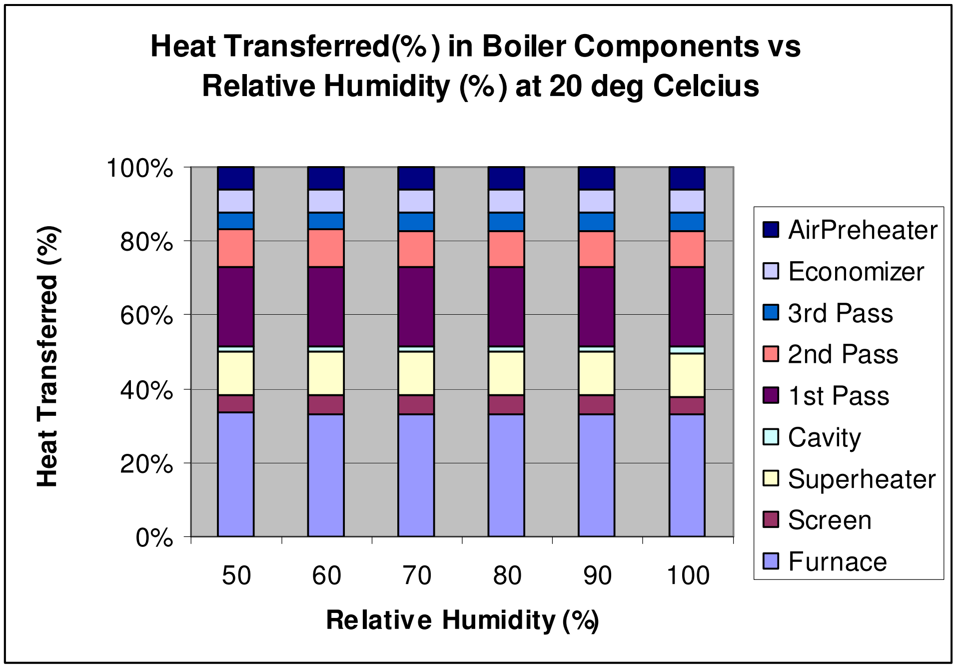



**Figure 6.9: Effectiveness of Heat Transfer surfaces vs Relative Humidity for boiler Heat transfer . Components at 20°C**

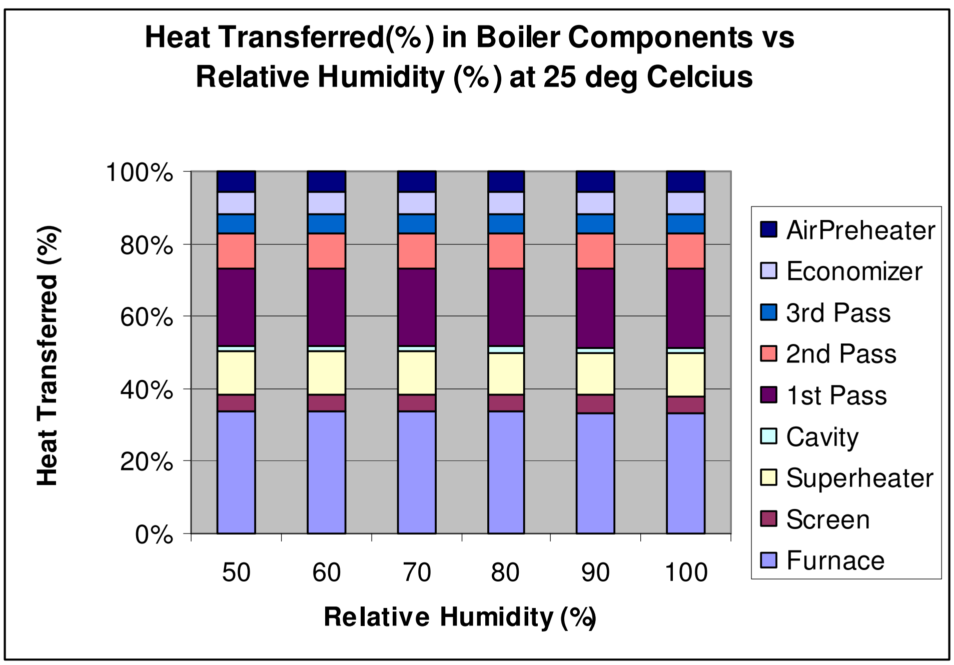



**Figure 6.10: Effectiveness of Heat Transfer surfaces vs Relative Humidity for boiler Heat transfer Components at 25°C.**

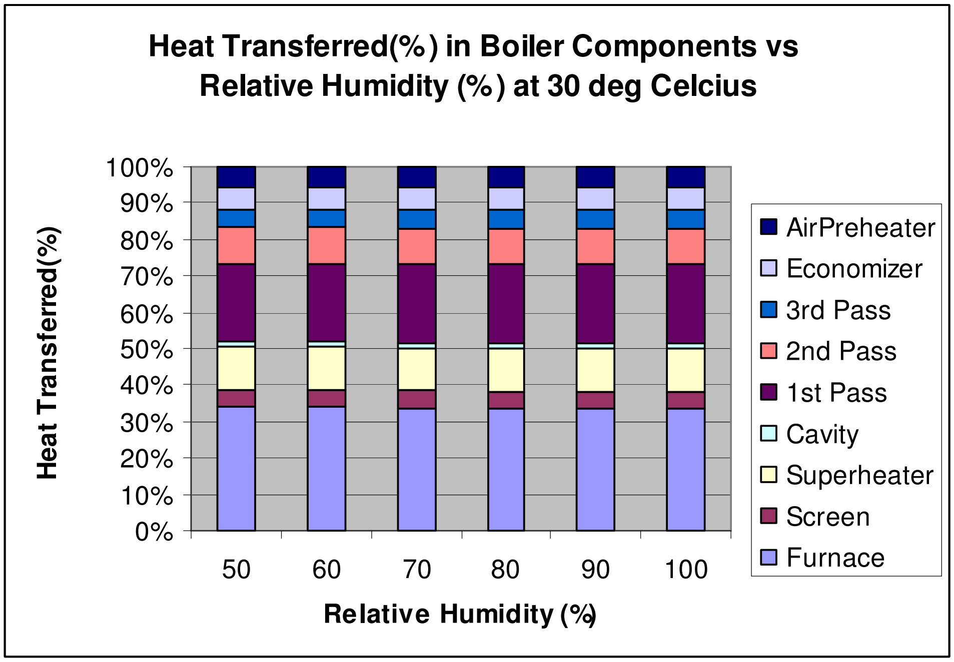



**Figure 6.11: Effectiveness of Heat Transfer surfaces vs Relative Humidity for boiler Heat transfer Components at 30°C.**

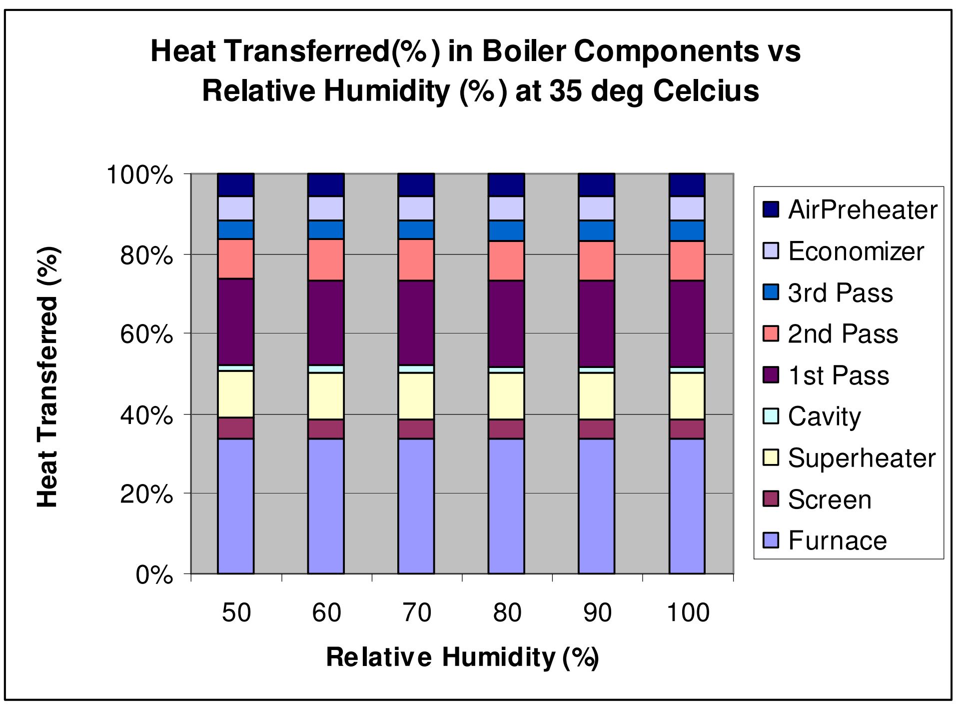



**Figure 6.12: Effectiveness of Heat Transfer surfaces vs Relative Humidity for boiler Heat transfer Components at 35°C.**

## 6.7 Cost Analysis and Payback Period

Cost analyses for the best and worst condition are shown as follows:

CPO production rate = 12.08 ton/hr

CPO selling price = Rm 2536.30 per tonne

Fiber market value = Rm 810 per tonne

Shell market Value = Rm 255 per tonne

Mixture cost = Rm 643.50 per tonne

Highest fuel consumption = 9078.20 kg/hr (20°C and 100% RH)

### 6.7.1 Cost Analysis and Payback based on Fuel selling price.

a) Lowest fuel consumption = 8972.60 kg/hr (35°C and 50% RH)

Fuel Saved = 105.60kg/hr

Annual Fuel saving = 105.60kg/hr x 24hr x 6days x 52week

= 790.64 ton/annual

Cost saving = 790.64 ton/annual x Rm 643.50/ton

= Rm 508,778.00/annual

Payback period = Rm 5,076,800 / Rm 508,778.00

= 9.98 years

b) Site fuel consumption = 9033.30 kg/hr (27°C and 85% RH)

Fuel Saved = 44.90 kg/hr

Annual Fuel saving = 44.90 kg/hr x 24hr x 6days x 52week

= 336.22 ton/annual

Cost saving = 336.22 ton/annual x Rm 643.50/ton

= Rm 216,357.60/annual

Payback period = Rm 5,076,800.00 / Rm 216,357.60

= 23.5 years

### 6.7.2 Cost Analysis and payback based on additional production days

a) Lowest fuel consumption = 8972.60 kg/hr (35°C and 50% RH)

Annual Fuel saving = 105.60kg/hr x 24hr x 6days x 52week

= 790.64 ton/annual

Total additional hours = 790.64 / 8.972 hr/annual

= 88.12 hrs

Additional CPO production = 88.12hrs x 12.08 ton/hr

= 1064.49 ton

| | |
|---|---|
| Additional savings | = 1064.49 ton x Rm 2536.30/ton |
| | = 2,699,865.99 |
| | = 2,699,866.00 |
| Payback period | = Rm 5,076,800 / Rm 2,699,866 |
| | = 1.88 years |
| b) Site fuel consumption | = 9033.30 kg/hr (27°C and 85% RH) |
| Annual Fuel saving | = 44.90 kg/hr x 24hr x 6days x 52week |
| | = 336.22 ton/annual |
| Total additional hours | = 336.22 / 9.033 hr/annual |
| | = 37.22 hrs |
| Additional CPO production | = 37.22 hrs x 12.08 ton/hr |
| | = 449.62 ton |
| Additional savings | = 449.62 ton x Rm 2536.30/ton |
| | = 1,140,371.21 |
| Payback period | = Rm 5,076,800 / Rm 1,140,371.21 |
| | = 4.45 years |

Cost saving estimated by using fuel re-sell value and additional production days that could be gained annually. The comparison between two ways of estimating cost saving shows that the additional income from additional production days is relatively higher.

Additional income obtained by running extra hours of production can reduce the payback period of the boiler. The payback period for site condition shows that it can be achieved within the period of 4.45 years if the saved fuel can be used for running extra hours of production instead of selling the surplus fuel. Running the plant for extra hours shows that it is more feasible compared to selling the fuel since the maximum boiler guarantee that normally given is 15years before major maintenance should be carried out.

# CHAPTER 7: CONCLUSION AND RECOMMENDATION

## 7.0 Introduction

The impact of tropical climate towards boiler performance and heat transfer of the boiler components is the main focus for this study. The variables that have been considered in this study are the relative humidity and the ambient temperature of the air. Manuelita 35 ton/hr boiler was simulated by using spreadsheet with a certain margin of error to represent the actual boiler condition and the effects of the variables are discussed briefly in the previous chapter. Conclusion of the study will be given based on the discussion made where it will include recommendations on the design in order to improve the efficiency. Other than that suggestion for future studies and methodology will also be made to find out how other factors might affect the boiler and its component efficiency.

## 7.1 Conclusion

Boiler efficiency plays an important role in order to minimize the fuel consumed and increase the evaporation ratio of the particular boiler. Tropical climate impact towards the boiler efficiency was studied in terms of ambient temperature and relative humidity. The results show that the efficiency becomes higher as the ambient temperature increased which results into lower fuel consumption and increased in the evaporation ratio. The heat input required was also reduced as the temperature increased due to reduction in heat loss but the required heat input increased as the humidity increases.

Heat loss due to dry flue gas linked with Nitrogen ($N_2$) that enters the together with the combustion air and takes part in the combustion before leaves the combustion chamber at a higher temperature together with the product of combustion. Increase in ambient temperature will cause the heat loss due to dry flue gas to drop while it the heat loss increases as the humidity rises. Sensible heat loss due to moisture in fuel decreasing as the ambient temperature becomes higher but the sensible heat loss remains the same when the humidity in the air increases. Sensible heat loss due to moisture in the air remains the same as the ambient temperature gets higher but it increases as the moisture in the air becomes higher. The water vapour entering the boiler will be heated until it reaches the boiling point where the sensible heat capacity of the moisture will cause the heat to be taken out. Water vapour formed during combustion of hydrogen causes heat loss because the water vapour in the product of combustion will be superheated and it leaves the boiler as latent heat.

Heat transfer rate in furnace increases as the ambient temperature gets higher but it decreases as the moisture content in the air is higher. The heat transfer rate in other components reduces as the ambient temperature increases but reduce with increasing humidity. The efficiency of the heat exchanging components shows that better efficiency are obtained at higher ambient temperature but at lower humidity.

## 7.3 Recommendations

The ultimate engineering goal is to solve problems that arise which are in this case the design of the boiler for tropical climate which is known for high ambient temperature and humidity. From the analysis it is found that the combustion chamber exit temperature increases as the humidity increases. We could propose water cooled membrane wall type combustion chamber to increase the absorption rate of furnace wall and bring down the furnace gas exit temperature. Other than that, super heater efficiency can be increased by using a cross flow type super heater. Higher combustion air and feed water temperature can be used to reduce the fuel consumption. Higher humidity throughout the year in a tropical country causes the fuel moisture to increase and this will reduce the efficiency once fed into the boiler. Supplying hot air extracted from the hot flue gas to the fuel storage will help to dry the fuel before being supplied to the boiler. Dry fuel will reduce the heat losses in the boiler and increase the efficiency and production hours.

## 7.4 Future works

There are other factors that can be considered in order to carry out studies for the tropical boiler. Factor such as the fuel moisture effect towards the boiler efficiency plays an important role in the boiler design. This study will be important in a tropical country that has a hot and humid condition throughout the year because it is not possible to provide fuel with designed moisture content. The study on mixture ratio affect towards boiler efficiency cannot be neglected as it has a major effect on the boiler sizing and not all the time the mill owners can provide the proposed fuel mixture as it depends on the factory production.

Boiler simulation can be studied more efficiently in the future by using Computational Fluid Dynamics modelling (CFD). From the literatures it is found that ANSYS is one of the simulation packages that can be used for the boiler studies as it predicts the heat transfer and flue gas flow more accurately with minimum error percentage. ANSYS provides a better result as it helps us to study the dynamics that happen in the boiler. User Defined Function (UDF) need to be developed separately while modelling combustion in ANSYS. UDF helps to simulate the actual combustion process that happen in the boiler accurately.

# BIBLIOGRAPHY


1. Magasiner, N. *Boiler Design an Selection in the Cane Sugar Industry*. in *Proceedings of The South African Technologist Association*. 1966.
2. Zhang, X., et al., *Experimental investigation and mathematical modelling of wood combustion in a moving grate boiler.* Fuel Processing Technology, 2010. **91**(11): p. 1491-1499.
3. Yin, C., L.A. Rosendahl, and S.K. Kær, *Grate-firing of biomass for heat and power production.* Progress in Energy and Combustion Science, 2008. **34**(6): p. 725-754.
4. Mondal, S.S., S.K. Som, and S.K. Dash, *Influences of inlet air pressure, air temperature and secondary air swirl on penetration histories of pulverized coal particles in a tubular combustor.* International Journal of Thermal Sciences, 2006. **45**(2): p. 190-202.
5. Sami, M., K. Annamalai, and M. Wooldridge, *Co-firing of coal and biomass fuel blends.* Progress in Energy and Combustion Science, 2001. **27**(2): p. 171-214.
6. Kalisz, S., M. Pronobis, and D. Baxter, *Co-firing of biomass waste-derived syngas in coal power boiler.* Energy, 2008. **33**(12): p. 1770-1778.
7. Hansen, P., et al., *Co-firing straw and coal in a 150-MW< sub> e</sub> utility boiler: in situ measurements.* Fuel Processing Technology, 1998. **54**(1): p. 207-225.
8. Demirbaş, A., *Sustainable cofiring of biomass with coal.* Energy conversion and management, 2003. **44**(9): p. 1465-1479.
9. Paces, N. and M. Kozek. *Modeling of a Grate-Firing Biomass Furnace for Real-Time Application*. in *International Symposium on Models and Modeling Methodologies in Science and Engineering: MMMse 2011*. 2011.
10. Gómez-Barea, A. and B. Leckner, *Modeling of biomass gasification in fluidized bed.* Progress in Energy and Combustion Science, 2010. **36**(4): p. 444-509.
11. Ryu, C., et al., *Effect of fuel properties on biomass combustion: Part I. Experiments—fuel type, equivalence ratio and particle size.* Fuel, 2006. **85**(7): p. 1039-1046.
12. Bauer, R., et al., *Modelling of grate combustion in a medium scale biomass furnace for control purposes.* Biomass and Bioenergy, 2010. **34**(4): p. 417-427.
13. Rusinowski, H., et al., *Methods of choosing the optimal parameters for solid fuel combustion in stoker-fired boilers.* Energy Conversion and Management, 2002. **43**(9–12): p. 1363-1375.
14. Wienese, A. *Boilers, boiler fuel and boiler efficiency*. in *Proc S Afr Sug Technol Ass*. 2001.
15. Wienese, A. *Boilers, boiler fuel and boiler efficiency*. 2001.
16. Saidur, R., J.U. Ahamed, and H.H. Masjuki, *Energy, exergy and economic analysis of industrial boilers.* Energy Policy, 2010. **38**(5): p. 2188-2197.
17. van den Broek, R., A. Faaij, and A. van Wijk, *Biomass combustion for power generation.* Biomass and Bioenergy, 1996. **11**(4): p. 271-281.
18. Babcock, G.H., AND Wilcox, S *Steam Its Generation and Use* Vol. 40th Edition. 1978.
19. Ayhan, D., *Combustion characteristics of different biomass fuels.* Progress in Energy and Combustion Science, 2004. **30**(2): p. 219-230.
20. Haykırı-Açma, H., *Combustion characteristics of different biomass materials.* Energy conversion and management, 2003. **44**(1): p. 155-162.

21. Zhang, L., C.C. Xu, and P. Champagne, *Overview of recent advances in thermo-chemical conversion of biomass.* Energy conversion and management, 2010. **51**(5): p. 969-982.
22. Aghamohammadi, N., N.M. Nik Sulaiman, and M.K. Aroua, *Combustion characteristics of biomass in SouthEast Asia.* Biomass and Bioenergy, 2011. **35**(9): p. 3884-3890.
23. Saidur, R., et al., *A review on biomass as a fuel for boilers.* Renewable and Sustainable Energy Reviews, 2011. **15**(5): p. 2262-2289.
24. Demirbaş, A., *Calculation of higher heating values of biomass fuels.* Fuel, 1997. **76**(5): p. 431-434.
25. Demirbaş, A., *Fuel characteristics of olive husk and walnut, hazelnut, sunflower, and almond shells.* Energy sources, 2002. **24**(3): p. 215-221.
26. Abbas, T., et al., *The influence of burner injection mode on pulverized coal and biomass co-fired flames.* Combustion and flame, 1994. **99**(3): p. 617-625.
27. Ebeling, J. and B. Jenkins, *Physical and chemical properties of biomass fuels.* Transactions of the ASAE (American Society of Agricultural Engineers), 1985. **28**(3): p. 898-902.
28. Shen, J., et al., *The prediction of elemental composition of biomass based on proximate analysis.* Energy conversion and management, 2010. **51**(5): p. 983-987.
29. Friedl, A., et al., *Prediction of heating values of biomass fuel from elemental composition.* Analytica Chimica Acta, 2005. **544**(1): p. 191-198.
30. Mohan, T., *An integrated approach for techno-economic and environmental analysis of energy from biomass and fossil fuels.* 2007.
31. Tillman, D.A., *Biomass cofiring: the technology, the experience, the combustion consequences.* Biomass and Bioenergy, 2000. **19**(6): p. 365-384.
32. Huang, C., et al., *Ultimate analysis and heating value prediction of straw by near infrared spectroscopy.* Waste management, 2009. **29**(6): p. 1793-1797.
33. Khan, A., et al., *Biomass combustion in fluidized bed boilers: Potential problems and remedies.* Fuel Processing Technology, 2009. **90**(1): p. 21-50.
34. Wilson, L., et al., *Thermal characterization of tropical biomass feedstocks.* Energy conversion and management, 2011. **52**(1): p. 191-198.
35. Husain, Z., Z. Zainal, and M. Abdullah, *Analysis of biomass-residue-based cogeneration system in palm oil mills.* Biomass and Bioenergy, 2003. **24**(2): p. 117-124.
36. Miles, T.R., et al., *Boiler deposits from firing biomass fuels.* Biomass and Bioenergy, 1996. **10**(2): p. 125-138.
37. Verma, V.K., et al., *Agro-pellets for domestic heating boilers: Standard laboratory and real life performance.* Applied Energy, 2012. **90**(1): p. 17-23.
38. Bazooyar, B., A. Ghorbani, and A. Shariati, *Combustion performance and emissions of petrodiesel and biodiesels based on various vegetable oils in a semi industrial boiler.* Fuel, 2011. **90**(10): p. 3078-3092.
39. Ghorbani, A., et al., *A comparative study of combustion performance and emission of biodiesel blends and diesel in an experimental boiler.* Applied Energy, 2011. **88**(12): p. 4725-4732.
40. Yang, Y.B., et al., *Effect of fuel properties on biomass combustion. Part II. Modelling approach—identification of the controlling factors.* Fuel, 2005. **84**(16): p. 2116-2130.
41. Wikipedia. *Tropical climate*. n.d 25th October 2012 [cited 2012 October 31]; Available from: http://en.wikipedia.org/wiki/Tropical_climate.
42. Klason, T., et al., *Investigation of radiative heat transfer in fixed bed biomass furnaces.* Fuel, 2008. **87**(10): p. 2141-2153.

43. Chau, J., et al., *Techno-economic analysis of wood biomass boilers for the greenhouse industry.* Applied Energy, 2009. **86**(3): p. 364-371.
44. Zhou, J., et al., *Improving Boiler Efficiency Modeling Based On Ambient Air Temperature*. 2002.
45. Harimi, M., et al., *Numerical study of heat loss from boiler using different ratios of fibre-to-shell from palm oil wastes.* Journal of Scientific and Industrial Research, 2008. **67**(6): p. 440-444.
46. Prasad, S.B., *Biomass-fired steam power cogeneration system: a theoretical study.* Energy conversion and management, 1995. **36**(1): p. 65-77.
47. Codes, A.P.T., *Steam Generating Units (ASME PTC 4.1).* ASME, New York, 1998.
48. Sandberg, J., et al., *Dynamic simulation of fouling in a circulating fluidized biomass-fired boiler.* Applied Energy, 2011. **88**(5): p. 1813-1824.
49. Purbolaksono, J., et al., *Failure case studies of SA213-T22 steel tubes of boiler through computer simulations.* Journal of Loss Prevention in the Process Industries, 2010. **23**(1): p. 98-105.
50. Belosevic, S., et al., *Three-dimensional modeling of utility boiler pulverized coal tangentially fired furnace.* International Journal of Heat and Mass Transfer, 2006. **49**(19-20): p. 3371-3378.
51. Bordbar, M.H. and T. Hyppänen, *Modeling of radiation heat transfer in a boiler furnace.* Advanced Studies in Theoretical Physics, 2007. **1**(12): p. 571-584.
52. Kær, S.K., *Numerical modelling of a straw-fired grate boiler.* Fuel, 2004. **83**(9): p. 1183-1190.
53. *Energy Performance Assesment of Boilers*, Bureau of Energy Efficiency India.

## APPENDIX A- FURNACE EXIT GAS TEMPERATURE (FEGT) FOR PALM WASTE FIRING BASED ON HEAT RELEASE RATE. (Field Data by Okutech Sdn. Bhd)

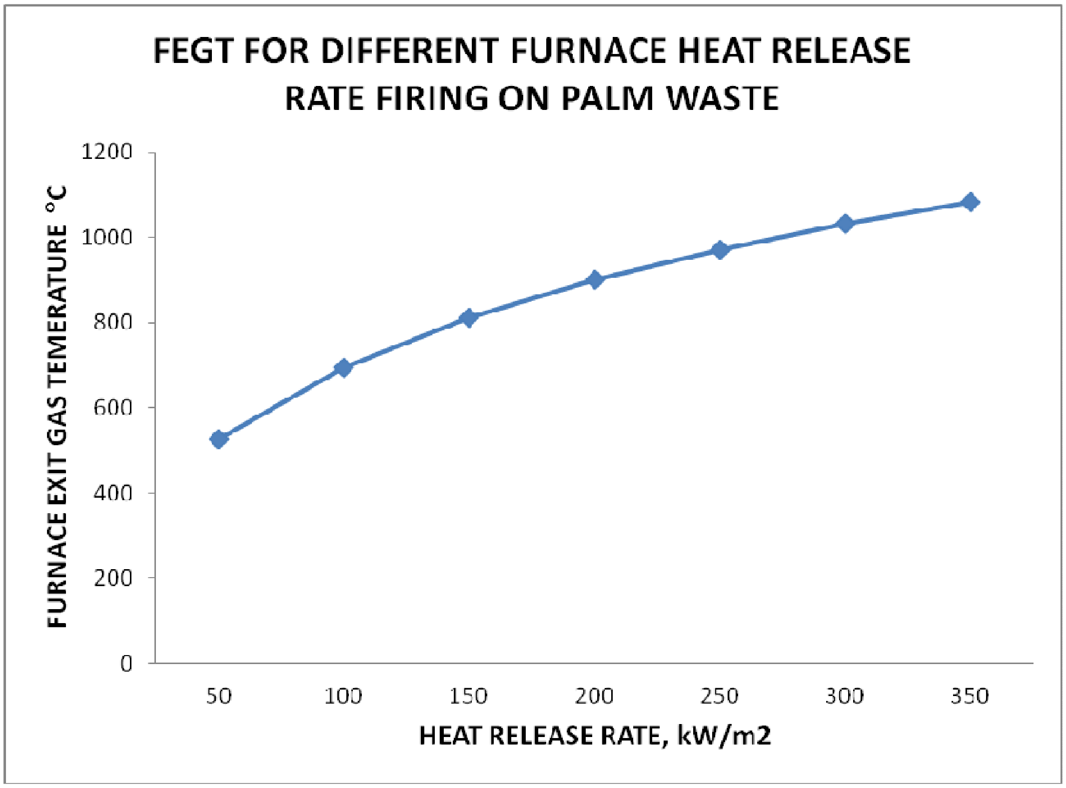



**Figure A.1 – Furnace exit gas temperature for palm waste firing by Okutech Sdn.Bhd**

# APPENDIX B- PSYCHOMETRIC CHART

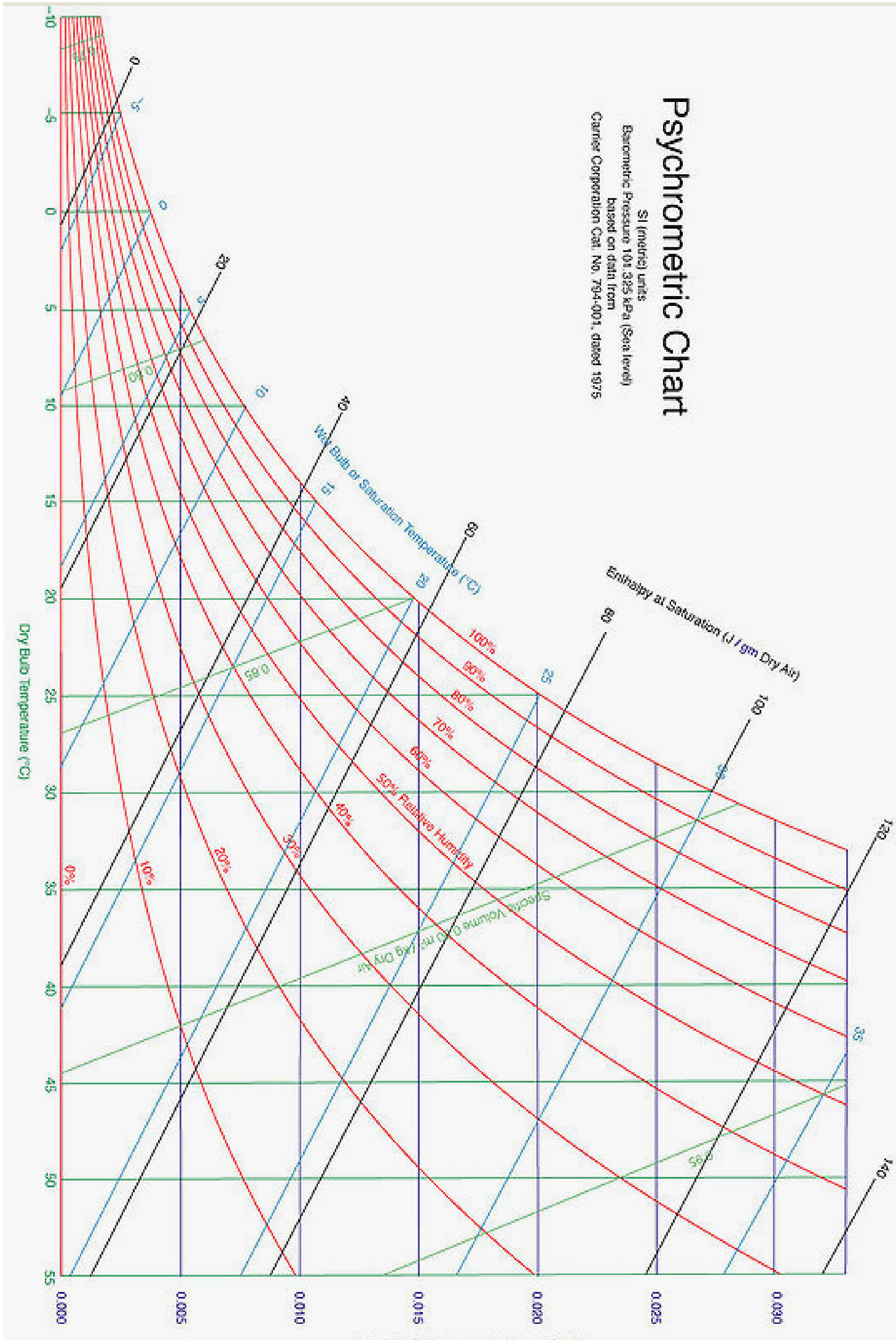



**Figure B.1 – Psychometric chart**

# APPENDIX C- <u>ABMA RADIATION HEAT LOSS CHART</u>

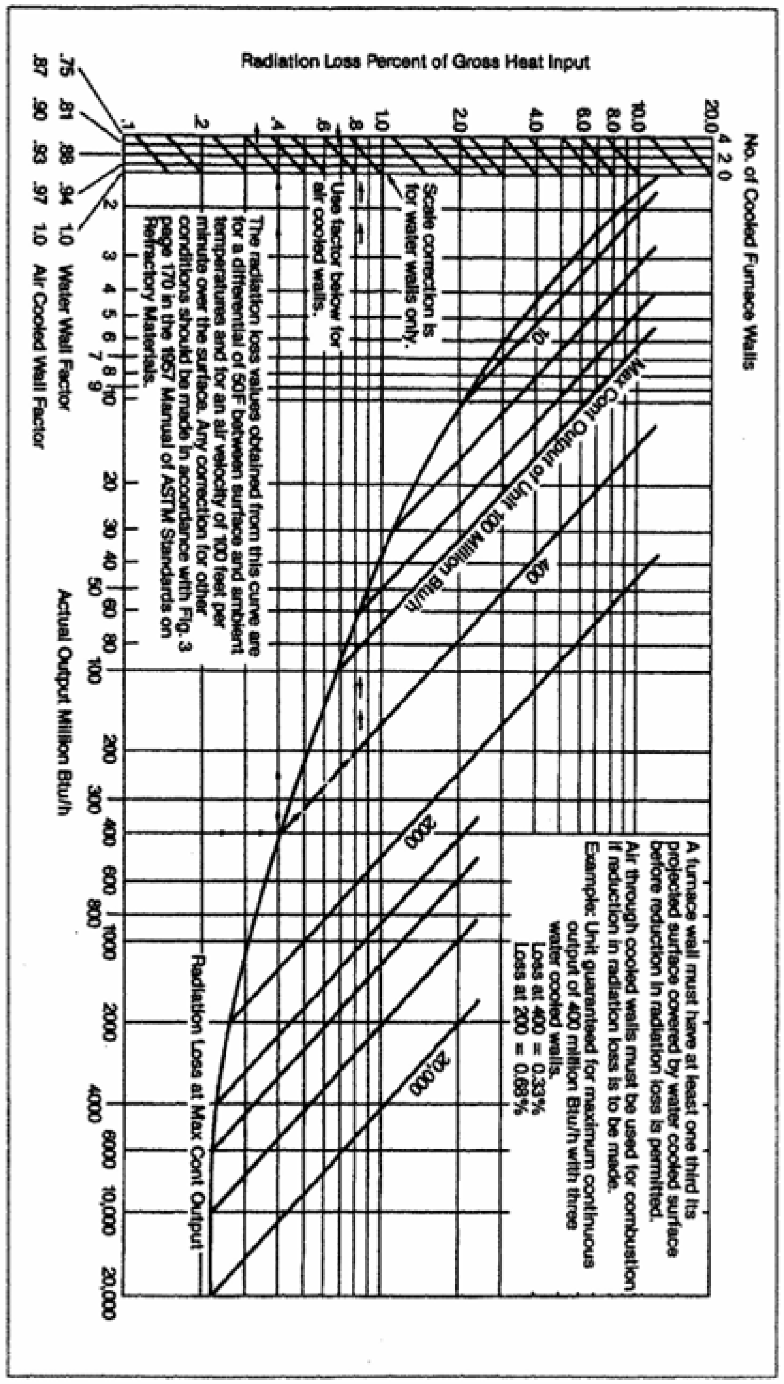



**Figure C.1 – ABMA Radiation heat loss chart**

# APPENDIX D – HEAT TRANSFER GRAPHS

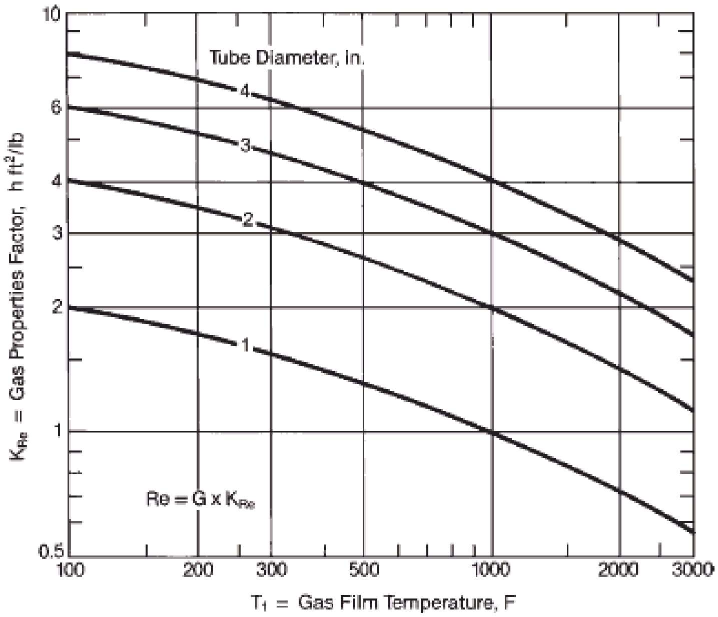



**Figure D.1 – Gas properties factor vs Gas film Temperature[18]**

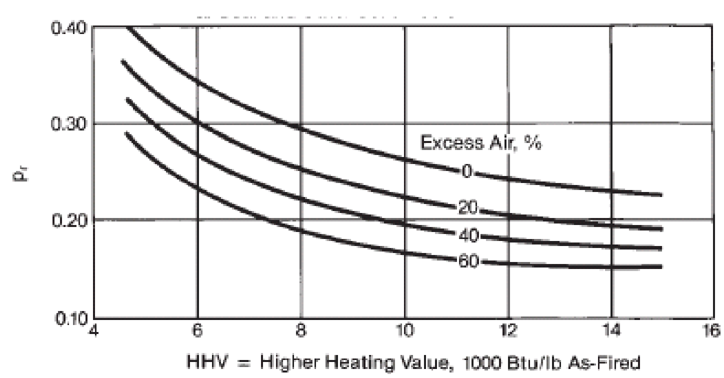



**Figure D.2 – Partial pressure vs Higher Heating Value[18]**

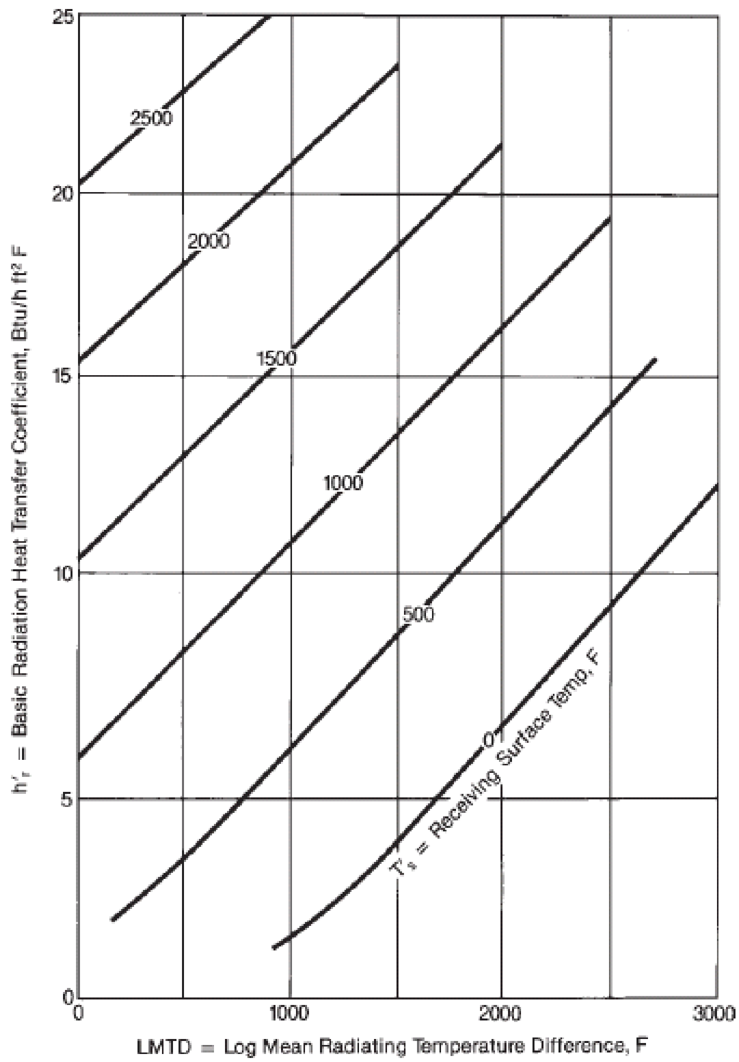



**Figure D.3 – Basic radiation heat transfer coefficient, hr'[18]**

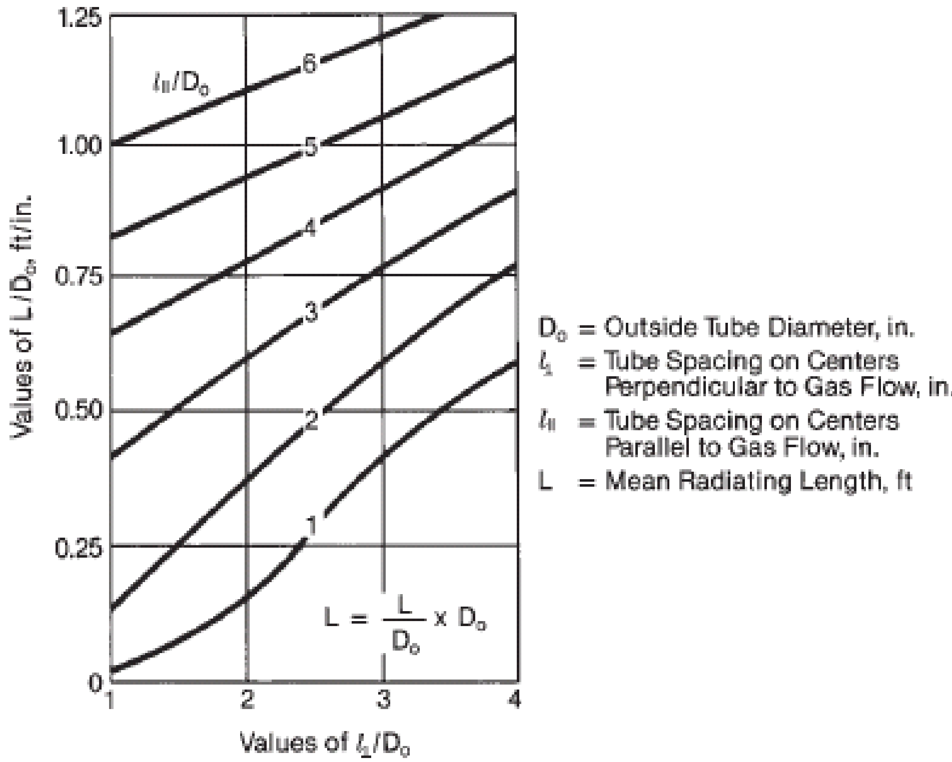



**Figure D.4 – Mean radiating length,L for different tube OD and pitch arrangement[18]**

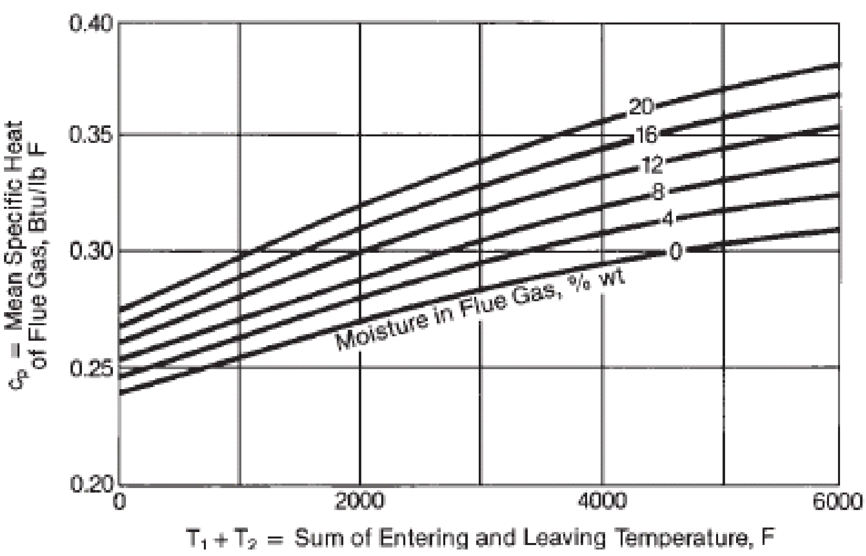



**Figure D.5 – Approximate mean specific heat, $C_p$, of flue gas[18]**

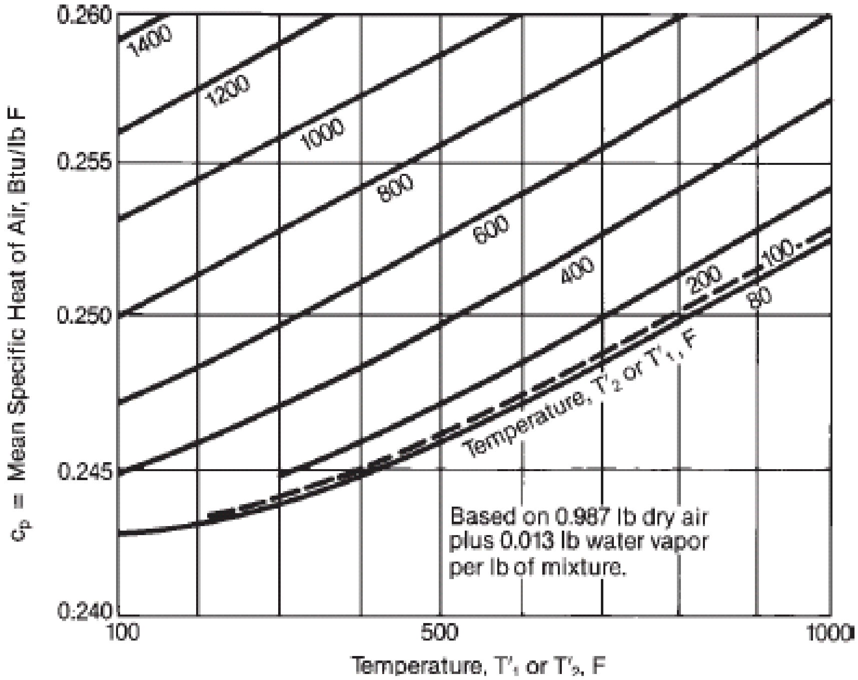



**Figure D.6 – Mean specific heat, $C_p$, of air at one atmosphere[18]**

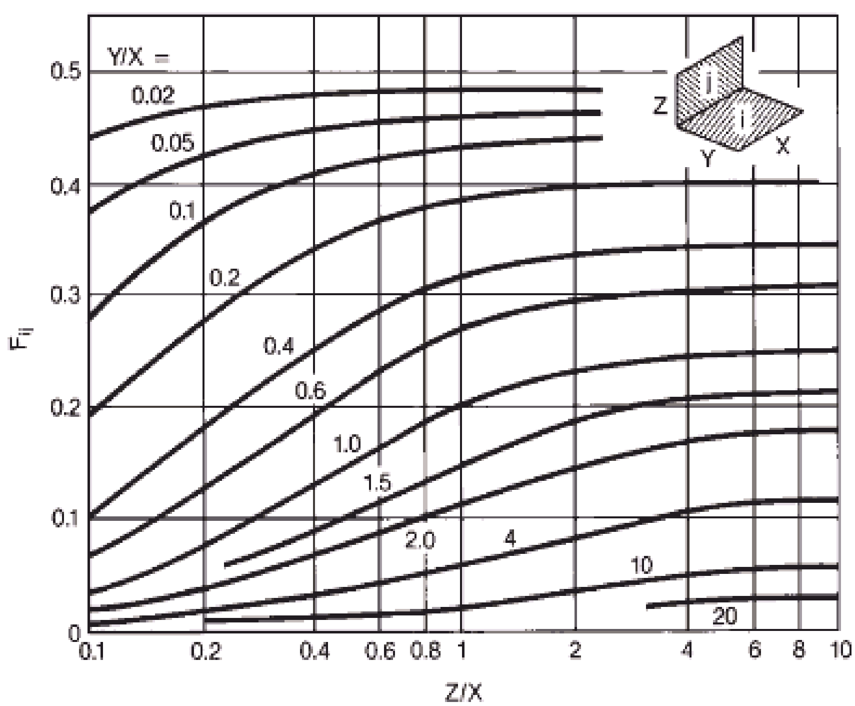



**Figure D.7 – Shape factor[18]**

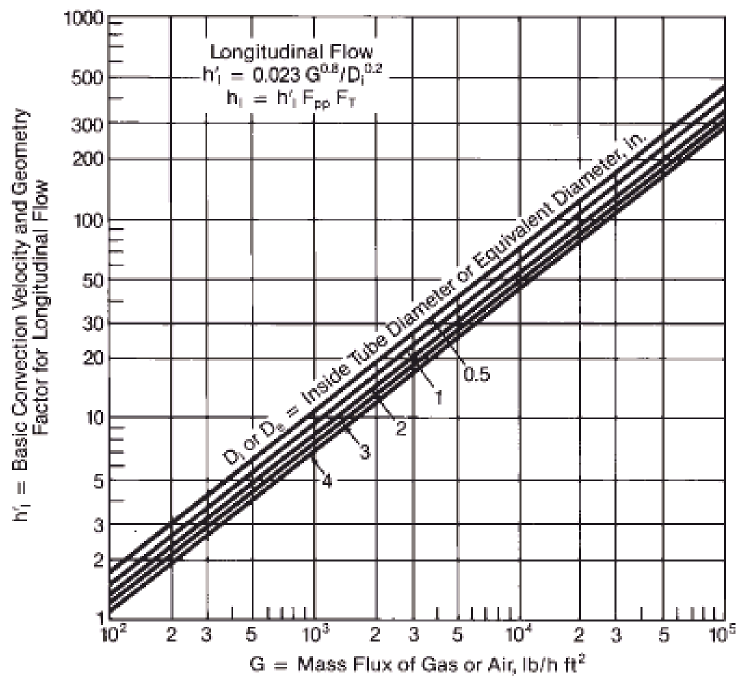



**Figure D.8 – Basic convection velocity and geometry factor for air, gas or steam; Turbulent flow inside the tubes or longitudinal flow over the tubes. [18]**

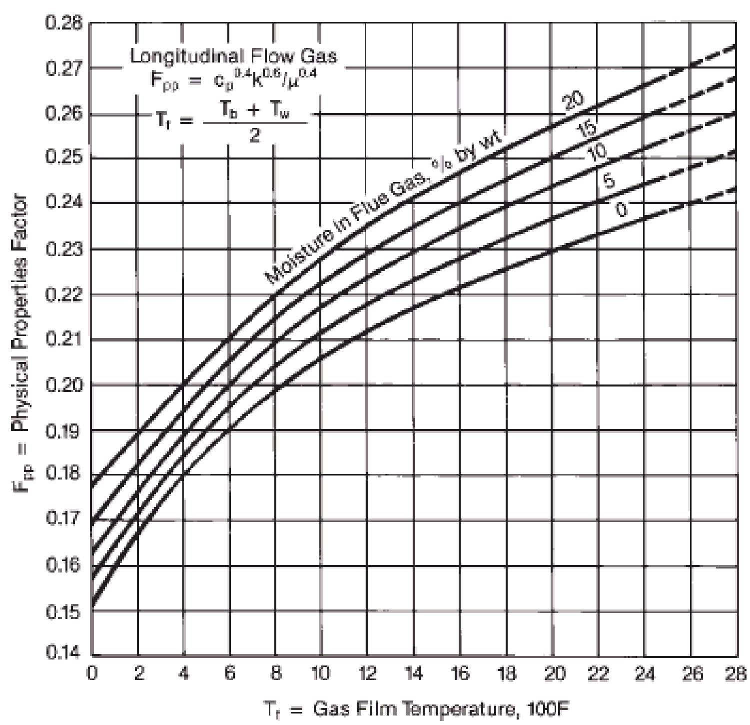



**Figure D.9 – Effect of Film Temperature, $T_f$ and moisture on the physical properties factor, $F_{pp}$ for gas: turbulent flow inside tubes and longitudinal flow over the tubes. [18]**

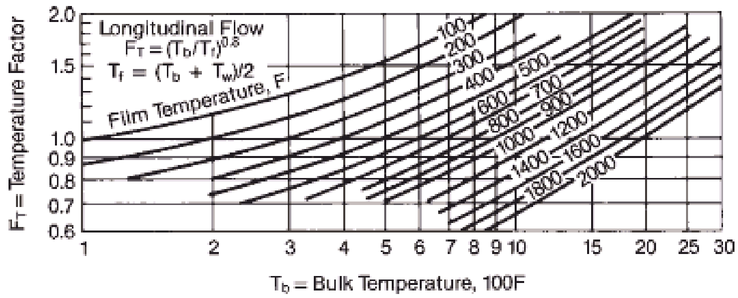



**Figure D.10 – Effect of Film Temperature, $T_f$ and moisture on the physical properties factor, $F_{pp}$ for gas: turbulent flow inside tubes and longitudinal flow over the tubes. [18]**

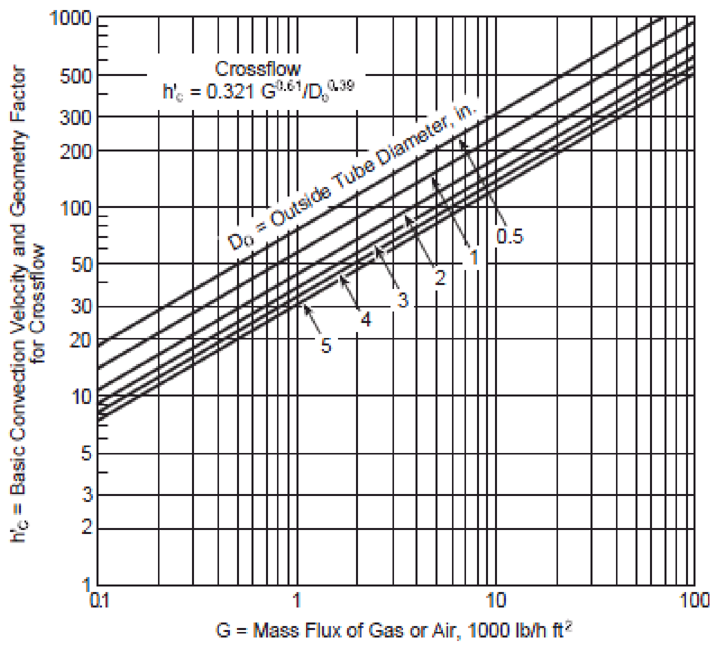



**Figure D.11 – Basic cross flow convection velocity and geometry factor h'$_c$ for gas and air[18]**

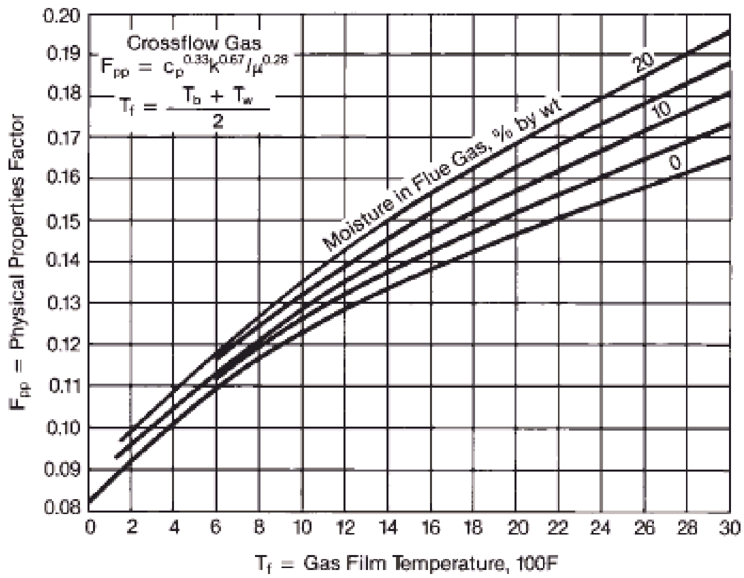



**Figure D.12 – Effect of Film Temperature, $T_f$ and moisture on the physical properties factor Fpp for gas in turbulent cross flow over tubes[18]**

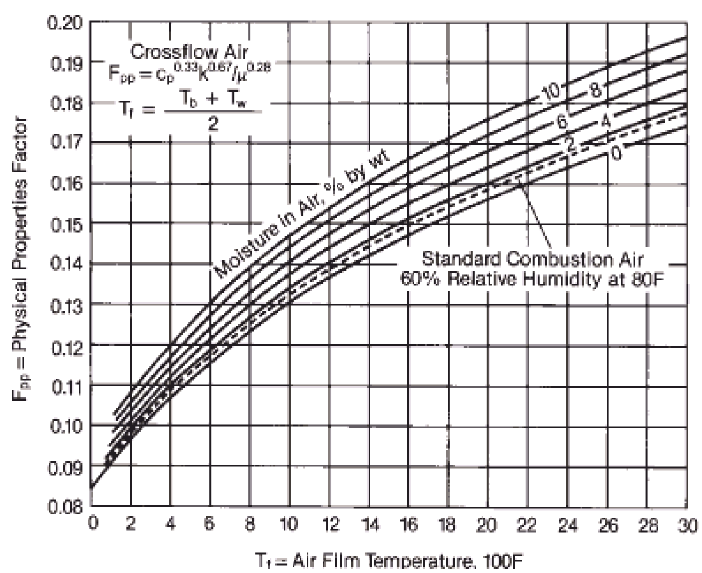



**Figure D.13 – Effect of Film Temperature, $T_f$ and moisture on the physical properties factor Fpp for air in cross flow over tubes. [18]**

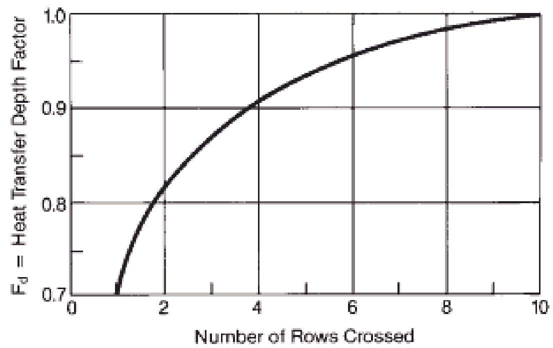



**Figure D.14 – Heat transfer depth factor for number of tube rows crossed in convection banks. (Fd = 1.0 if tube bank is immediately preceded by a bend, screen or damper[18]**

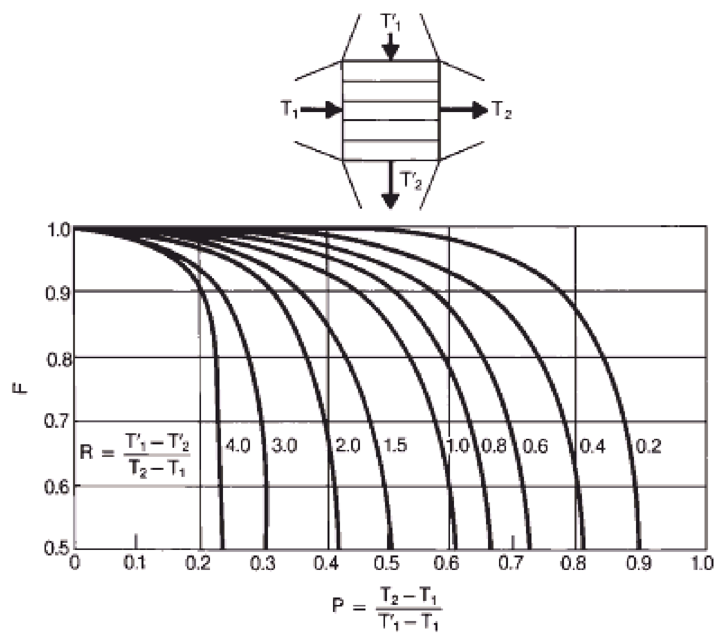



**Figure D.15 – Depth factor, $F_d$ , of cross flow arrangement[18]**

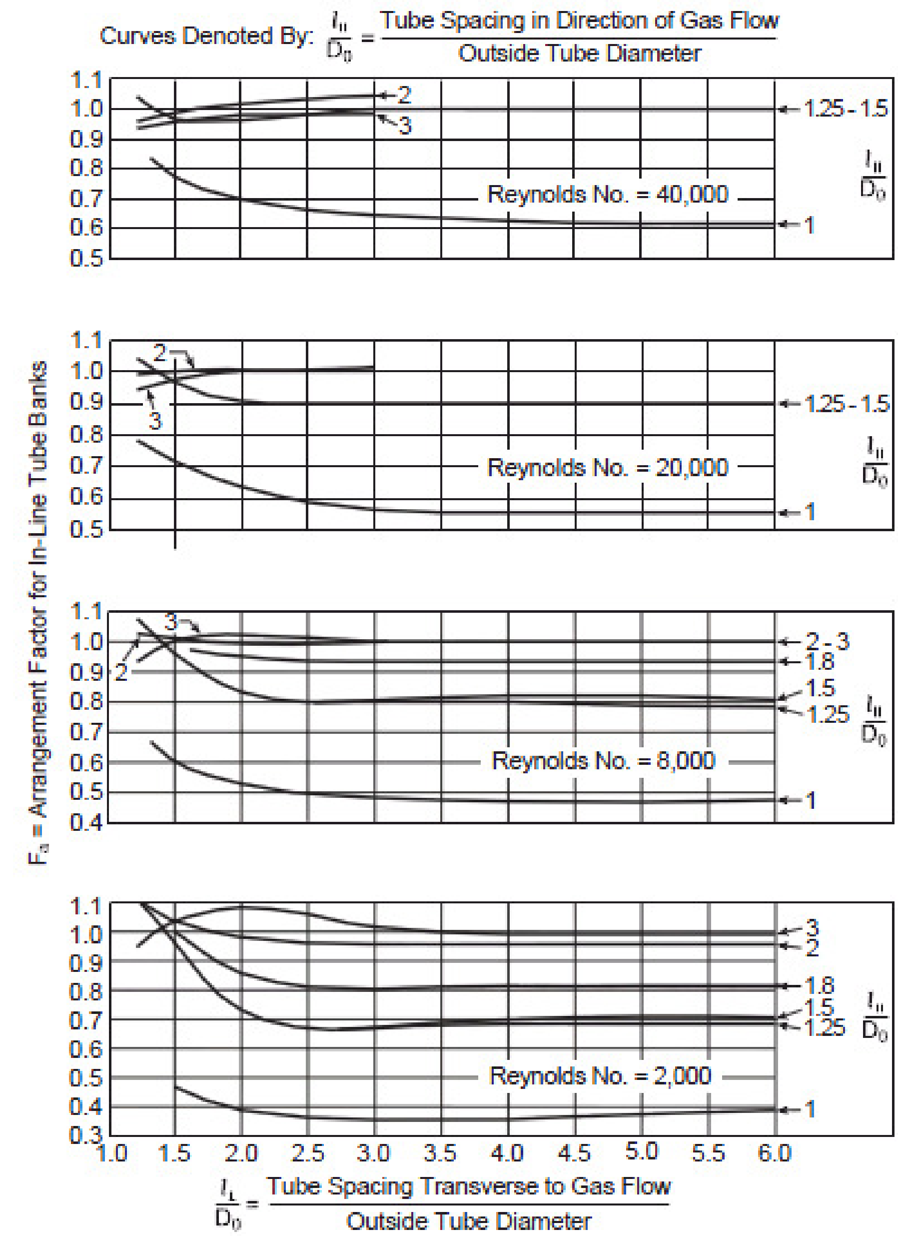



**Figure D.16 – Arrangement factor Fa as affected by Reynolds number for various in-line tube patterns, clean tube conditions for cross flow of air or natural gas combustion products.** [18]

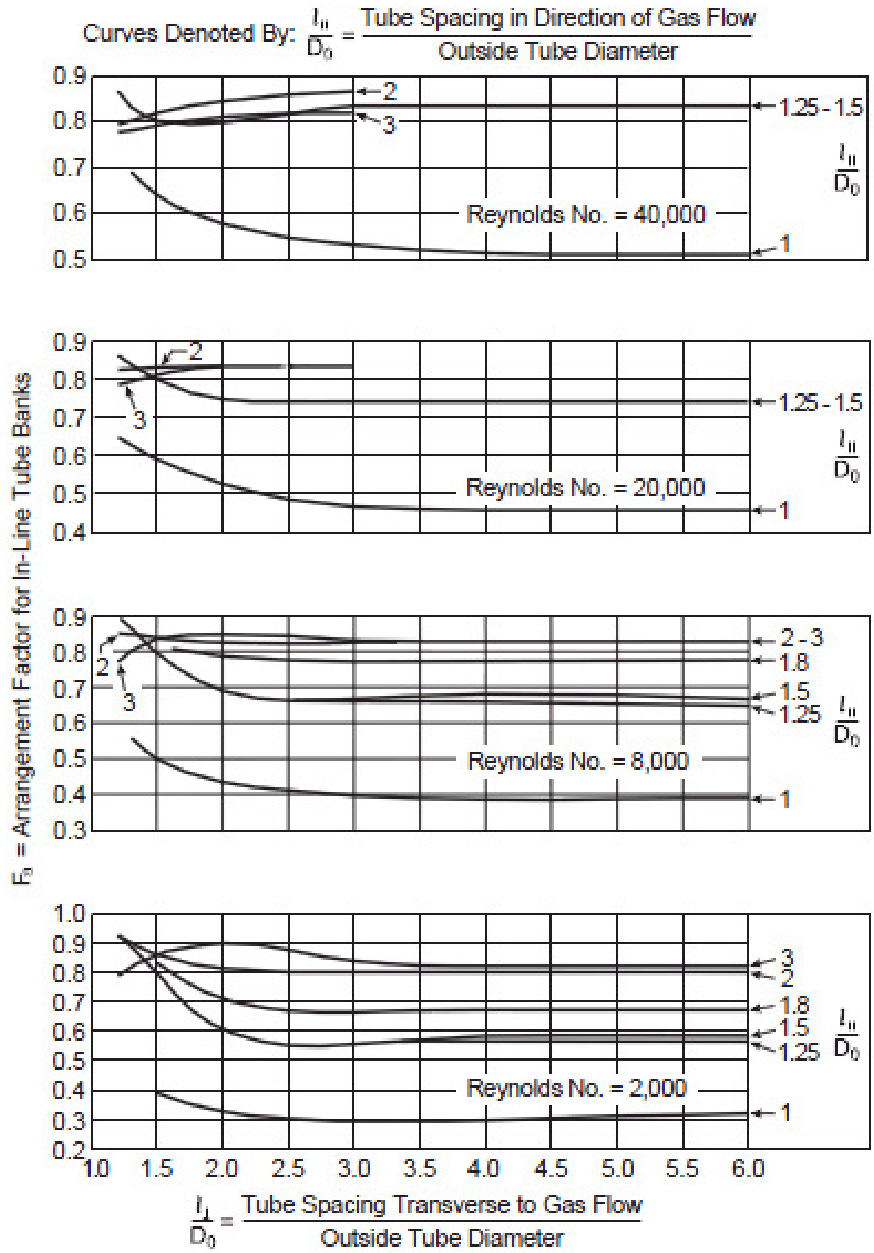



**Figure D.17 – Arrangement factor Fa as affected by Reynolds number for various in-line tube patterns, clean tube conditions for cross flow of ash-laden gases. [18]**