DEVELOPMENT OF RECUPERATORS FOR ROTARY FURNACES

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CERTIFICATION

I certify that this research work was carried out by **Oyelami, Adekunle Taofeek** of the Department of Mechanical Engineering, Federal University of Technology, Akure and has been approved to have met the requirements for the award of M.Eng. degree in Mechanical Engineering (Production Option) of the Federal University of Technology, Akure, Ondo State, Nigeria.

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07/03/06
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DEDICATION

This project work is dedicated to the Almighty God who has been very faithful to His promise of being my Protector, Guardian and Provider.

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ABSTRACT

The design of a radiation-recuperator incorporated in rotary furnace has been undertaken in this study. It was observed that an appreciable percentage of the heat input into the furnace is being lost to the surrounding, most especially through the waste gases produced from the fuel combustion. The coefficient of heat utilization is greatly dependent on the temperature of the waste gases leaving the furnace.

Therefore, the new design of recuperator, which is a heat exchanger, incorporated into the rotary furnace is done in such a way that it returns substantial part of the waste heat back into the furnace. A radiation type recuperator was designed to achieve this objective. It was established that the design results in 35% increase in the thermal efficiency of the furnace system and also leads to a 44% reduction in the fuel consumption.

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CHAPTER ONE

1.0 INTRODUCTION

1.1 FURNACES

Almost all manufactured articles go through a process of heating at some stage. The equipment used for heating can be broadly termed a 'furnace'. Because of the diversity of manufacturing processes and raw materials used, the types of furnace required are numerous. Kazantsev (1997) defines a furnace as a brick-lined chamber, capable of holding/conveying the material to be treated, to which heat is applied by one of various means so as to achieve the required final results. In fuel-fired furnaces, according to Cottrell (1975) and as the name implies, heat is supplied by combustion of fuel and is transmitted by a combination of heating processes. Rotary furnace is a typical example of a fuel-fired furnace. In general, fuel costs are a small fraction of the total manufacturing cost, and in the past, fuel economy has often been considered unimportant.

However, with increasing costs and decreasing availability of fuels, there is now a considerable financial and a strong conservational incentive to watch fuel economy aspects. It is in this regard, among other reasons, that this research work focuses on incorporating a recuperator into the design of a rotary furnace so as to reduce considerably the operational/running cost of the furnace in addition to increasing its thermal efficiency appreciably.

1.2 BACKGROUND OF STUDIES

Quite a large number of rotary furnaces are found to perform below the expected capacity, most especially in terms of thermal efficiency, mainly because of non-standardization of designs and more often than not, non-inclusion of a heat exchanger. Rotary furnaces designed without taking into consideration the designs of the burner, the blower (the air flow rate), the fuel supply system and the possibility of incorporating a heat exchanger, cannot perform with the expected thermal efficiency. The reason being that the resultant effect of the items mentioned above has a direct impact on the flame length of the furnace, which is one of the most important characteristics of the furnace, that need to be accurately determined. This research work thus optimizes the design of a recuperator-incorporated rotary furnace by giving adequate consideration to the design of the entire rotary furnace system, with more emphasis on the design of the recuperator which is the main unit responsible for an appreciable increase in the thermal efficiency of the furnace.

Rotary furnace is a traditional method of recycling of aluminium. These furnaces are one of the original methods for the production of aluminium from scraps (Krivadin and Markov 1980). Today, besides their usage in processing a wide variety of scraps materials, from coated strip to used beverage cans and furnace dross having different composition and content of aluminium, rotary furnaces now play a major role in the recycling of cast iron. In addition to this, it could also be adapted for the production of Austempered Ductile Iron (ADI) (EMDI 2005). The results of this research work also show that it could be adapted for the melting of mild steel.

The rotary furnace consists of a cylindrical, refractory lined, steel drum, supported by a structural steel frame, in between two conical frustums. The conical frustums are of two different diameters. The primary heat source of the furnace could either be a natural gas burner or a liquid – fuel burner (EMDI 2005) and (Krivadin and Markov 1980). A liquid fuel burner with diesel as the liquid fuel is used in this research work. The burner is usually positioned at the smaller-diameter end of the frustum, while the exhaust, in this case coupled with the heat exchanger, is positioned at the larger-diameter end.

All metallurgical furnaces by technological principle are classified as either melting or heating furnaces. Melting furnaces serve to make metals from ores and remelt metal for obtaining the desired properties. A characteristic feature of melting furnaces is that the materials processed in them change their state of aggregation. Heating furnaces on the other hand are employed to heat materials for roasting (limestone, magnesite, refractories, etc) or drying (foundry mould, ore, sand etc). They are also meant for increasing the plasticity of metals before plastic working. Many heating furnaces are used for heat treatment of metals, in which the structure of metals undergoes certain changes (Krivadin and Markov 1980) and (Kazantsev 1997).

1.3 OBJECTIVES

The specific objectives of the research are to:

(a) design and develop a radiation-type recuperator for rotary furnace,

(b) develop a geometric model for the recuperator-incorporated rotary furnace using solid modeler software,

(c) ascertain the reliability of the developed software using statistical method, and

(d) establish a significant improvement in the thermal efficiency of the recuperator-incorporated furnace over the existing non-recuperator incorporated ones.

1.4 Expected Contribution of the Research to Knowledge:

The research work will provide a quantitative framework for designing a recuperator specifically for a rotary furnace system and determining its quantitative effects on the furnace performance. The geometric model to be developed for the recuperator – incorporated rotary furnace will be in three dimensions thereby providing the user with the vision of the object very much like it would be seen in real life, and the software to be developed for the design process will widen the scope of computer applications for solving design problems for different furnace capacities.

1.5 RESEARCH METHODOLOGY

The recuperator was designed based on the principles of heat transfer processes. Geometric models were developed for all the principal components of the rotary furnace system, together with the designed recuperator, through the use of Solid Modeler Software. C++ programming language was then used to develop the software used in the design of the recuperator-incorporated rotary furnace, so as to obviate the rigour associated with design for different capacities of rotary furnace.

The overall efficiency of the furnace system was shown to have increased considerably with the incorporation of a recuperator through the following procedure: Necessary mathematical and geometric models were developed; the developed models were used to develop a program using C++ programming language. The reliability of the program developed was tested by obtaining a set of actual data for a known capacity rotary furnace (without a recuperator) from the industry; generating a corresponding set of data from the developed program; ascertaining the reliability of the developed program by establishing that there is no significant difference between the two set of data using the necessary statistical methods.

It was established that the design results in 35% increase in the thermal efficiency of the furnace system and also leads to a 44% reduction in the fuel consumption. The developed software was later used to design and develop a radiation-type recuperator for a 500kg-capacity rotary furnace (Appendix C).

CHAPTER TWO

2.0 LITERATURE REVIEW

2.1 FURNACE CLASSIFICATION

2.1.1. TECHNOLOGICAL AND STRUCTURAL CLASSIFICATION

By technological principle, all metallurgical furnaces are classified by (Krivadin and Markov 1980) into melting and heating furnaces. Melting furnaces serve to make metals from ores and remelt metals for obtaining the desired properties. A characteristic feature of melting furnaces is that the materials processed in them change their state of aggregation.

Heating furnaces are employed to heat materials for roasting (limestone, magnesite, refractories, etc) or drying (foundry moulds, ore, sand, etc), as also for increasing the plasticity of metals before plastic working. Many heating furnaces are used for heat treatment of metals, in which the structure of metal undergoes certain changes. All heating furnaces are characterized by the common feature that the materials processed in them remain in the same state of aggegation.

Furnaces in each of these two groups are classified according to the particular operations they are intended to perform. For instance, melting furnaces are used for melting iron, steel, copper, etc. Heating furnaces can be used for roasting of ores or refractories, heating of metal before rolling or forging or for heat treatment of metals. These groups of furnaces, intended for specific methods of metal transporting in the furnace, and type of product before rolling are divided into the following groups:

(a) soaking pits, continuous furnaces, batch-type furnaces, etc;

(b) in-and-out, pusher-type, rotary-hearth furnaces;

(c) furnaces for heating ingots, blums, tubes, bar strips, etc.(Dryden 1982), (Krivadin and Markov 1980) and(Kazantsev 1997).

Cottrell (1975) classifies fuel-fired furnaces by the type of fuel used. For instance, steel-making open hearth furnaces may be gas-fired or fuel oil-fired. Furnaces may be regenerative or recuperative according to the method the heat of waste gases is utilized. Electric furnaces are classified by the method of conversion of electric energy into heat: arc furnaces, resistance furnaces and induction furnaces. Modern furnaces are complex thermal plants usually composed of the furnace proper and auxiliary equipment. The furnace proper commonly has the reaction chamber (furnace space) and means for producing thermal energy, such as burners in flame furnaces or electrodes and resistors in electric furnaces. (Khanna 2005) The auxiliary equipment comprises means for utilization of the heat of waste gases, ventilators and exhausters, stacks, various valves, gates, etc. The technological process for which a furnace is intended is performed in the reaction chamber. All other elements of a furnace serve to ensure the most effective conditions for the main technological process (Kern 1984).

2.1.2 FURNACE CLASSIFICATION BY THE PRINCIPLE OF HEAT GENERATION

According to (Krivadin and Markov 1980) all furnaces rely on conversion of some or other kind of energy into heat. The main primary sources of energy are

(a) Chemical energy of fuel (in fuel- fired furnaces)

(b) Chemical energy of molten metal and

(c) Electric energy.

In fuel fired furnaces, the chemical energy of fuel is converted into heat through combustion. In ferrous metallurgy, this group of furnaces includes flame and shaft furnaces. In flame (reverberatory) furnaces, the material being processed is usually placed on the furnace hearth and occupies only a small portion of the reaction chamber volume. The main portion of the reaction chamber is filled with flames and incandescent combustion products, which transfer their heat to the material (mainly by radiation). These furnaces can be fired with gaseous, liquid or sometimes with dusty solid fuel.

In some types of furnace, the evolution of heat may be due to both chemical energy of fuel and that of molten metal. Examples of such are open–hearth furnace, which occupy an intermediate place between fuel–fired furnaces and converters (furnace in which the chemical energy of metal is converted into heat through combustion of the impurities present in them). In these furnaces, fuel is burned above the surface of metals and the flame and incandescent gases also pass over the surface of metal, that is the processes occur which are typical of flame furnaces (Kazantsev 1997). On the other hand, the impurities present in molten metal are burned out and liberate heat, sometime very much heat, so that in some period of a melt, the heat generation due to the chemical energy of molten metal may be substantial.

Electric energy can be converted to heat by:

(a) Passage of electric current through a gas

(b) The action of electric current onto a magnetic field, with eddy current being formed in the metal;

(c) Remagnetization and polarization of dielectrics;

(d) Passage of electric current through solid (or sometimes liquid) eletroconductive bodies; and

(e) Owing to kinetic energy of electrons.

These principles of conversion of electric energy to heat determine the types of existing electric furnaces, namely:

(a) Electric – arc and plasma furnaces;

(b) Induction furnaces

(c) Dielectric heating plants

(d) Resistance furnace and

(e) Electron- beam furnaces. (Krivadin and Markov 1980)

2.1.3. CLASSIFICATION OF FURNACES BY OPERATING MODE

FUNDAMENTALS OF THE MODERN THEORY OF FURNACES

A deserving contribution to the theory of furnaces has been made by Russian and Soviet scientists. The first attempt to formulate the main principles of the theory of furnaces was done by Grum-Grzhimailo (1911), a prominent Russian metallurgist. He developed what was called the hydraulic theory of furnaces which explained well the processes in furnaces of that time characterized by languid free motion of gases.

The progress in furnace construction in later years followed the path of development of furnaces with intensive forced motion of gases. As forced-motion furnaces gained in significance, the concepts of the hydraulic theory of furnaces became more and more inapplicable until the theory has fully lost its meaning. In the present theory, there are distinguished two groups of furnaces differing in the nature of the processes occurring in them. In one group, the evolution of heat takes place in the processed material proper. With these furnaces, the processes of heat generation are of prime importance (Khanna 2005).

The conditions to be created in a furnace, according to Oyelami et al (2003), should be such that the heat flow to the surface of material is at its maximum and that it can be absorbed completely. This can be ensured by the following measures:

- the temperature difference between the heating medium and heated surface should be as high as feasible under the given process conditions;

- the furnace should be supplied with heat intensively with the maximum heat utilization within the furnace space;
- gases in the furnace space should be made to circulate vigorously, mainly under the action of air and fuel jets from burners;
- fuel should be burned completely and combustion occur within the furnace space where possible;
- the design of a furnace and the pressure conditions in it should be such as to minimize the contact of furnace atmosphere with the surroundings.

Furnaces should be designed so as to ensure the conditions enumerated above and provide the optimum conditions for heat transfer.

2.2 CLASSIFICATION AND GENERAL CHARACTERISTICS OF MODES OF FURNACE OPERATION

In most furnaces, there are two main stages of heat transfer: from a heat carrier (flame, electric arc, etc) to the surface of material and from that surface into the depth of material. The former case is what is called the external problem and the latter, the internal problem. As regards the external problem, heat transfer is mainly through thermal radiation and convection. With the internal problem, heat transfer occurs predominantly by conduction. But with heated liquids, convective heat transfer is also possible (Krivandin and Markov 1980)

2.2.1 Radiation Mode

The radiation mode is characterized by the prevailing role of thermal radiation. If the temperature and emissivity of a flame or of the volume of incandescent gases are the same across the depth (for instance, along the y-axis), the heat transfer to the metal and roof (lining) of the furnace is uniform. With a non-uniform temperature field and variable emissivity across the depth of flame ur gas volume, heat transfer in various directions will be non-uniform. Thus, the possible modes can be determined in the general case as follows:

- Uniformly distributed radiant heat transfer :

$Q_l^m = Q_l^l$ -- (2.1)

where Q_l^m and Q_l^l are respectively heat flows from the flame to metal and lining;

- Oriented direct radiant heat transfer

$Q_l^m > Q_l^l$ -- (2.2)

- Oriented indirect radiant heat transfer

$Q_l^m < Q_l^l$ -- (2.3)

In all these cases of radiant heat transfer, the role of lining as of an intermediate link of heat transfer from flames will be different.

2.2.1.1 Oriented Direct Radiant Heat Transfer

Akinsipe (2004) described this mode of heat transfer as being employed most effectively in furnaces for heating large-mass articles. In that case the rated of heating is determined by the conditions of internal heat transfer, so that the relatively low intensity of external heat transfer is not essential. This heat transfer is ensured by creating a temperature gradient across the flame, the highest heat flow is emitted by the flame later having the highest temperature. When propagating towards the metal and lining, the heat flow undergoes a change owing to the absorptive action of other flame layers. If the absorptivity of all flame lagers is the same, the absorption will be lower at a less thick gas volume between the flame layer considered and the heat-absorbing surface of metal lining.

2.2.1.2 Oriented Indirect Radiant Heat Transfer

This mode of heat transfer is found in the cases where the heat of fuel combustion is transferred to the heated material not directly from the flame, but indirectly, that is via an intermediary part is most often played by the lining of furnace roof. A fuel-air mixture forms incandescent gases on combustion (for instance in injection burners); these gases have a rather high temperature and relatively low emissivity corresponding to the selective radiation of CO_2 and H_2O. The metal being heated has a continuous absorption spectrum. The heat flow falling on it should therefore have a continuous emission spectrum where possible. In the connection, it is rational to intensify heat transfer to the metal by directing the incandescent gases along the surface of furnace roof so that the region of maximum temperature be as close as possible to the surface of refractory lining. This can increase the temperature of the lining and transform the selective radiation of gases into continuous radiation of the lining (Akinsipe 2004) and (Krivadin and Markov 1980)

2.2.2 Convective Mode

At temperatures below 550-600^{0}C, heat transfer by convection is predominant. Convective mode is typical of low-temperature heat-treatment and drying furnaces. The temperature of the working space of these furnaces should be lower than the temperature of fuel combustion, because of which fuel is burned in a separate volume to provide developed motion of gases in order to heat the material uniformly. Convective mode is also employed in heating baths in which a hot liquid is the heat carrier.

2.2.3 Layer-wise Mode of Furnace Operation

Layer-wise mode is used in the processing of lumpy materials (mostly in shaft furnaces). The material usually fills the whole volume of a vertical shaft furnace and incandescent gases flow between its lumps, or particles of material are suspended in a gaseous heat carrier. The layer-wise mode of furnace operation has the typical feature that all three kinds of heat transfer (radiation, convection and conduction) are interlinked so closely that practically cannot be separated from one another (Kazantsev 1997). (Akinsipe 2004) and (Krivadin and Markov 1980).

2.3 FURNACE MATERIALS

The principal materials for building industrial furnaces are various refractories, building brick, concrete and metals. Of all these, the most important one are the refractories. The reason being that the lifespan of the furnace can actually be said to be the lifespan of the refractory itself.

2.3.1 Refractories

Refractories are materials, which can withstand high temperatures (above 1580 ^{0}C) and the physical and chemical action of molten metal, slag and gases in furnaces. The progress in refractory manufacture is closely connected with metallurgy, which consumes up to 60 per cent of all refractories produced (Cottrell1975). With the progress in metallurgy more and more heavy requirements were being placed on refractories; new types of refractory have been developed, such as high-alumina, periclase-spinelide and some oxide refractories.

2.3.1.1 Classification of Refractories

(Krivadin and Markov 1980) classified refractories by their properties and characteristics.

(i) By refractoriness, they are classified in to the following groups:

(a) of common refractoriness (up to 1580-1770C);

(b) of high refractoriness (1770-2000C);

(c) of the highest refractoriness (above 2000C).

(ii) By their chemical and mineral composition, refractories are classed as:

(a) siliceous, in which the refractory base is SiO_2 (silica, quartzite);

(b) alumina-siliceous, where refractory components are Al_2O_3 and SiO_2 (fireclay, semi-acid and high-alumina refractories);

(c) magnesian, i.e. based on MgO oxide (magnesite, dolomite, forsterite, talc, and spinlides);

(d) chromic, with the refractory base composed of Cr_2O_3 and MgO oxide (chromite, chrome-magnesite and magnesite and magnesite-chromite);

(e) carbon refractories, with the base carbon (C); these include carbon and graphite refractories.

(f) zircon refractories,based on ZrO_2 oxide (zirconia and zircon refractories);

(g) carbide refractories which contain refractory compounds of the type MeC

(h) oxide refractories composed mainly of pure oxides MgO, Al_2O_3, BeO, etc.

(iii) By the type of refractory oxide, all refractories are divided into

(a) acid (SiO)

(b) inert or neutral (Al_2O_3); and

(c) basic (MgO, CaO).

(iv) By the method of manufacture, all refractories are divided into natural and artificial. The latter may be pressed, fused or rammed. They also can be classed as roasted and unroasted according to the method of heat treatment.

(v) By their shape, all refractory materials are classified into common and specially shaped of diverse types. (Khanna 2005), (Robert and Don 1998) and (Krivadin and Markov 1980)

2.3.1.2 Physical Properties of Refractories.

(i) Porosity: The life of refractories is largely dependent on their porosity which may vary within considerable limits: from 1 per cent in melted refractories to 80 per cent in insulative refractory materials (Kazantsev 1997). Pores in refractories may be open, i.e. communicating with the mosphere; through, i.e. passing through the whole thickness of an article; or closed.

ccordingly, three kinds of porosity are distinguished:

a) *General porosity*- determined as the ratio of the volume of all pores to the total volume of an article;

b) *Apparent porosity*- the ratio of the volume of open pores to the total volume; and

c) *Closed porosity*- the ratio of the volume of closed pores to the total volume of an article. At a higher apparent porosity, a refractory material is less resistant to the eroding action of slag and metal which separate through pores inside the material. Porosity is determined by a standard test.

(ii) **Gas Permeability**. If there is a pressure gradient between the furnace space and surroundings, gases can penetrate through pores from one side to the other, i.e. the material is permeable to gases. There is no definite relationship between the general porosity and gas permeability, since the latter is only due to through pores. Gas permeability decreases with increasing viscosity of the gas.

(iii) **Heat Conductivity**. Refractories should have a low heat conductivity, except for rare cases when a high conductivity is desirable in order to transfer heat through a refractory material (such as in crucibles or recuperators). Heat conductivity depends on the nature and porosity of a material and on temperature. Crystalline substances have a higher conductivity than amorphous ones. Heat conductivity of refractories increases with temperature, exceptions being magnesite and forsterite. Heat conductivity diminishes with increasing porosity.

(iv) **Electric Conductivity**. Most refractories behave as dielectrics at low temperatures. At higher temperatures, when a liquid phase appearsin the material, their electric conductivity may rise. For instance, fireclay and silica refractories become electrically conductive at temperatures above 1200C. The electric resistivity of refractories depends on temperature by the formula $\log R = A/T + B$

Where A and B are constants for a given material; T is the absolute temperature, K.

Electric resistivity of refractories can be different depending on their composition. It diminishes as the content of iron and titanium oxides increases. This should be taken into consideration for refractories to be used in electric steelmaking furnaces.

(v) **Heat Capacity**. Heat capacity of refractories is of importance when selecting materials for regenerators and periodically operating furnaces. In general, heat capacity of refractories may vary in the range from 0.4 to 1.7 kJ/(kg ^{o}C). Heat capacity of refractories increases with temperature. It is measured by calorimeter method.

2.3.1.3 Service Properties of Refractories

(i) **Refractoriness:** Refractoriness is the ability of a material to retain its mechanical strength at high temperatures with no load applied. It is one of the principal properties which determine the possibility of using a refractory material under specified temperature conditions. It depends on chemical composition and content of impurities in a refractory material. Refractoriness is measured by a standard composition and content triangular pyramids – pyroscopes are cut from the material to be tested and placed into a kryptol kiln together with reference to pyramids of a specified refractoriness. The temperature in the furnace can be 1000, 1000-1500^{o}C or >1500 ^{o}C and can be raised at these temperatures respectively at a rate of 25, 10 and 5 degrees C per hour. When a refractory material is being heated, a liquid phase appears in it and grows in volume with increasing temperature and holding time until a pyroscope made up of the material begins to deform under its own weight.

Most often this occurs owing to the reactions between the refractories and slags and dust or because of insufficient thermal stability and less frequently, owing to a low mechanical strength. Silica has a refractoriness of 1710 to 1750^{o}C but begins to break down in checker work of open-hearth furnaces already at 1200 to 1300°C. For that reason, refractoriness cannot alone be indicative of a suitability of a refractory material for some or other furnace elements.

(ii) **Thermal Stability (Spalling Resistance):** Thermal stability is the ability of refractory to withstand sharp temperature changes without fracture (spalling). Thermal stability is characterized by the number of thermal cycles, i.e. heating and sharp quenching in water or air. With water quenching, a material is heated up to 850°C and cooled in running water (one cycle). The test is finished after the specimen has lost more than 20 per cent of its mass (owing to spalling under thermal stresses). Thermal stability is measured by a standard test (Khanna 2005), (Robert and Don1998), (Krivadin and Markov 1980) and (Cottrell1975)

2.4 THERMAL CONDITIONS IN A ROTARY FURNACE

The operation of a rotary furnace is decided to a large extent by the quantity of heat supplied to it. The quantity of heat supplied to the furnace at a given time instant is called thermal load. The highest quantity of heat that can be normally absorbed by furnace (without unburnt fuel in the furnace space) is called thermal power (Oyelami et al 2004). The thermal conditions of a furnace are essentially the time variation of thermal load. Thermal conditions are closely related to the temperature conditions of a furnace. In periodic furnaces which operate at a time –variable temperature, the thermal load varies in time, whereas continuous furnace operate at a constant thermal load.

2.5 SIGNIFICANCE OF WASTE HEAT RECYCLING

From the preceding section, it has been shown that two major sources of input heat into a rotary furnace are the chemical heat of fuel combustion and the physical heat of preheated air. A reasonable percentage of the heat liberated during the combustion of a liquid fuel is lost to the waste gases. A unit of heat recovered from waste gases and returned into the furnace with air (unit of physical heat) is therefore much more valuable than a unit of heat produced in the furnace through fuel combustion (unit of chemical heat), since the heat of preheated air or gas involves no heat losses with waste gases. The value of a unit of physical heat is greater at a lower coefficient of fuel utilization and higher temperature of waste gases (Oyelami et al 2003), (Oyelami et al 2004) and (Kazantsev 1997).

For normal operation, the working space of a furnace must be supplied every hour with a specified quantity of heat, which includes the physical heat of preheated air (Q_{ph}) and the chemical heat of fuel (Q_{ch}), i.e

$$Q_{\Sigma} = Q_{ch} + Q_{ph} \quad \text{-- (2.4)}$$

As can be inferred from equation (2.4) above, with Q_{Σ} taken as being constant, Q_{ch} can be diminished by increasing Q_{ph}. In other words, utilization of the heat of waste gases can save fuel. The quantity of saved fuel is dependent on the degree of heat utilization (Oyelami et al 2004).

$$R = I_a / I_{wg} \quad \text{-- (2.5)}$$

Where I_a is the enthalpy of preheated air in KW or KJ / period, and I_{wg} is the enthalpy of waste gases leaving the furnace in KW or KJ / period.

The degree of heat utilization, if expressed as percentage, can also be called the efficiency of a recuperator (or regenerator)

$$\eta_r = I_a / I_{wg} \times 100\% \quad \text{-------------------------------------- ----- (2.6)}$$

with η_r known, the economy in fuel can be found by the expression

$$\eta = R\frac{i_a/i^1_{wg}}{1-i_a/i^1_{wg}(1-R)}\times 100\% \quad \text{-- (2.7)}$$

where $i_{wg}{}^1$ is the enthalpy of waste gases at the temperature of burning, KJ/m^3; and i_{wg} is the enthalpy of the waste gases leaving the furnace, KJ/m^3. (Edward, 2000)

The implication of this is that utilization of the heat of waste gases can lower fuel consumption, and therefore, give an appreciable economic effect. It is one of the principal ways for reducing the cost of metal heating in industrial furnaces.

Apart from saving fuel, preheating of air increases the calometric temperature of combustion. This also forms one of the principal reasons for recuperation of the waste heat. At a constant value of the lower calorific value of the liquid fuel (i.e at Q_w^l = constant), an increase of Q_{ph} results in an increased temperature of combustion (Oyelami et al 2004).

2.6 MEANS OF RECYCLING THE WASTE HEAT

The heat of waste gases can be effectively recycled back into the furnace through the use of heat exchangers of regenerative or recuperative type. A heat exchanger is a device in which heat is transferred between a warmer and a colder substance, usually fluids. The regenerative heat exchangers operate under unstationary conditions while the recuperative type operate under stationary conditions (Oyelami and Ogunkoya 2002).

2.6.1. REGENERATIVE HEAT EXCHANGER

The cyclic, unstationary operation of regenerative heat exchangers is responsible for their scarce usage in the recycling of waste heat in a rotary furnace setup. In the words of Kern (1984) their drawbacks include:

a) regenerators are incapable of ensuring a constant temperature of air preheating; the temperature drops down as the regenerator checkerwork is being cooled; this fact limits the possibilities of automatic operation;

b) the supply of heat to the furnace is interrupted at a reversal of regenerator values;

c) with regenerative preheating, some fuel is carried off to stack, the loss of fuel being 5 to 6 percent of the total fuel consumption;

d) the dimensions and mass of regenerators are quite appreciable.

Those drawbacks notwithstanding, regenerators are still used extensively in high-temperature furnaces (open-hearth, blast furnaces, and soaking pits), mainly because they can operate at very high temperatures of waste gases ($1500^0C - 1600^0C$) at which the existing recuperators fail to function stably yet.

In a regenerator, the hot and the cold fluids occupy alternately the same space in the exchanger core. The exchanger core or "matrix", serves as a heat storage device that is periodically heated by the warmer of the two fluids and then transfers heat to the colder fluid (Kern 1984) and (Alan 1984). In a fixed matrix configuration, the hot and cold fluids pass alternately through a stationary exchanger, and for continuous operation, two or more matrices are necessary as shown in figure2.1.1. Another approach is the rotary regenerator in which a circular matrix rotates and alternately exposes a portion of its surface to the hot and then to the cold fluid, as shown in figure2.1.2.

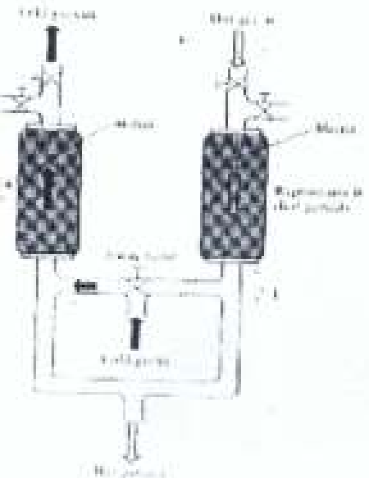

Figure2.1.1:Matrix Configuration Regenerator

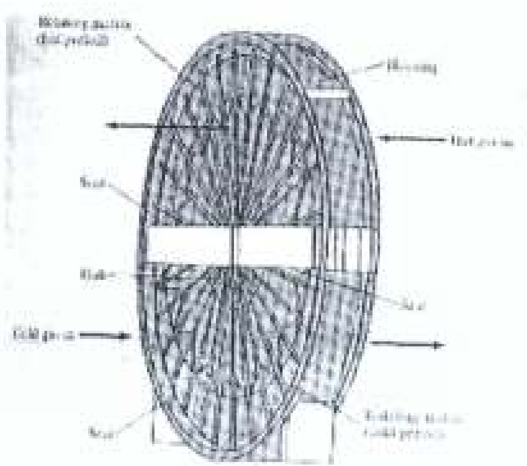

Figure2.1.2: Rotary Regenerator

Figure2.1: Regenerative Heat Exchangers

Source: Kern 1984

2.6.2 RECUPERATIVE HEAT EXCHANGER

The recuperative principle of waste heat recycling is more advanced and perfect. Recuperators ensure a constant temperature of air preheating and require no reversing means; this results in a smoother operation of furnaces and provides greater possibilities for automatic control. Recuperators have smaller dimensions and mass than regenerators and their operation involves no loss of gases to stack (Oyelami and Ogunkoya 2002). They, however, possess some drawbacks among which the principal ones are low fire resistance (of metallic recuperators) and poor gas tightness (of ceramic recuperators). In this type of heat exchanger, the hot and cold fluids are separated by a wall and heat is transferred by a combination of the three established means of heat transfer. The process of heat transfer in recuperators comprises three stages:

a) Heat transfer from waste gases to recuperator walls. This is transferred to the wall by radiation as well as by convection.

b) Heat transfer by conduction through the wall. This depends on the thermal state of the wall and the state of its surface;

c) Heat transfer from the wall to the air being heated. Heat is transferred from the wall to air by convection only.

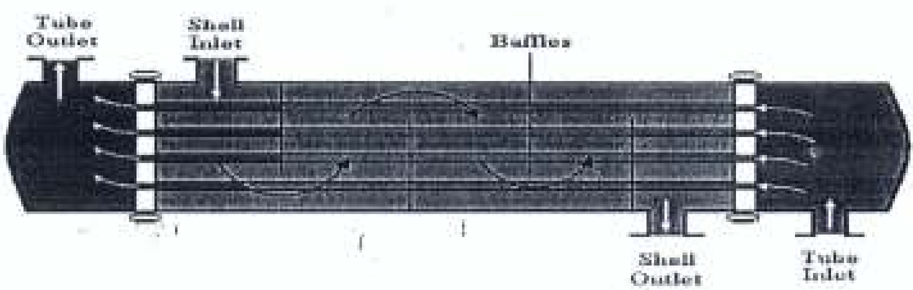

Fig2.2.1 Convection Recuperator (Shell-and-tube-heat exchanger with one shell pass and one tube pass; cross- counterflow operation.)

Source: www.me.wustl.edu/ME/labs/thermal/fig-a1.htm

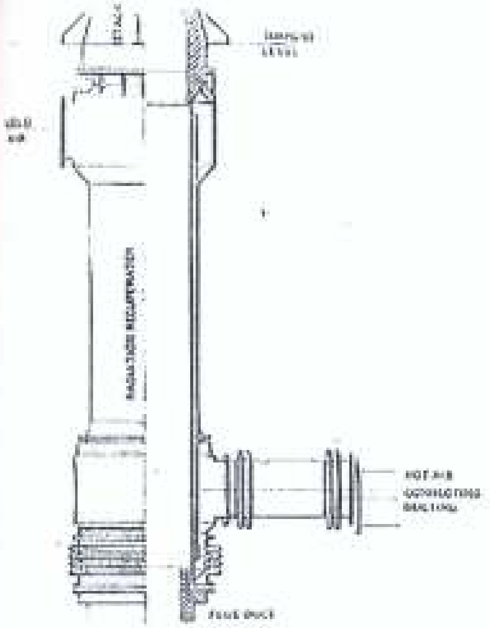

Figure2.2.2: Radiation Recuperator

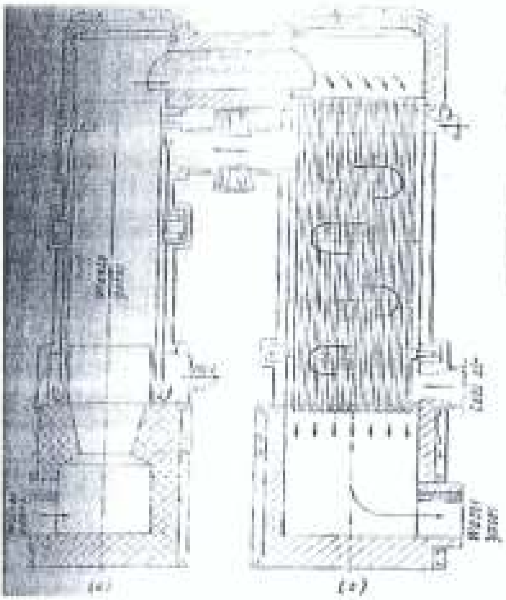

Figure2.2.3: Combined radiation-convection recuperator

(a) radiation section (b) convection section

Figure2.2: Recuperative Heat Exchanger

Source: Kazantsev (1997)

2.6.3 GENERAL PRINCIPLE OF RECUPERATIVE HEAT EXCHANGE

As earlier stated, any recuperator is a heat exchanger operating under steady state thermal conditions, with heat being continuously transferred, in this case, from cooled waste gases to heated air through partition walls. A heat exchanger could also be defined as a device in which energy is transferred from one fluid to another across a solid surface. Exchanger analysis and design therefore involve both convection and conduction (Dryden, 1982) and (Edward 2000).

Two important problems in heat exchanger analysis are (i) rating existing heat exchangers and (ii) sizing heat exchangers for a particular application. Rating involves determination of the rate of heat transfer, the change in temperature of the two fluids, and the pressure drop across the heat exchanger. Sizing involves selection of a specific heat exchanger from those currently vailable or determining the dimensions for the design of a new heat exchanger, given the required rate of heat transfer and allowable pressure drop. The LMTD method can be readily used when the inlet and outlet temperatures of both the hot and cold fluids are known. When the outlet temperatures are not known, the LMTD can only be used in an iterative scheme. In this case the ε-NTU method can be used to simplify the analysis (Robert and Don 1998) and (Kern 1984).

- **Energy Considerations**

The first Law of Thermodynamics, in rate form, applied to a control volume (CV), can be expressed as $\sum_{in} \dot{m} h = \sum_{out} \dot{m} h + \dot{Q}_{surr} + \dot{E}_{st}$ ---------------------------- (2.8)

where $\dot{m}$ stands for mass-flow rate (e.g., 1bm/min or kg/min) crossing the CV boundaries, h is specific enthalpy (energy/mass), $\dot{Q}_{surr}$ is the rate of heat transfer from the CV to its surroundings, and $\dot{E}_{st}$ is the rate of change of energy stored in the CV. This simplified form of the First Law assumes no work- producing processes, no energy generation inside the CV, and negligible kinetic and potential energy in the fluid streams entering and leaving the CV. In steady state operation the energy residing in the CV is constant, meaning that $\dot{E}_{st}=0$. If, furthermore, the boundary of the CV is adiabatic (i.e., perfectly insulated), then $\dot{Q}_{surr}=0$. Under these circumstances Eq. (2.8) reduces to a simple balance of enthalpy inflow and enthalpy outflow:

$\sum_{in} \dot{m} h = \sum_{out} \dot{m} h$ -------------------------------- (2.9)

Applied to a heat exchanger with two streams passing through it, Eq. (2.9) can be rearranged to give $\dot{m}_h (h_{h,i} - h_{h,o}) = \dot{m}_c (h_{c,o} - h_{c,i})$ ---------------------------- (2.10) the subscripts h and c indicate the hot and cold fluids, respectively, and i and o indicate inlet outlet conditions. In words, Eq. (2.10) says that the rate of energy loss by the hot fluid (left-side) equals the rate of energy gain by the cold fluid.

Shell and Tube Heat Exchanger

the schematic diagram of a shell-and-tube heat exchanger with one shell pass and one tube in Figure2.2.1. The cross-counterflow mode of operation is indicated. Inside the heat exchanger the hot and cold fluid temperature distributions would have the form

sketched in Figure 2.3a

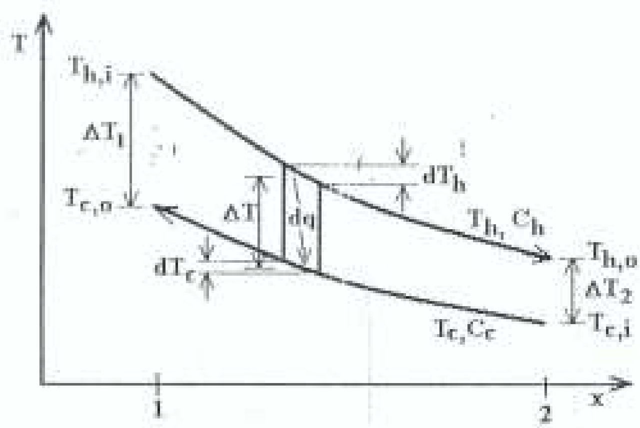

Figure 2.3a: Temperature distributions in a counterflow heat exchanger.

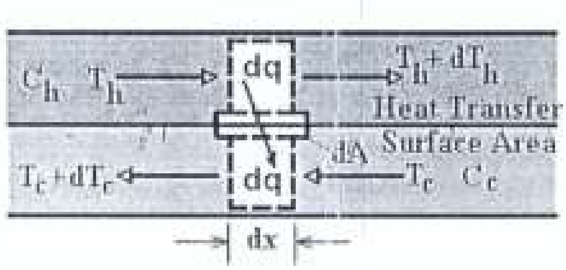

Figure 2.3b Energy balance in a differential length element.

The points 1 and 2 on the x-axis represent the two ends of the heat exchanger. Provided there is no energy loss to the environment and that the exchanger has reached steady state, then dq, the rate of heat transfer from the hot fluid, is exactly equal to the rate of heat transfer to the cold fluid in a differential length dx of the exchanger surface. For the special case of fluids that are not changing phase and have constant specific heats

$$dq = -m_h c_{p,h} dT_h = -C_h dT_h \text{ ------------------------------------- } (2.11)$$

$$dq = -m_c c_{p,c} dT_c = -C_c dT_c \text{ ------------------------------------- } (2.12)$$

where C_h and C_c are called the hot and cold fluid heat-capacity rates, respectively. Integration of Eqs (2.11) and (2.12) along the heat exchanger (from 1 to 2) gives

$$q = C_h (T_{h,i} - T_{h,o}) \text{ ------------------------------------- } (2.13)$$

and

$$q = C_c (T_{c,o} - T_{c,i}) \text{ ------------------------------------- } (2.14)$$

- **Logarithmic Mean Temperature Difference (LMTD) Method**

The differential heat-transfer rate dq across the surface area element dA can also be expressed as

$$dq = U \Delta T dA \text{ ------------------------------------- } (2.15)$$

where ΔT is the local temperature difference between the hot and cold fluids and U is the overall coefficient of heat transfer at dA. Both U and ΔT vary with position inside the heat exchanger (i.e., x), but by combining Eqs (2.11) and (2.12) with Eq. (2.15) it is possible for a single pass exchanger to integrate over the exchanger contact surface from inlet to out. The result of the integration is

$$q = AU_m \Delta T_{lm} \text{ ------------------------------------- (2.16)}$$

where q is the total heat-transfer rate (BTU/min), A is the total internal contact area (ft^2), U_m is the mean overall coefficient of heat transfer (BTU/min ft^2 °F), defined as

$$U_m = \frac{1}{A}\int_0^1 U dA \text{ ------------------------------------- (2.17)}$$

and is the logarithmic mean temperature difference (LMTD), given by

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \text{ ------------------------------------- (2.18)}$$

As shown in Fig2.3a,$\Delta T_1 = T_{h,i} - T_{c,o}$ and $\Delta T_2 = T_{h,o} - T_{c,i}$ for the counterflow, single pass case. Equation (2.18) is also applied to more complicated heat-exchanger designs with multipass and cross- flow arrangements with a correction factor applied to the LMTD (Kern 1984), (Alan, 1984), (Dryden, 1982) and (Krivadin and Markov, 1980)

2.6.4 HEAT TRANSFER COEFFICIENTS

The total quantity of heat transferred in a recuperator is found by the equation

$$\mathbf{Q = K\Delta t_{av}A \quad , \quad W} \text{ --- (2.19)}$$

Where **K** is the total coefficient of heat transfer from waste gases to air, which characterizes the total level of heat exchange in the recuperator, W/(m^2,^{0}C); $\mathbf{\Delta t_{av}}$ is the average (over the heating surface) temperature difference between waste gases and air, ^{0}C; and **A** is the area of the surface through which heat is transferred from waste gases to air, m^2. The heat of waste gases is ansferred to the wall by radiation as well as by convection. Therefore the local coefficient of heat transfer at the waste-gas side of the recuperator is

$$\alpha_{wg} = \alpha'_{wg} + \alpha'_{wg} \quad \text{--} \ (2.20)$$

where α'_{wg} is the coefficient of heat transfer from waste gases to the wall by convection, W/(m^2.^{0}C); and α'_{wg} is the coefficient of heat transfer by radiation, W/(m^2.^{0}C).

The heat transfer by conduction through the wall depends on the thermal state of the wall, $R = s/\lambda$ and the state of its surface (where s = thickness of the wall, m; λ = thermal conductivity of the wall, W/(m^2.^{0}C)). On the air side of a recuperator, heat is transferred from the wall to air by convection only. If a gas is heated in a recuperator, the heat transfer from the wall to the gas can occur both by convection and radiation. Thus, with air preheating, the heat transfer process is decided by the local convective heat transfer coefficients $\alpha_a = \alpha_a^c$

All these heat transfer coefficients can be combined into the total heat transfer coefficient

$$K = \frac{1}{1/\alpha_{wg} + s/\lambda + 1/\alpha_a} \quad \text{--} \ (2.21)$$

With tubular recuperator, the total heat transfer coefficient should be determined as for a cylindrical (linear heat transfer coefficient)

$$K = \frac{2\pi}{1/\alpha_1 r_1 + (1/\lambda)\ln r_2/r_1 + 1/\alpha_2 r_2} \ \mathrm{W/(m^2.^0C)} \quad \text{------------------------} \ (2.22)$$

The coefficient K is called the heat transfer coefficient of a tube. If the quantity of heat transferred should be related to the area of the outer or inner surface of a tube, the total heat transfer coefficient can be found as follows:

$$K_1 = K/2\pi r_1 \ \ \mathrm{W/(m^2.^0C)} \quad \text{--} \ (2.23)$$

And $$K_2 = K/2\pi r_2 \ \ \mathrm{W/(m^2.^0C)} \quad \text{--------------------------------------} \ (2.24)$$

where α_1 is the coefficient of heat transfer at the inner surface of a tube, $W(m^2\,^0C)$; α_2 is the same, at the outer surface , $W/(m^2\,^0C)$; r_1 and r_2 are the inside and outside radii of tube, m.

In metallic recuperators, the thermal resistance of the wall, s/λ can be neglected and the total heat transfer coefficient will be written as

$$K = \frac{\alpha_{wg}\alpha_a}{\alpha_{wg} + \alpha_a}, \quad W/(m^2\,^0C) \text{ .... } (2.25)$$

All local heat transfer coefficients in this formula can be found from the general laws of heat transfer by convection and radiation.

- **THE DIRT FACTOR**

The overall heat transfer coefficient expressed by equations 2.21 and 2.22 is known as the clean overall coefficient of heat transfer. By using the clean overall coefficient of heat transfer, it is assumed that the recuperator remains in the same clean condition from at the day it is produced till its life span. This is not however so in practice. In practice, when the recuperator has been in service from some time, dirt and scale deposit on its inside and outside, adding two more resistances than were included in the calculation of the clean overall coefficient of heat transfer (K_c). The additional resistances reduce the original value of K_c, and the required amount of heat is no longer transferred by the original surface A; the final temperature of the waste gases rises above and that of the air falls below the desired outlet temperatures, although α_{wg} and α_a remain substantially constant. To overcome this eventuality, it is customary in designing the recuperator to anticipate the deposition of dirt and scale by introducing a resistance R_d called the *dirt, scale or fouling factor* or resistance. If R_{di} is the dirt factor for the inner cylinder fluid at its inside diameter and R_{do} the dirt factor for annulus fluid at the outside diameter of the inner cylinder. These may be considered very thin for dirt but may be appreciably thick for scale, which has a higher thermal conductivity than dirt. The coefficient of heat transfer which includes the dirt resistance is called the *design or dirty overall coefficient of heat transfer* (K_d). The value of surface area A corresponding to K_d rather than K_c provides the basis on which the recuperator is ultimately built.

The relationship between the two overall coefficients K_d and K_c is

$$\frac{1}{K_d} = \frac{1}{K_c} + R_{di} + R_{do} \quad \text{(2.26)}$$

if $R_{di} + R_{do} = R_d$

$$\frac{1}{K_d} = \frac{1}{K_c} + R_d \quad \text{(2.27)}$$

(Robert and Don 1998), (Alan 1984) and (Kern, 1984)

2.6.5 THE TEMPERATURE FIELD OF RECUPERATOR

Fluids in recuperators can move in cocurrent, crosscurrent or countercurrent flow. Graphs of temperature variation over the heating surface in countercurrent and cocurrent flows are shown in figure2.4 (Krivadin and Markov 1980)

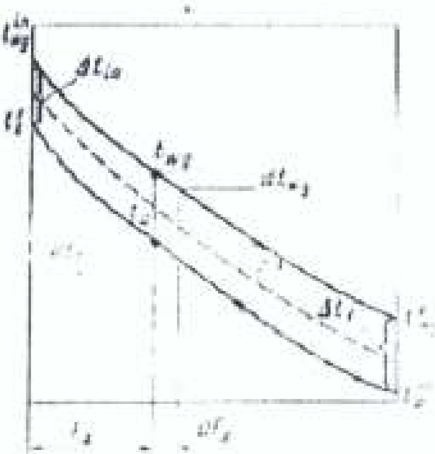

Figure2.4.1: Countercurrent Flow

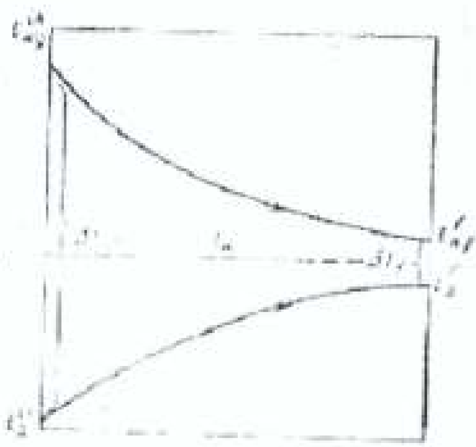

Figure2.4.2: Co-current Flow

Figure2.4: Diagrams of motion of gas media in Recuperative Heat Exchanger

Source: (Krivadin and Markov 1980)

It can be seen from the graph that with the countercurrent flow, the final temperature of air, $t_a^{/}$ be can higher than the final temperature of waste gases, $t_{wg}^{/}$. This is impossible with the cocurrent flow in a recuperator. On the other hand, with countercurrent flow the temperature of a recuperator wall can be appreciably higher than with co-current flow. For that reason counter flow is effectively used in ceramic and metallic recuperators at relatively low temperature of waste gases. Co-current flow is employed in metallic recuperators in cases where the temperature of waste gases is so high that that the recuperator material can lose its stability.

Both with countercurrent and co-current flows, the temperature of waste gases and that of air vary along the heating surface, resulting in a variable temperature difference between waste gases and air. This is why use is made of average temperature difference over the whole heating surface, Δt_{av}.

2.7 FUELS AND COMBUSTION

Fuels are a source of energy, which when converted to heat can serve the purpose of raising the temperature of furnaces. Selection of the fuel depends on various factors, but availability is often the sole deciding factor. To avoid dependence on a single fuel, combustion equipment is often chosen in which alternative or combined use of fuels is possible.

All major fuels basically contain carbon, hydrogen e.t.c in organic or inorganic forms. Combustion is an oxidation reaction producing carbon dioxide, water vapor and sulghur dioxide accompanied by heat release. The total amount of heat obtainable by complete combustion, in air, of unit quantity of a fuel is its calorific value (CV). If the fuel contains hydrogen, water vapor is one of the products of combustion and if condensed the latent heat of condensation is added to the heat of combustion, giving the gross calorific value. In general practice, however, combustion gases are not condensed in working parts of the furnace, and deducting this latent heat from the gross calorific value gives the net calorific value. Fuels can be classified in the broadest terms as gaseous, liquid and solid fuels. The selection of the fuel best suited to a particular case necessitates possession of knowledge concerning their more important properties and the equipment needed for their handling and combustion (Oyelami et al 2003) and (Dryden, 1982).

CHAPTER THREE

3.0 MATERIALS AND METHODS

3.1 MATERIALS SELECTION

Furnaces are built up of metal parts and heat resisting non-metallic materials (refractories). The strength and durability of furnaces are affected by factors such as vibrations, temperature and its variations, the effects of chemicals and slags. This makes it highly essential to have a proper knowledge of the available materials and their properties.

3.1.1 METALS USED IN FURNACE CONSTRUCTION

Scale-resistant (heat-resistant) and refractory steels and alloys are needed for the construction of the furnace structures. Heat-resistant steels are those capable of operating in unloaded or weakly loaded state and withstanding chemical aggression in gases at temperatures above 550°C. Refractory steels on the other hand can operate under load at high temperatures for a sufficiently long time; they are also scale-resistant. **(Cottrel, 1975)**

A comprehensive list of corrosion and heat resistant steels with their characteristics and typical applications needed in the construction of the furnace and its incorporated heat exchanger is as shown below.

Table 1.0: Corrosion and Heat Resistant Steels with their Characteristics and Typical Applications

Corrosion and Heat Resistant Steels				
BS 3146 Part II, 1975	Grade	Material	Characteristics	Typical Applications
ANC 1	A B C	13% Cr Martensitic steels	A general engineering type stainless steel, offering a range of strengths and hardnesses. Medium corrosion resistance.	A. Gas chemical and petroleum industries; high ductility engineering fittings, golf club heads. B. Heat resistant parts not subject to high stresses. C. Cutting blades, pump and steam turbine parts.

ANC 2		18% Cr 2% Ni Martensitic steels	A high tensile stainless steel with improved corrosion resistant properties. Resistant to oxidising atmospheres up to 760°C.	Pump and valve parts; highly stressed aircraft and general engineering fittings.
ANC 3	A B	18% Cr 10% Ni Austenitic steels	A. corrosion and acid restraint stainless steel; excellent stability down to - 225°C. B. Weldable version of the above, usable up to 800°C.	A. Chemical, pharmaceutical, textile, dairy and oil industries, e.g. pump and valve parts. B. Exhaust systems and marine fittings to a certain extent. Corrosion/acid resistant parts not heat-treated after welding.
ANC 4	A B C	18% Cr 11% Ni 3% Mo Austenitic steels	Stainless steels with good corrosion and acid resistance, with medium tensile strength.	In the chemical and processing industries, e.g. valves and pumps handling acids at high temperatures and also chlorides and salts.
ANC 5	A B C	Ni-Cr steels	Heat resistant alloys with resistance to cyclic heating and useful creep strength up to 650°C and good resistance to scaling.	Furnace parts, salt and lead baths.
ANC 6	A B C	Cr-Ni steels.	Heat resistant alloys with good strength to 900°C and useful creep strength to 650°C.	Heat treatment parts and superheaters, welding fixtures. High temperature castings. Nozzle guide vanes for gas turbines.
ANC 8		**Ni-base 20% Cr; 0.4% Ti alloy**	**A readily weldable heat-resistant alloy with excellent resistance to oxidization up to 1100°C and with useful strength.**	**Furnace parts.**
ANC 9		Ni-base 20% Cr; 2.5% Ti, 1.2% Al alloy	A heat-resistant alloy with good creep and oxidization resistance up to 870°C.	Diesel engine pre-combustion chambers, gas turbine parts.
ANC 10		Ni-base 20% Cr 16.5% Co, 2.4% Ti, 1.3% Al alloy	Increased strength over the ANC 9 alloy with good creep and oxidization resistence up to 870°C.	Turbine and turbocharger rotors.
ANC 11		Ni-base 20% Cr 10% Co, 10% Mo alloy	Medium strength alloy with excellent resistance to oxidization and thermal fatigue at temperatures over 1000°C, good thermal shock resistance up to 1100°C.	Gas turbine and stator blades.
ANC 13		Co-base 26% Cr 10% Ni, 7% W alloy	A high strength heat-resistant alloy. Resistant to oxidisation at high temperatures, corrosion, galling and wear. Good resistance to creep and thermal shock.	Impellers, hot metal dies and valve components.
ANC 14		Ni-base 27% Cr 5.6% Mo, 2.7% Ni alloy	A high strength heat-resistant alloy. Resistant to oxidisation at high temperatures and corrosion resistant.	Impellers, gas turbine components and valve components for high

			Resistant to wear and with good low temperature impact properties.	temperature service.
ANC 15		Ni-base 28% Mo alloy	A corrosion and heat-resistant alloy. Good resistance to sulphuric and phosphoric acids; excellent resistance to concentrated, hot hydrochloric acid and acid chlorides. Useful creep strength up to 800°C.	Chemical and petroleum plant components and pickling equipment.
ANC 16		Ni-base 17% Mo 16.5% Cr, 4.5% W alloy	A corrosion and heat-resistant alloy. Resistant to oxidising acids (e.g. nitric) at high temperatures. Useful creep strength up to 800°C.	Chemical and petroleum plant components.
ANC 17		Ni-base 9% Si 3% Cu alloy.	A corrosion resistant alloy, particularly against hot sulphuric acid.	Chemical and petroleum plant components.
ANC 18	A B C	Ni-base 31% Cu Si alloy	Corrosion resistant alloys with a range of hardness for general engineering purposes. Resistant to superheated steam, sea-water, mineral acids. Retention of strength and toughness up to 450°C.	Power plant, marine equipment, chemical and process industry components.
ANC 19		Ni-base 20% Cr, 7%Nb, 6% Mo, 3% Fe, 3% W alloy	A high strength precipitation hardening alloy resistant to thermal shock and oxidation.	Diesel engine combustion chamber inserts.
ANC 20	A B	14% Cr, 5% Ni, 2% Cu, 1% Mo steels	A high strength precipitation hardening steel with good corrosion resistance and good weldability.	Marine applications where high strength and good corrosion resistance are required.
ANC 21		26% Cr, 5% Ni, 3% Cu, 2% Mo steel	Good corrosion resistance; comparable to ANC 3 with higher strength.	Marine applications.

SOURCE: http://www.ceramicast.com/materials.html

3.1.2 REFRACTORIES USED

As earlier stated, refractories are materials which can withstand high temperatures (usually above 1580°C) and the physical and chemical actions of molten metal, slag and gases in furnaces. The materials used for the lining of the furnace, the exhaust and the conical base of the heat exchanger are

- Silica sand
- Kaolin
- Building Brick (from fusible clay)

- Sodium Silicate and
- Water

3.2 DESIGN CONSIDERATION

Under same conditions, the atmosphere affects all metals and alloys. Carbon dioxide and excess oxygen may react with the metal heated and cause scaling. Scale can be removed by expensive method like pickling, but surface finish may alter and dimensional tolerances may be impaired.

Since rotary furnaces are now being widely used in heating/melting ferrous metals, besides scaling, decarburization is a common problem. Oxidation or decarburization is linked closely with the nature of the furnace atmosphere and to the time and temperature of exposure. Typical furnace atmosphere (resulting from complete or partial combustion or introduction of prepared atmosphere) may contain one or more of a number of gases such as oxygen, carbon dioxide, carbon monoxide, water vapour, hydrogen, methane etc. Even though the main aim of this project is to design a recuperator-incorporated rotary furnace with more emphasis on the recuperator, the resultant effect of the conditions inside the furnace highlighted above make it essential to take cognizance of the designs of other essential parts.

3.2.1 DESIGN CONDITIONS

The conditions to be created in a furnace should be such that the heat flow to the surface of material is at its maximum and that it can be absorbed completely. This can be ensured by the following measures:

- The temperature difference between the heating medium and heated surface should be as high as feasible under the given processed conditions
- The furnace should be supplied with heat extensively with the maximum heat utilization within the furnace space

- Gases in the furnace space should be made to circulate vigorously, mainly under the action of air fuel jets from burners
- Fuel should be burned completely and combustion should occur within the furnace space where possible
- The design of a pressure conditions in it should be such as to minimize the contact of furnace atmosphere with the surroundings

3.2.2 THE BURNER SYSTEM

The burner, which is a fuel-burning device, converts the chemical energy of fuel into thermal energy which is required to effect technological processes in the furnace. A low-pressure liquid-fuel burner is used in the operation of a recuperator-incorporated rotary furnace. Liquid-fuel burners should meet the following requirements:

- they should atomize liquid fuel and mix it thoroughly with air;
- ensure stable combustion and form an uninterrupted flame of a desired length;
- should be reliable in operation, simple in design and convenient for cleaning.

A typical characteristic of burner is the mode of fuel-air mixture. There are the premixing (injector) burners, burners with partial premixing and the outside-mixing (flame) burners. Outside-mixing (flame) burner is the most effective type of burner used for a rotary furnace most especially when incorporated with a recuperator. The reason being that it allows for variations in the flame length. The flame length must be controllable because before reaching a steady state, the recuperator first produces variations in thermal load.

Assuming that the optimal thermal load of the burner surface is H_b and is approximately equal to 930×10^3 W/m^2, then the quantity of heat emitted by the burner is

$$q_b = \frac{H_b \pi d^2}{4} \quad \text{---} \quad (3.1)$$

The other most important parameters in low-pressure burners are the outlet cross-sectional areas for liquid fuel and air. The outlet cross-sectional area for liquid fuel is found as

$$f_f = Ab/\mu_f\sqrt{p_f\rho_f}\ , \quad m^2 \text{ -- } (3.2)$$

where

A = a coefficient which is equal to 195.625

b = fuel oil flow rate, kg/h

p_f = fuel oil pressure, N/m^2

μ_f = flow rate coefficient for fuel oil (ranges from 0.2 - 0.3)

ρ_f = fuel oil density

The air outlet cross-sectional area for the low-pressure burner is

$$f_a = A^1V_a/\mu_a\sqrt{P_a/\rho_a} \qquad \text{ ------------------------------------ } (3.3)$$

where

f_a = outlet cross-sectional area for air, mm^2

V_a = air flow rate through the burner, m^3/h

P_a = total head of air at the outlet, N/m^2

μ_a = air flow rate coefficient (ranges from 0.7 – 0.8)

A^1 = a coefficient, which is equal to 618.75

ρ_a = air density, kg/m^3

3.2.2.1 FLAME LENGTH

As was rightly pointed out at the beginning of this report, flame length is critical to the design of a rotary furnace, whether incorporated with a recuperator or not. A burning gas or an atomized liquid or solid fuel forms a flame, when burnt in an oxygen-containing atmosphere. A visible flame is a luminous stream of incandescent gases. A flame has a certain geometric shape and dimensions which are required to suit the hearth or working space of a furnace. Failure to optimize these two parameters is bound to lower furnace efficiency, shorten the service life of refractories, pollute the environment and cause other undesirable effects. The length of a free turbulent flame, L_f is calculated from

$$L_f = d_0[N_0(1-w) - \frac{1}{b_0}]\sqrt{\frac{\rho_g}{\rho_{mb}[1+I_b+E_n+f(w)f_1(\rho)\frac{z^1}{F_r}]}} \text{ --------------------- (3.4)}$$

where

$z = L_f/d_0$ = dimensionless length of the flame expressed in diameters of gas nozzle, d_0

N_0 and b_0 = constants of a turbulent jet respectively equal to 11.2 and 0.47;

$\frac{1}{b_0}$=2.14

w = stoichiometric number that shows the amount of air (kg) required to burn 1kg of fuel according to a stoichiometric calculation of combustion, kg/kg; $w = \frac{L_{th} \cdot \rho_{n.a}}{\rho_{n.g}}$ where L_{th} is the theoretical volume of air, m^3, required to burn 1m^3 of fuel, m^3/m^3

$\rho_{n.a}$ and $\rho_{n.g}$ = respectively densities of air and gas under normal conditions, kg/m^3

$f(w)$ = dimensionless function of the stoichiometric number w, which in calculations can be assumed to equal 14.8x10^{-4}

$f_1(\rho)$ = dimensionless function of characteristic densities

F_r = Froude number

$$f_1(\rho) = \frac{\rho_a}{\rho_g}\left[\frac{\rho_a}{\rho_{atm}} - 1\right] \quad \text{------------------------------------} \quad (3.5)$$

where ρ_a and ρ_g respectively densities of air and gas, kg/m^3

3.2.2.2 TEMPERATURE OF COMBUSTION AS A FUNCTION OF TIME

A ***stationary*** thermal state is a situation whereby the transfer of heat in a body involves no change in temperature with time, that is, $dt/d\tau = 0$. However the situation in a rotary furnace is a ***non-stationary*** thermal state whereby $dt/d\tau \neq 0$, that is, the temperature of the body changes with time, and the enthalpy of the body varies. The analytical relationship between the change in temperature and in the quantity of transferred heat with time for any point of a body may be obtained by solving the basic differential equation of heat conductivity which in the Cartesian coordinates has the form

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} + \frac{\partial^2 t}{\partial z^2}\right) = a\nabla^2 t \quad \text{--} \quad (3.6)$$

where ∇^2 is the differential Laplacian operator.

In the cylindrical system of coordinates the differential equation of heat conductivity takes on the form

$$\frac{\partial t}{\partial \tau} = a\left(\frac{\partial^2 t}{\partial r^2} + \frac{1}{r}\frac{\partial t}{\partial r} + \frac{1}{r^2}\frac{\partial^2 t}{\partial \Phi^2} + \frac{\partial^2 t}{\partial z^2}\right) \quad \text{--} \quad (3.7)$$

To obtain a single-valued solution for the equation of heat conductivity, it is combined with the equations for edge conditions consisting of (a) starting conditions and (b) surface or boundary conditions.

For a linear change in temperature of the surface of a body, that is, heating of a body at a constant rate C_h and uniform distribution of temperatures at the starting moment, the boundary conditions are

$$t_{x=\pm S} = t_i + C_h \tau \; ; \; t_{\tau=0} = t_i = F(x) \quad \text{------------------------} \quad (3.8)$$

The solution in general form for a cylinder of infinite length is

$$\frac{t - t_{sw}}{C_h \tau} = \Phi\left(\frac{a\tau}{R^2}; \frac{r}{R}\right) \quad \text{------------------------} \quad (3.9)$$

Values of function Φ depend on the Fourier number and the dimensionless coordinates of points.

At $\frac{a\tau}{R^2} \geq 0.5$, the expression for t, which represents the average temperature of combustion in the furnace, is

$$t = t_i + \frac{C_h R^2}{4a}\left(\frac{r^2}{R^2} - 1\right) \quad \text{------------------------} \quad (3.10)$$

where

t_i = room temperature, ^{0}C;

C_h (heating rate, KJ/h) = B Q_n^l ;

Q_n^l = Lower Calorific value of the fuel used, KJ/kg;

B = fuel flow rate, kg/h;

τ = heating period, h;

r = internal radius of the furnace cylindrical shell, m;

S = thickness of the cylindrical shell, m;

R = r + S, m;

a (coefficient of conductivity, m^2/h) = $\frac{\lambda}{\rho c}$;

c = heat capacity, KJ/(kg ^{0}C)

λ = coefficient of heat conductivity

ρ = density, kg/m^3. **(Kazantsev, 1997)**

3.2.3 DETERMINATION OF THE FURNACE-CYLINDER INTERNAL DIAMETER AND LINING THIÇKNESS

The above-mentioned parameters are two of the major parameters needed in the design of the rotary furnace in addition to the flame length desired. To effectively calculate the parameters, it is highly essential to do an analysis of the heat transfer in the furnace.

3.2.3.1 HEAT TRANSFER ANALYSIS IN ROTARY FURNACE

In analyzing the heat transfer process, basic assumptions made are:

1. The temperature and radiant characteristics of the gases are taken to be constant over the whole furnace volume
2. The temperature in various points of metal surface is assumed to be constant;
3. The heat flow transferred from the gases to furnace lining by convection is taken to be equal the heat flow given up by the lining to the surrounding. This last assumption makes it possible to regard the lining as adiabatic relative to the radiant flux falling onto it.

3.2.3.1.1 INPUT HEAT

1. Chemical Heat of fuel combustion

$$Q_{ch} = BtQ^l_w \quad KJ \qquad \text{------------------------------------} (3.11)$$

Where t = total time taken to preheat and melt the charges (h)

Q^l_w = lower Calorific value of the fuel used in KJ/kg

B = fuel flow rate in kg/h

2. Physical Heat of preheated air

$$Q_{ph} = Btc_a t_a n v_a \quad \text{------------------------} (3.12)$$

Where t_a = temperature of preheated air (^0C)

c_a = specific heat capacity of air at T_a ($KJ/m^{3\,0}C$)

n = air excess factor

v_a = quantity of air theoretically required to burn a unit of fuel m^3/kg

3. Physical Heat of preheated fuel

$$Q_f = Bt_f C_f T_f \quad (KJ) \quad \text{------------------------} (3.13)$$

Where t_f = time taken for preheating the fuel (h)

C_f = specific heat capacity of preheated fuel (KJ/kgK)

T_f = temperature of preheated fuel (K)

4. Heat of exothermic reactions which is basically the heat liberated through oxidation of metal. Assuming the oxidation less of iron is 1% and its heating value is 5652KJ/kg

$$\Rightarrow \quad Q_{ex} = 5652\ Mt \times 0.01 \quad (KJ)$$

$$= 56.52Mt\ (KJ) \quad \text{------------------------} (3.14)$$

where M = mass of the charges (kg).

Total Heat input $Q_i = Q_{ch} + Q_i + Q_f + Q_{ex}$ (KJ) ------------------------ (3.15)

Since only the air is preheated

$$Q_i = Q_{ch} + Q_{ex} + Q_a \quad \text{------------------------} (3.16)$$

3.2.3.1.2 OUTPUT HEAT

1. Useful heat required to heat and melt the materials:

$$Q_1 = [MC_m(T_m - T_0) + ML_m]t$$

$$= Mt[C_m(T_m - T_0) + L_m] \quad (KJ) \quad \text{------------------------} (3.17)$$

where

M = mass of the charges (kg/h)

C_m = specific heat capacity of the charges (KJ/kgK)

T_m = melting point temperature of the charges (K)

T_0 = initial temperature of the charges (K)

L_m = Specific latent heat of melting of the charges (KJ/kg)

2. Heat lost with waste gases

$$Q_2 = B\, t\, v_{wg}\, t_{wg} \ (KJ) \quad \text{--} \ (3.18)$$

Where

c_{wg} = mean specific heat of waste gases ($KJ/m^3\,{}^0C$)

t = total time taken (h)

B = fuel flow rate in kg/h

v_{wg} = quantity of the waste gases in a unit mass of the fuel (m^3/kg)

t_{wg} = temperature of the waste gases (${}^0C$)

3. Heat of the chemical incompleteness of fuel combustion

$$Q_3 = B\, v_{wg}\, a\, t \times 12142KJ \quad \text{--} \ (3.19)$$

Where a = proportion of unburnt CO in the waste gases = 0.015

4. Heat from the mechanical incompleteness of combustion

$$Q_4 = 0.01\, Q_{ch}\, (KJ) \quad \text{--} \ (3.20)$$

5. Heat losses by conduction through the lining

$$Q_c = Q_5 = [\pi K_L (r_2 + r_1)(T_1 - T_2)(L_1 + L_2)t] / (r_2 - r_1) \quad \text{----------------------} \ (3.21)$$

Where K_L = thermal conductivity of the lining (W/mK)

r_2 = junction radius = internal radius of the shell without the lining (m)

r_1 = internal radius of the shell with the lining (m)

T_1 = average temperature in the furnace (K)

T_2 = outside (surface) temperature of the shell (K)

T_0 = room temperature (K)

L_1 = length of the shell (m)

L_2 = slant length of the conical frustum (m)

t = total time taken (h)

$T_2 = 0.2T_1$; $L = L_1 + L_2$

$$\Rightarrow \quad Q_c = [\pi LtK_1(T_1 - 0.31T_1)(r_2 + r_1)] / (r_2 - r_1)$$

$$= 0.69\pi LtK_1 T_1(r_2 + r_1) / (r_2 - r_1)$$

Let $y = 0.69\pi LtK_1 T_1$

$$\Rightarrow \quad Q_c = y(r_2 + r_1) / (r_2 - r_1) \quad \text{------------------------------} (3.22)$$

$$r_2Q_c - r_1Q_c = yr_2 + yr_1$$

$$r_1(Q_c + y) = (Q_c - y)r_2$$

$$r_1 = (Q_c - y)r_2 / (Q_c + y) \quad \text{------------------------------} (3.23)$$

6. Heat losses by radiation through open doors of the furnace

$$Q_6 = C_0(T_1/100)^4F\Phi\varphi \quad (KJ) \quad \text{------------------------------} (3.24)$$

Where C_0 = emissivity of a black body = 50768 W/m^2K^4

F = surface area of open door (m^2)

Φ = diaphragming coefficient ≈ 0.67

φ = time when the door is open (h)

7. Heat lost with the cooling water (steam) in the furnace

$$Q_7 = 0.15(\text{Heat input})$$

$= 0.15Q_{i}$ -- (3.25)

8. Loss of the heat accumulated in the lining

$Q_8 = 0.2Q_i$ -- (3.26)

9. Unaccounted losses

$Q_9 = 0.2(Q_3 + Q_4 + Q_c + Q_6 + Q_7 + Q_8)$

Let $Q_o = Q_3 + Q_4 + Q_6 + Q_7 + Q_8$

$\Rightarrow \quad Q_9 = 0.2(Q_o + Q_5)$

Equating the heat input and heat output

$\Rightarrow \quad Q_i = (Q_1 + Q_2 + Q_3 + Q_4 + Q_c + Q_6 + Q_7 + Q_8 + Q_9)$

If $Q_t = Q_1 + Q_2 + Q_o$

$\Rightarrow Q_i = Q_t + Qc + Q_9$

$Q_i = Q_t + Q_c + 0.2Q_o + 0.2Q_c$

$= Q_t + 0.2Q_o + 1.2Q_c$

$\therefore \quad Q_c = (Q_i - Q_t - 0.2Q_o) / 1.2$ -- (3.27)

from (3.18)

$r_1 = (Q_c - y)r_2 / (Q_c + y)$

let $(Q_c - y) / (Q_c + y) = u$

$\Rightarrow \quad r_1 = ur_2$ -- (3.28)

3.2.3.2 Compressive strength of the lining

Let the Compressive strength be σ

Then $\sigma = \delta \Delta t_c E_c / [r_2(r_2 - r_1)\{1 + \Delta t_c E_c / (E_L(r_2 - r_1))\}$ ---------------- (3.29)

Where

δ = lining shrinkage allowance (m)

Δt_c = thickness of the cylindrical shell without the lining (m)

E_c = Young's modulus of cylinder (Pa)

E_L = Young's modulus of lining (Pa) **(Dryden, 1982)**

$\Rightarrow \quad \sigma = \delta\Delta t_c E_c E_L / r_2[E_L(r_2 - r_1) + \Delta t_c E_c]$

$[E_L(r_2 - r_1) + \Delta t_c E_c]r_2 = \delta\Delta t_c E_c E_L / \sigma$

let $k_1 = \delta\Delta t_c E_c E_L / \sigma$

$k_2 = \Delta t_c E_c$

$E_L r_2(r_2 - r_1) + k_2 r_2 = k_1$

$E_L r_2^2 - E_L r_1 r_2 + k_2 r_2 - k_1 = 0$

$E_L r_2^2 - (E_L r_1 - k_2)r_2 - k_1 = 0$ ---------------------------- (3.30)

Substitute (3.23) into (3.25)

$\Rightarrow \quad E_L r_2^2 - (uE_L r_2 - k_2)r_2 - k_1 = 0$

$E_L r_2^2 - uE_L r_2^2 + k_2 r_2 - k_1 = 0$

$(E_L - uE_L)\, r_2^2 + k_2 r_2 - k_1 = 0$

$r_2 = [-k_2 \pm \sqrt{\{k_2^2 + 4k_1(E_L - uE_L)\}}\ ] / 2(E_L - uE_L)$

let $k_3 = \sqrt{\{k_2^2 + 4k_1(E_L - uE_L)\}}$

$\Rightarrow r_2 = [-k_2 + k_3] / 2(E_L - uE_L)$

$$r_2 = \frac{-t_c E_c + \sqrt{t_c^2 E_c^2 + \dfrac{4\delta t_c E_c E_L}{\sigma}[E_L - E_L(\dfrac{Q_c - 0.69\pi LtK_L T_1}{Q_c + 0.69\pi LtK_L T})]}}{2[E_L - E_L(\dfrac{Q_c - 0.69\pi LtK_L T_1}{Q_c + 0.69\pi LtK_L T})]} \text{ ---- (3.31)}$$

$r_1 = ur_2$

lining thickness $= \Delta t_L = r_2 - r_1$ -- (3.32)

3.2.4 DESIGN OF THE TRANSMISSION SHAFT

The design of the transmission shafting is based on the maximum shear stress theory and takes into account the effect of fatigue and shock by introducing two constants in to the equation of maximum shear stress. The diameter of the solid transmission shaft used is thus calculated from

$$d^3 = \frac{16}{\pi S_s}\sqrt{(K_b M_b)^2 + (K_t M_t)^2} \qquad \text{-------------------------------------- (3.33)}$$

where

M_t = torsional moment, Nm

M_b = bending moment, Nm

K_b = combined shock and fatigue factor applied to bending moment

(K_b=1.5)

K_t = combined shock and fatigue factor applied to torsional moment

(K_t=1.0)

S_s = maximum allowable shear stress N/m^2 **(Sadhu, 1999)**

3.2.5 DESIGN OF KEY AND KEY-WAYS

A key is a device by means of which the relative motion between the two members connected by the key is prevented. In this case, keys are used to connect the rollers and the shafts.

Sunk keys are used in this design. A sunk key is a type of key which when inserted into the keyway is partly in the shaft and partly in the hub.

The width of the key is

$$b = \frac{d}{4} \qquad \text{-------------------------------------- (3.34)}$$

and its length is

$$l = \frac{\pi d^2}{8b} \quad \text{-- (3.35)}$$

where

d = shaft diameter

b = key width and

l = key length **(Allen et al, 1980)**

3.2.6 BEARING SELECTION

A rolling-contact bearing is chosen to be the most appropriate one because of the following reasons:

- More accurate centering of the shaft
- Low coefficient of friction
- Weak dependence of the coefficient of friction on operating conditions
- Low resisting moments during starting periods
- Small axial dimensions
- Ability to operate with a small oil feed
- Ability to operate within a wide temperature range from temperatures close to absolute zero to 600^{0}C coupled with the ability to operate in high vacuum.

The service life (durability) of rolling-contact bearings is determined by the number of load cycles that the bearing material can withstand at a given load and, therefore, depends on the speed of rotation. The durability sharply decreases with increased loads. **(Orlov,1980)**

3.2.7 BLOWER DESIGN/SELECTION

Blower is the medium through which the combustion air needed for the generation of heat from burner is supplied. While a medium-pressure blower (of pressure up to 3000N/m^2 or 300mm H_2O)

could be used effectively with a rotary furnace without a recuperator, a high-pressure blower of well over 10,000 N/m^2 (1,000mm H_2O) pressure is needed when incorporated with a recuperator.

One of the most important characteristics of the blower is its specific speed or specific number of revolutions n_{sp}. The specific speed n_{sp} of a blower is a value that interrelates air delivery v m^3/h, total pressure H mm H_2O under standard atmospheric conditions (at P=760mm Hg, t=20°C) and relative humidity of 50%, and the number of revolution per minute, n

that is $$n_{sp} = \frac{nv^{1/2}}{H^{3/4}} \qquad \text{------------------------------------} \ (3.36)$$

The technical data obtainable for single-inlet exhaust blowers with multi-blade wheel at a preheated air temperature of 200°C (air preheated by the recuperator) are as shown in the table below:

Table 3.0: Single-Inlet Multi-Blade Blowers

Type	Maximum delivery Q_{max}, m^3/h	n, rpm	Total Pressure H, mm H_2O	Efficiency, $\frac{Q_{max}}{Q_{sp}}$	Consumed power N, kW	Initial moment of rotor, kg.m^2	Net mass, kg
Д-10-12.5 $H_{ш}$	105000 135000	730 960	155 265	54/60 54/60	80 180	93 93	1915 1915
Д-10-13 $H_{ш}$	115000 155000	730 960	165 285	54/60 54/60	95 220	111 111	1960 1960
Д-10-13.5 $H_{ш}$	130000 170000	730 960	175 310	54/60 54/60	120 265	128 128	2000 2000
Д-10-14 $H_{ш}$	145000 190000	730 960	190 330	54/60 54/60	140 320	127 127	1980 1980
Д-10-14.5 $H_{ш}$	160000 210000	730 960	160 210	54/60 54/60	165 385	154 154	2025 2025
Д-10-12.5 H_{40}	105000 130000	730 960	135 235	54/60 54/60	70 155	86 86	1900 1900
Д-10-13 H_{40}	115000 155000	730 960	150 255	54/60 54/60	85 200	102 102	1935 1935
Д-10-13.5 H_{40}	130000 170000	730 960	160 275	54/60 54/60	100 240	116 116	1975 1975
Д-10-14 H_{40}	145000 190000	730 960	175 295	54/60 54/60	125 290	123 123	1955 1955
Д-10-14.5 H_{40}	160000 210000	730 960	185 320	54/60 54/60	145 340	140 140	1995 1995

SOURCE: [Kazantsev, 1980]

3.2.8 RECUPERATOR SPECIFIC DESIGN CALCULATIONS

(FOR A 100KG ROTARY FURNACE)

These specific design calculations are meant to demonstrate the procedures for determining the recuperator's basic dimensions with the aim of incorporating the procedures into the program to be written (the program shall be for evaluating the essential dimensions and parameters needed for the design of a recuperator-incorporated rotary furnace)

The first step taken in the design of the recuperator is to determine the process conditions. This includes the analysis of the fluid (waste gases) compositions, determination of the temperatures of operation and the pressures drop. The second step was to obtain all the required physical properties over the temperature and pressure ranges of interest. The third stage involved choosing the type of recuperator needed while the fourth and the last stage was the real design and re-design (whenever necessary) to get the necessary dimensions and ensure that the process specifications with respect to both heat transfer and pressure drop are met.

3.2.8.1 ANALYSIS OF STREAM COMPOSITIONS

The analysis is done to obtain not only the percentage compositions of the combustion products (the waste gases), but also to evaluate some important parameters that are needed for the analysis of heat transfer. Those parameters include among others the total quantity of waste gases formed through combustion of unit mass of fuel and the quantity of air theoretically required to burn a unit of fuel. Due to its availability, diesel is chosen as the working fuel.

Diesel contains 98.5% $C_{12}H_{26}$ (1%S, 0.5%N & O) and has a combustion equation

$$C_{12}H_{26} + \frac{37}{2}O_2 \rightarrow 12CO_2 + 13H_2O$$

$$(12\times12+26\times1)\text{kg } C_{12}H_{26} + (37/2 \times 32)\text{kg } O_2 \rightarrow 12(12+32)\text{kg } CO_2 + 13(2+16)\text{kg } H_2O$$

$$170\text{kg } C_{12}H_{26} + 592\text{kg } O_2 \rightarrow 528\text{kg } CO_2 + 234\text{kg } H_2O$$

1kg $C_{12}H_{26}$ + 592/170 kg $O_2 \rightarrow$ 528/170kg CO_2 + 234/170 kg H_2O

1kg of $C_{12}H_{26}$ + 3.482kg O_2 produces $\rightarrow$ 3.106kg CO_2 + 1.376kg H_2O

(but 1kg of fuel contains 98.5% $C_{12}H_{26}$)

$\Rightarrow$ 0.985kg of $C_{12}H_{26}$ + 3.43kg O_2 produces 3.06kg CO_2 + 1.355kg H_2O

Amount of O_2 needed = 3.43/32 x 22.4 = 2.401m^3/kg of fuel

Amount of N_2 that accompanies the air needed to produce 3.43kg O_2 is

76.8/23.2 x 3.43 = 11.35kg/kg

= 11.35/28 x 22.4 = 9.08m^3/kg

Air needed per unit mass of fuel is v_a = 2.401 + 9.08 = 11.481m^3/kg

CO_2 : 3.06x22.4/44 = 1.558m^3/kg

H_2O : 1.355x22.4/18 = 1.686m^3/kg

Waste gases produced: v_{wg} = 1.558 + 1.686 + 9.08 = 12.324m^3/kg

Table 2.0: Volumetric analysis of the combustion per 100m^3 of $C_{12}H_{26}$ (101.5m^3 of fuel)

Fuel		Air, m^3			Combustion products, m^3					
Component	Content, m^3	O_2	N_2	Total	CO_2	SO_2	H_2O	O_2	N_2	Total
$C_{12}H_{26}$	98.5	18822.25	1823.75x3.76 = 6860.77	1823.75+6860.77 = 8684.52	1182	-	1280.5	-	0.25+6860.77 = 6861.02	2462.5
S	1	1			-	1	-	-		1
O_2	0.25	-			-	-	-	0.25		0.25
N_2	0.25	0.5			-	-	-	-		6861.02
Σm^3 (%) at n = 1	100	1823.75 (21.0)	6860.77 (79.0)	8684.52 (100)	1182 (12.676)	1 (0.011)	1280.5 (13.732)	0.25 (0.003)	6861.02 (73.578)	9324.77 (100)
at n = 1.2	100	2188.5 (21.0)	8232.92 (79.0)	10421.42 (100)	1182 (10.686)	1 (0.009)	1280.5 (11.576)	365 (3.3)	8233.17 (74.429)	11061.67 (100)

Source: Oyelami et al, 2003

3.2.8.2 DETERMINATION OF OPERATIONAL TEMPERATURES

The average temperature of the waste gases leaving the furnace is 1050°C and that of the surrounding air is taken to be at the room temperature of 27°C. Since one of the main purposes of signing the recuperator is to increase the temperature of combustion significantly, it is pertinent to determine the existing temperature of combustion (for the existing rotary furnace without recuperator), choose a new desired temperature to be accomplished as a result of the recuperator incorporation and calculate the final temperatures of both the air and the waste gases necessary for the attainment of the new combustion temperature.

- **FURNACE COMBUSTION TEMPERATURE WITHOUT RECUPERATOR**

To calculate the calometric temperature in the furnace, enthalpy of combustion I_0 is

$$I_0 = \frac{Q_0^l}{V_{ng}} \qquad \text{--} (3.37)$$

$$I_0 = \frac{44000}{12.323} = 3570.27 \text{ KJ/m}^3$$

where $Q_0^l = Q_w^l = 44000\text{KJ/kg}$

Enthalpy of combustion products at 2000^0C is calculated thus (using the appropriate percentage composition of combustion products of diesel from table 2.0)

$$I_{CO_2} = 0.10686 \times 4910.51 = 524.74 \text{ KJ/m}^3$$

$$I_{N_2} = 0.74429 \times 2970.25 = 2210.73 \text{ KJ/m}^3$$

$$I_{O_2} = 0.033 \times 3142.76 = 103.71 \text{ KJ/m}^3$$

$$I_{H_2O} = 0.11576 \times 3889.72 = \underline{450.27 \text{ KJ/m}^3}$$

$$I_t = 3289.45 \text{ KJ/m}^3$$

Enthalpy of combustion products at 2300^0C is

$$I_{CO_2} = 0.10686 \times 5746.39 = 614.06 \text{ KJ/m}^3$$

$$I_{N_2} = 0.74429 \times 3457.20 = 2573.16 \text{ KJ/m}^3$$

$$I_{O_2} = 0.033 \times 3662.33 = \quad 120.85 \text{ KJ/m}^3$$

$$I_{H_2O} = 0.11576 \times 4485.34 = 519.22 \text{ KJ/m}^3$$

$$I_2 = 3827.29 \text{ KJ/m}^3$$

e calorimetric temperature is

$$t_c = t_1 + \frac{I_0 - I_1}{I_2 - I_1}(t_2 - t_1)$$

$$t_c = 2000 + \frac{3570.27 - 3289.45}{3827.29 - 3289.45}(2300 - 2000)$$

$$= 2156.6\ ^0\text{C}.$$

he furnace maximum combustion temperature is

$$t_{comb} = \eta t_c \quad \text{---} \quad (3.38)$$

here η = pyrometric coefficient.

Using a pyrometric coefficient of 0.7,

$$t_{comb} = 2156.6 \times 0.7$$

$$= 1509.62\ ^0\text{C}.$$

This value compares favourably well with the 1492 ^{0}C maximum combustion temperature obtained from an operational rotary furnace (check table 4.1)

- **CALCULATION OF FINAL TEMPERATURE OF AIR EXPECTED FROM THE RECUPERATOR**

As calculated from the preceding section, the maximum combustion temperature obtainable from the rotary furnace without a recuperator is 1509.62 ^{0}C. Now, it is desired to raise the temperature to 2000 ^{0}C through the incorporation of the recuperator. This is the basis used for calculating the final temperature of the air to be preheated in order to have the furnace temperature raised to

2000 ^{0}C.

From equation 3.38,

$$t_t = \frac{t_{comb}}{\eta} \text{ -- } (3.39)$$

$$t_t = \frac{2000}{0.7} = 2857.14^0C$$

Enthalpy of combustion products at 2857.14^{0}C is

I_{CO_2} = 0.10686 x 6303.53 = 673.60 KJ/m^3

I_{N_2} = 0.74429 x 3786.09 = 2817.95 KJ/m^3

I_{O_2} = 0.033 x 4014.29 = 132.47 KJ/m^3

I_{H_2O} = 0.11576 x 5076.74 = 587.68 KJ/m^3

I_0 = 4211.70 KJ/m^3

From equation 3.37

$$Q_0^l = I_0 \times v_{\infty g} = 4211.70 \times 12.324$$

= 51905 KJ/kg

$$Q_0^l = Q_u^l + Q_{ph}^l \text{ -- } (3.40)$$

where $Q_{ph}^l = \dfrac{Q_n}{Bt}$ and $Q_u^l = \dfrac{Q_{ch}}{Bt}$ = 44000KJ/kg (using equations 3.11 and 3.12)

$$Q_{ph}^l = Q_0^l - Q_u^l$$

=51905 - 44000 = 7905KJ/kg

From equation 3.12,

$$t_e = \frac{Q_n}{Bt} \cdot \frac{1}{nc_n v_n}$$

where $\frac{Q_a}{Bt} = Q'_{ph} = 7905 \times 10^3$ J/kg; n = 1.2; c_a (at 400^{0}C) = 1.3302 x 10^3 J/m$^{3\,0}$C; v_a = 11.481m^3/kg

$$t_a = 7905 \times 10^3 . \frac{1}{1.2 \times 1.3302 \times 10^3 \times 11.481}$$

$$= 431.34^0C$$

Thus the final temperature of air t_a^f used for the design is 450^{0}C

- **CALCULATION OF FINAL TEMPERATURE OF WASTE GASES EXPECTED FROM THE RECUPERATOR**

This is done through the analysis of the heat balance in the recuperator. On the assumption that the recuperator is perfectly gas-tight, the heat balance is analyzed based on further assumption that 10 per cent of the heat supplied is lost to the surroundings.

$$0.9V_{wg}^*(c_{wg}^{in}t_{wg}^{in} - c_{wg}^f t_{wg}^f) = V_a(c_a^f t_a^f - c_a^{in} t_a^{in}) \quad \text{------------------------------} \quad (3.41)$$

Using $t_{wg}^{in} = 1050^0C$, $t_a^f = 450^0C$, $t_a^{in} = 27^0C$

To find the specific heat of waste gases at the recuperator outlet, it is assumed that $t_{wg}^f = 900^0C$.

From table 2.0, the composition of waste gases is 12.7%CO_2; 13.7%H_2O and 73.6%N_2. The specific heats of waste gases are

at 900^{0}C

c_{CO_2} = 0.127 x 2.1915 = 0.2783 KJ/m^3

c_{N_2} = 0.736 x 1.3817 = 1.0169 KJ/m^3

c_{H_2O} = 0.137 x 1.6865 = 0.2783 KJ/m^3

$c_{wg}^f = c_{\sum}$ = 1.5263 KJ/m^3

at 1050^0C

$$c_{CO_2} = 0.127\times[2.2266+\frac{1050-1000}{1100-1000}(2.2593-2.2266)] = 0.2849 \text{ KJ/m}^{3\,0}\text{C}$$

$$c_{N_2} = 0.736\times[1.3938+\frac{1050-1000}{1100-1000}(1.4056-1.3938)] = 1.0302 \text{ KJ/m}^3$$

$$c_{H_2O} = 0.137\times[1.7133+\frac{1050-1000}{1100-1000}(1.7397-1.7133)] = 0.2365 \text{ KJ/m}^3$$

$$c_{wg}^{in} = c_{\sum} = 1.5516 \text{ KJ/m}^3$$

The specific heat capacity of air is

at 27^0C

$$c_a^{in} = 1.3009+\frac{27-0}{100-0}(0.3051-1.3009) = 1.03203\text{KJ/m}^{3\,0}\text{C}$$

at 450^0C

$$c_a^f = 1.3302+\frac{450-400}{500-400}(1.3440-1.3302) = 1.3371\text{KJ/m}^{3\,0}\text{C}$$

Using $1\frac{1}{2}$ % hrs as the standard cycle period, from table 2.0

$V_{wg} = 9324.72/1.5 = 6217\text{m}^3/\text{h}$

$V_a = 8684.52/1.5 = 5790\text{m}^3/\text{h}$

The heat balance equation, with 10 per cent of heat being lost to the surroundings from equation 3.41 is

$$0.9\times 6217(1.5516\times 1050-1.5263t_{wg}^f) = 5790(1.3371\times 450-1.03203\times 27)$$

$$5595.3(1629.18-1.5263t_{wg}^f) = 3322476.8$$

$$t_{wg}^{f} = (1629.18 - \frac{3322476.8}{5595.3}) \times \frac{1}{1.5263} = 678.36^{0}C$$

To close up the gap between the assumed t_{wg}^{f} of $900^{0}C$ and the calculated t_{wg}^{f} of $678.36^{0}C$, the specific heat of the waste gases is re-calculated at an assumed t_{wg}^{f} of $700^{0}C$ and the process repeated

at $700^{0}C$

$c_{CO_2} = 0.127 \times 2.1077 = 0.2677$ KJ/m^3

$c_{N_2} = 0.736 \times 1.3553 = 0.9975$ KJ/m^3

$c_{H_2O} = 0.137 \times 1.6338 = \underline{0.2279}$ KJ/m^3

$$c_{wg}^{f} = c_{\Sigma} = 1.4931 \text{ KJ/m}^3$$

Using the same heat balance equation with the ten percent of the heat being lost to the surrounding

$$0.9 \times 6217(1.5516 \times 1050 - 1.4931 t_{wg}^{f}) = 5790(1.3371 \times 450 - 1.03203 \times 27)$$

$$5595.3(1629.18 - 1.4931\, t_{wg}^{f}) = 3322476.8$$

$$t_{wg}^{f} = (1629.18 - \frac{3322476.8}{5595.3}) \times \frac{1}{1.4931} = 693.44^{0}C$$

- **AVERAGE TEMPERATURE DIFFERENCE IN THE RECUPERATOR**

For a co-current flow of the type shown in figure2.4.2, using the logarithmic mean temperature difference (LMTD) approach in section 2.6.3, equation 2.18 can be modified to give

$$\Delta t_{m} = \frac{\Delta t_{m} - \Delta t_{f}}{\ln(\Delta t_{m} / \Delta t_{f})} \qquad \text{------------------------------------ (3.42)}$$

$$\Delta t_{av} = \frac{(t_{wg}^{in} - t_a^{in}) - (t_{wg}^{f} - t_a^{f})}{\ln[\frac{t_{wg}^{in} - t_a^{in}}{t_{wg}^{f} - t_a^{f}}]} \quad \text{--} (3.43)$$

$$\Delta t_{av} = \frac{(1050 - 27) - (693.44 - 450)}{\ln[\frac{1050 - 27}{693.44 - 450}]}$$

$$\Delta t_{av} = \frac{1023 - 243.44}{\ln[\frac{1023}{243.44}]} = 543^{0}C$$

3.2.8.3 CHOICE OF APPROPRIATE RECUPERATOR

The heat transfer in a recuperator between the gases and the heating surface is by convection, which depends on the velocity of the waste gases across the heating surface or by radiation from the water vapour and carbon dioxide in the waste gases, which depends on the thickness of the layer, the concentration of these components in the gases, and on the temperature. The type of recuperator chosen for this design is the radiation type. The choice of radiation recuperator is chosen after due consideration to the following factors, as outlined by (Stephen, 2000):

- Initial cost
- Operating temperature
- Desired preheat
- Pressure level and pressure loss
- Campaign life and reliability
- Ease of repair and site considerations
- Resistance to corrosion
- Resistance to fracture

Basic requirements taken into consideration for the eventual design of the radiation recuperator are its ability to:

- ensure the maximum utilization of the heat of waste gases
- have an appropriate stability to withstand the high temperature of waste gases
- have dimensions as small as feasible, in order for it to have a high heating surface area per cubic metre of recuperator volume
- have its total heat transfer coefficient at a maximum; this minimizes the size of the recuperator
- have a minimum hydraulic resistance and
- ensure an appropriate gas tightness.

It was decided to make the chosen radiation-type recuperator have its working fluid flow in co-current flow because it has been established that the final temperature of the waste gases after transferring all the desired heat to the air, is still greater than the final temperature of the heated air. This condition perfectly suits co-current flow from which it is impossible to have the final temperature of the hot carrier not to be greater than the final temperature of the cold carrier. This possibility cannot be guaranteed in a counter-current flow situation.

3.2.8.4 DETERMINATION OF THE BASIC DESIGN PARAMETERS

A typical radiant recuperator, according to Kazantsev (1997), has an appreciable thermal load of the heating surface ranging from 250 to 335 $MJ/(m^2.h)$.

Taking the average of the two, the expected thermal load is

$$(250+335)/2 = 292.5 MJ/(m^2.h)$$

$$= 292.5 \times 10^6/3600 = 81250 W/m^2$$

Heat transfer coefficient is the thermal load per unit temperature.

Therefore, Heat Transfer of the heating surface is

$$K_c = \frac{ThermalLaod}{t_{wg}^{m}}$$

$K_c = 81250/1050 = 77.38\ W/m^{2\,0}C$

This is the clean overall heat transfer coefficient which shows that dirt has not been taken into account. The design overall coefficient is the dirty overall coefficient K_d.

From equation 2.27

$$\frac{1}{K_d} = \frac{1}{K_c} + R_d$$

or

$$K_d = \frac{K_c}{1 + R_d K_c} \quad \text{---} \ (3.44)$$

For diesel combustion products, $R_d = 0.00175\ m^{2\,0}C/W$

$$K_d = \frac{77.38}{1 + 0.00175 \times 77.38} = 68.15\ W/m^{2\,0}C.$$

The heating surface covered by the waste gases before reaching t_{wg}^{l} (from equation 2.19) is

$$A = \frac{Q}{K\Delta t_{av}} \quad \text{---} \ (3.45)$$

where Q as calculated from the heat balance is

$$Q = \frac{332247.8 \times 10^3 / 2}{3600} = 46145.43W; \qquad K = K_d = 68.15W/m^{2\,0}C$$

$$\Delta t_{av} = 543^0C$$

$$A = \frac{46145.43}{68.15 \times 543} = 1.247m^2$$

The heating surface area is $2\pi rl$.

From equation 3.31 and 3.32 for a 100kg rotary furnace:

Diameter of the cylindrical shell = 0.496929m = 496.929mm

Thickness of the lining = 0.12732m = 127.32mm

Inner shell diameter (after lining) = 496.929-127.32x2

= 242.289mm

The recuperator inner diameter is then chosen as 250mm

Recuperator sprout base diameter = 496.929-127.32

= 369.609

The recuperator's sprout base diameter is then chosen as 375mm and its slant length as 370mm.

From A = 2πrl = πdl, $l = \frac{A}{\pi d} = \frac{1.247}{3.142 \times 0.25} = 1.588\ m^2$

- **ANNULUS SPACE**

The equivalent diameter $D_e = \frac{4 \times FlowArea}{WettedPerimeter} = \frac{4\pi(D_0^2 - D_i^2)}{4\pi D_i} = \frac{D_0^2 - D_i^2}{D_i}$ ---------------(3.46)

Using 44%D_i as D_e

$$0.44D_i = \frac{D_0^2 - D_i^2}{D_i},$$

$$D_0 = \sqrt{1.44}D_i = \sqrt{1.44} \times 0.25 = 0.3m = 300mm$$

- **DETERMINATION OF THE HEAT TRANSFER COEFFICIENTS**

If the heat transfer coefficient of the waste gases is represented by h_{wg}, then

$$h_{wg} = \frac{\alpha_1 \lambda_{wg}}{D_i}\left(\frac{\rho_{wg} u_{wg} D_i}{\mu_{wg}}\right)^{-0.9}\left(\frac{c_{wg}\mu_{wg}}{\lambda_{wg}}\right)^{-1/3}\left(\frac{\mu_{wg}}{\mu_w}\right)^{0.6} \text{------------------------- (3.47)}$$

where α_1 = 1.3718

D_i = recuperator's inner diameter = 0.25m;

u_{wg} = 14.58m/s;

$$\mu_{wg}\,(Absolute Viscosity) = v_{wg}\,\rho_{wg}$$

$$v_{wg}\,(\text{Kinematic Viscosity}) = [93.61 + \frac{650-600}{700-600}(112.1-93.61)]\times 10^{6}$$

$$= 102.345\text{x}10^{6}\ \text{m}^2/\text{s}$$

$$\rho_{wg} = 0.405 + \frac{650-600}{700-600}(0.363-0.405) = 0.384\text{kg/m}^3$$

$$\mu_{wg} = 102.345\text{x}10^{6}\ \text{x}\ 0.384 = 39.3\ \text{x}\ 10^{6}\ \text{kg/m.s}$$

c_{wg} = 1493.1J/m$^{3\,0}$C

$$\lambda_{wg} = 744 + \frac{650-600}{700-600}(829-744) = 786.5\text{W/m}^0\text{C}$$

$$v_{w} = [60.38 + \frac{450-400}{500-400}(76.30-60.38)]\times 10^{6} = 68.34\ \text{x}\ 10^{6}$$

$$\rho_{w} = 0.525 + \frac{450-400}{500-400}(0.457-0.525) = 0.491\text{kg/m}^3$$

$$\mu_{w} = 68.34\ \text{x}\ 10^{6}\ \text{x}\ 0.491 = 33.55\ \text{x}\ 10^{6}\ \text{kg/m.s}$$

Using equation 3.47

$$h_{wg} = \frac{1.3718\times 786.5}{0.25}(\frac{0.384\times 14.58\times 0.25}{39.3\times 10^{6}})^{-0.9}(\frac{1493.1\times 39.3\times 10^{6}}{786.5})^{-1/3}(\frac{39.3}{33.55})^{0.6}$$

$$= 293.2\ \text{W/m}^{2\,0}\text{C}$$

If the heat transfer coefficient of the air is represented by h_a, then

$$h_a = \frac{\alpha_2 \lambda_a}{D_a}(\frac{\rho_a u_a D_a}{\mu_a})^{-0.9}(\frac{c_a \mu_a}{\lambda_c})^{-1/3}(\frac{\mu_a}{\mu_w})^{0.6} \text{ ------------------------- (3.48)}$$

where α_2 = 1.652

D_o = recuperator's outer diameter = 0.30m;

u_a = 25m/s;

$\mu_a (Absolute Viscosity) = v_a \rho_a$

$$v_a \text{ (Kinematic Viscosity)} = [63.09 + \frac{450-400}{500-400}(79.38-63.09)]\times 10^6$$

$$= 71.235\text{x}10^6 \text{ m}^2/\text{s}$$

$$\rho_a = 0.524 + \frac{450-600}{500-400}(0.456-0.524) = 0.49\text{kg/m}^3$$

μ_a = 71.235x10^6 x 0.49 = 34.905 x 10^6 kg/m.s

c_a = 1337.1J/m^3 ^{0}C

$$\lambda_a = 522 + \frac{450-400}{500-400}(575-522) = 548.5\text{W/m}^0\text{C}$$

$$v_w = [60.38 + \frac{450-400}{500-400}(76.30-60.38)]\times 10^6 = 68.34 \text{ x } 10^6$$

$$\rho_w = 0.525 + \frac{450-400}{500-400}(0.457-0.525) = 0.491\text{kg/m}^3$$

μ_w = 68.34 x 10^6 x 0.491 = 33.55 x 10^6 kg/m.s

Using equation 3.48

$$h_a = \frac{1.652\times 548.5}{0.3}(\frac{0.49\times 25\times 0.3}{34.905\times 10^6})^{-0.9}(\frac{1337.1\times 34.905\times 10^6}{548.5})^{-1/3}(\frac{34.905}{33.55})^{0.6}$$

$$= 148.88 \text{ W/m}^2\,^0\text{C}$$

- **DETERMINATION OF THE RECUPERATOR'S WALL THICKNESS**

Thickness of the recuperator's wall is determined from the equation expressing the overall heat transfer coefficient in terms of the wall thickness, among other variables is as expressed in equation2.22 from which

$$K_e = \frac{2\pi}{1/\alpha_1 r_1 + (1/\lambda)\ln r_2/r_1 + 1/\alpha_2 r_2} \text{ W/(m}^2.^0\text{C)}$$

Re-arranging the equation gives

$$\ln r_2 + \frac{\lambda}{\alpha_2 r_2} + \frac{\lambda}{\alpha_1 r_1} - \frac{2\pi\lambda}{K_e} - \ln r_1 = 0 \text{ ------------------------------------ (3.49)}$$

where $K_e = 77.38$ W/m$^{2\,0}$C; $\alpha_1 = \alpha_a = 148.88$ W/m$^{2\,0}$C; $r_1 = r_i = 0.125$m

$\alpha_2 = \alpha_{wg} = 293.1$ W/m$^{2\,0}$C; $\lambda = 28.8$ W/m^{0}C

$$\ln r_2 + \frac{28.8}{293.2r_2} + \frac{28.8}{148.88\times0.125} - \frac{2\pi\times28.8}{77.38} - \ln 0.125 = 0 \text{ --------------------- (3.50)}$$

If s=0.001m, $r_2 = 0.126$m; equation 3.50 becomes

$$\ln 0.126 + \frac{28.8}{293.2\times0.126} + \frac{28.8}{148.88\times0.125} - \frac{2\pi\times28.8}{77.38} - \ln 0.125 = -0.0034619 \neq 0$$

If s=0.002m, $r_2 = 0.127$m; equation 3.50 becomes

$$\ln 0.127 + \frac{28.8}{293.2\times0.127} + \frac{28.8}{148.88\times0.125} - \frac{2\pi\times28.8}{77.38} - \ln 0.125 = -0.0016949 \neq 0$$

If s=0.003m, $r_2 = 0.128$m; equation 3.50 becomes

$$\ln 0.128 + \frac{28.8}{293.2\times0.128} + \frac{28.8}{148.88\times0.125} - \frac{2\pi\times28.8}{77.38} - \ln 0.125 = 0.00010601 \approx 0$$

If s=0.004m, $r_2 = 0.129$m; equation 3.50 becomes

$$\ln 0.129 + \frac{28.8}{293.2\times0.129} + \frac{28.8}{148.88\times0.125} - \frac{2\pi\times28.8}{77.38} - \ln 0.125 = 0.00193956 \neq 0$$

$\therefore$ $r_2 = 0.128$m satisfies the equation. The thickness of the recuperator is then chosen as 0.003m = 3mm thick.

3.2.8.5 CALCULATION OF THE PRESSURE DROP

The pressure-drop allowance is the static fluid pressure, which is expended to drive the fluid through the exchanger. The standard allowable pressure drop is 10 psi = 10 x 10^5/14.696 = 68046 N/m^2

For the inner cylinder

$$f = 0.0035 + \frac{0.264}{\left(\frac{\rho_{wg} u_{wg} D_i}{\mu_{wg}}\right)^{0.42}} \quad \text{-------------------------------------- (3.51)}$$

where f = Dimensionless factor in the Fanning equation.

$$f = 0.0035 + \frac{0.264}{\left(\frac{0.384 \times 14.58 \times 0.25}{39.3 \times 10^{6}}\right)^{0.42}} = 354.75$$

The Fanning equation is

$$\Delta F_{wg} = \frac{2 f u_{wg}^2 l}{g D_i} = \frac{2 \times 354.75 \times 14.58^2 \times 1.571}{9.81 \times 0.25} = 96.711 \times 10^3 \text{ Nm/kg}$$

$$\Delta P_{wg} = \Delta F_{wg} \rho_{wg} = 96.711 \times 10^3 \times 0.384 = 37.137 \times 10^3 \text{ N/m}^2$$

This falls well within the allowable pressure drop of 68046 N/m^2.

- For the pressure drop in the annulus, the equivalent diameter is

$$D_e = \frac{4 \times FlowArea}{FrictionalWettedPerimeter} = \frac{4\pi(D_0^2 - D_i^2)}{4\pi(D_0 + D_i)} = D_0 - D_i$$

$$= 0.3 - 0.25 = 0.05\text{m}$$

$$f = 0.0035 + \frac{0.264}{\left(\frac{0.49 \times 25 \times 0.05}{34.905 \times 10^{6}}\right)^{0.42}} = 477.57$$

$$\Delta F_a = \frac{2 f u_{a}^2 l}{g D_e'} = \frac{2 \times 477.57 \times 25^2 \times 1.571}{9.81 \times 0.05} = 1.9127 \text{ x } 10^3 \text{ Nm/kg}$$

Neglecting the entrance and exit losses

$$\Delta P_a = \frac{\Delta F_a \rho_a}{144} = \frac{1.9127 \times 10^6 \times 0.49}{144} = 65080 \text{ N/m}^2$$

This also falls well within the allowable pressure drop of 68046 N/m^2.

3.3 GEOMETRIC MODELING OF THE RECUPERATOR-INCORPORATED ROTARY FURNACE

Geometric modeling is the usage of mathematical description of the geometry/properties of an object in order to be able to display an image of the model on a graphics terminal and to perform certain operation, such as creating new geometric models from basic building blocks available in the system, moving the images around on the screen, zooming in on certain features of the images of the model.

The mathematical descriptions of the object, in this case a recuperator-incorporated rotary furnace; have been done in the preceding sections. Even though the desired geometric modeling could be done either as wire-frame models or solid models, this project lays emphasis on solid models using solid modeler software. The reason being that a wire frame model uses interconnecting lines to depict an object, which can become somewhat confusing for complicated geometry like a recuperator-incorporated rotary furnace because all the lines depicting the shape of the object are usually shown, even the lines representing the other side of the object. Solid models on the other hand are modeled in three dimensions thereby providing the user with the vision of the object very much like it would be seen in real life. Nevertheless, whenever deemed necessary, a wire-frame model is used to supplement the developed solid model in this project.

3.3.1 GEOMETRIC MODELS DEVELOPED FOR THE ROTARY FURNACE

Figures 3.1 to 3.32 represent the various geometric models developed for all the essential components of the rotary furnace and its drive system.

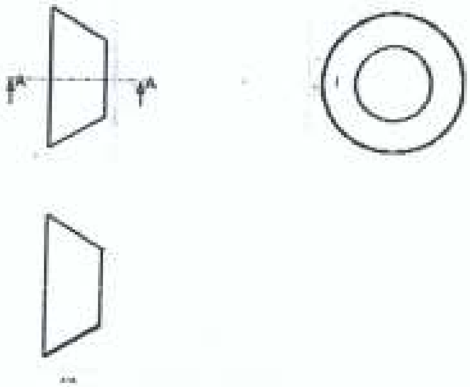

Figure 3.1: Barrel Funnel End1

Figure 3.2: Barrel Funnel End2

Figure 3.3: Barrel Funnel End3

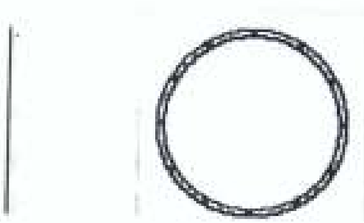

Figure 3.4: Barrel Flange

Figure 3.5: Barrel Funnel End Disc

Figure 3.6: Barrel Funnel End Spront

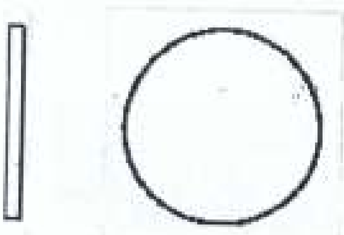

Figure 3.7: Barrel Ends

Figure 3.8: Barrel Ends Assembly1

Figure 3.9: Barrel Ends Assembly2

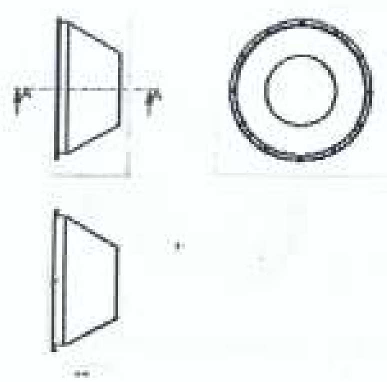

Figure 3.10: Barrel Ends Assembly3

Figure 3.11: Furnace Base Channel 1

Figure 3.12: Furnace Base Channel 2

Figure 3.13: Furnace Base Plate

Figure 3.14: Furnace Shaft

Figure 3.15: Furnace Wheel/Roller

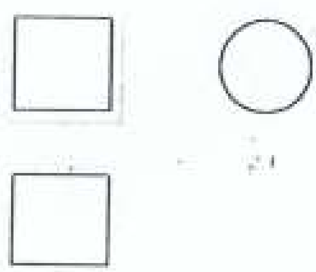

Figure 3.16: Main Barrel 1

Figure 3.17: Main Barrel 2

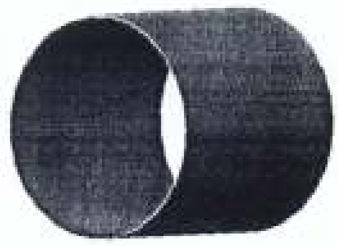

Figure 3.18: Main Barrel 3

Figure 3.19: Main Barrel Assembly1

Figure 3.20: Main Barrel Assembly2

Figure 3.21: Main Barrel Assembly3


Figure 3.22: Main Barrel Funnel Assembly1


Figure 3.23: Main Barrel Funnel Assembly2

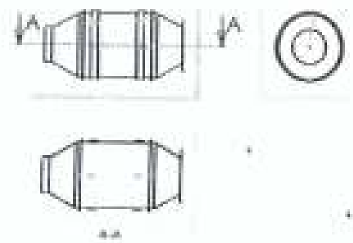
Figure 3.24: Main Barrel Funnel Assembly2


Figure 3.25: Plume Block


Figure 3.26: Roller Pad

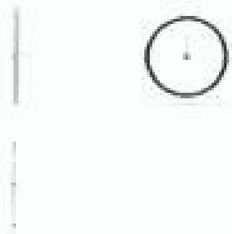
Figure 3.27: Roller Pad Ends Flange


Figure 3.28: Rotary Base Assembly1

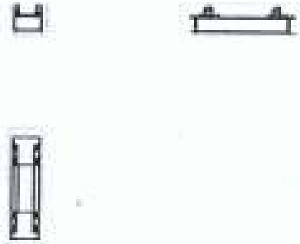
Figure 3.29: Rotary Base Assembly2


Figure 3.30: Main Rotary Assembly1


Figure 3.31: Main Rotary Assembly2


Figure 3.32: Main Rotary Assembly3

3.3.2 GEOMETRIC MODELS DEVELOPED FOR THE BLOWER ASSEMBLY

Figures 3.33 to 3.51 represent the various geometric models developed for all the essential units of the blower system, needed to supply the air needed for the combustion of fuel

Figure 3.33: Blower Overview

Figure 3.34: Blower Inlet

Figure 3.35: Blower Inlet Flange

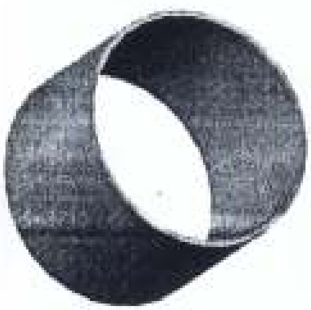

Figure 3.36: Blower Inlet Pipe

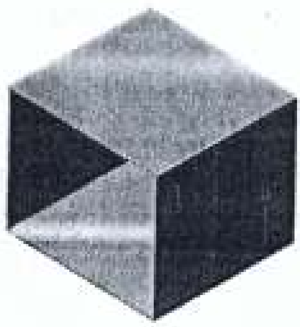

Figure 3.37: Blower Base

Figure 3.38: Blower to Recuperator Duct1

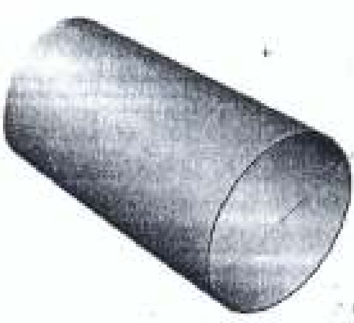

Figure 3.39: Blower to Recuperator Duct2

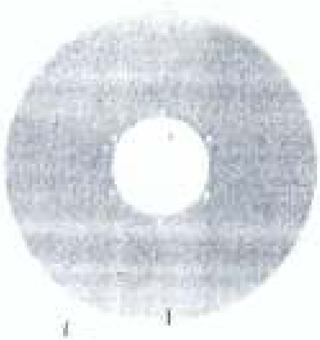

Figure 3.40: Blower Cover

Figure 3.41: Blower Cover Front

Figure 3.42: Blower Flange

Figure 3.43: Blower Middle Elbow

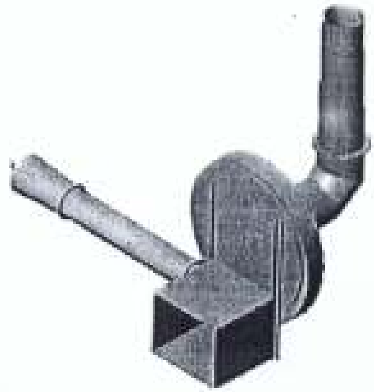

Figure 3.44: Overall Blower-Ducting Assembly

Figure 3.45: Blower Ducting Assembly1

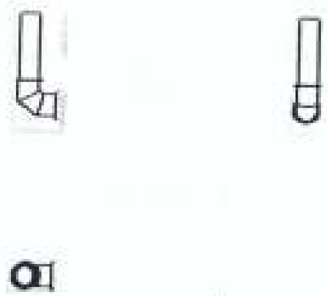

Figure 3.46: Blower Ducting Assembly2

Figure 3.47: Blower Ducting Flange

Figure 3.48: Blower Outlet Duct 1

Figure 3.49: Blower Outlet Duct 1 Assembly

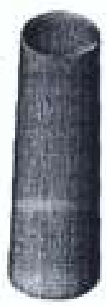

Figure 3.50: Blower Outlet Duct 2

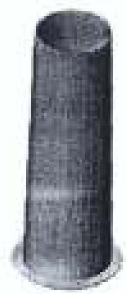

Figure 3.51: Blower Outlet Duct 2 Assembly

3.3.3 GEOMETRIC MODELS DEVELOPED FOR THE BURNER CASING ASSEMBLY

Figures 3.52 to 3.66 represent the various geometric models developed for all the essential units for the burner casing assembly, inside which the combustion of fuel takes place.

Figure 3.52: Burner Back Cover

Figure 3.53: Burner Base Plate

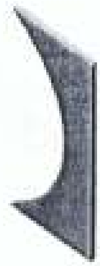

Figure 3.54: Burner Brace

Figure 3.55: Burner Brace Support

Figure 3.56: Burner Casing Duct1

Figure 3.57: Burner Casing Duct2

Figure 3.58: Burner Casing Duct Assembly

Figure 3.59: Burner Casing Duct Flange

Figure 3.60: Burner Sprout

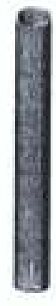

Figure 3.61: Burner Stand Pipe

Figure 3.62: Burner Stand Assembly

Figure 3.63: Burner Front Cover

Figure 3.64: Burner Middle Case

Figure 3.65: Pre Burner Air Casing Assembly

Figure 3.66: Burner Casing Assembly

3.3.4 GEOMETRIC MODELS DEVELOPED FOR THE EXHAUST ELBOW ASSEMBLY

Figures 3.67 to 3.81 represent the various geometric models developed for all the essential units of the exhaust elbow assembly, the top of which the recuperator will be attached.


Figure 3.67: U Channel 1


Figure 3.68: U Channel 2


Figure 3.69: U Channel 3


Figure 3.70: U Channel 4


Figure 3.71: Base Plate1


Figure 3.72: Base Plate2


Figure 3.73: Elbow Flange


Figure 3.74: Exhaust Elbow Base Assembly

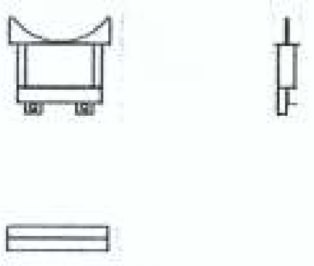
Figure 3.75: Exhaust Elbow Base Assembly2


Figure 3.76: Exhaust Shaft


Figure 3.77: Exhaust Support


Figure 3.78: Exhaust Wheel


Figure 3.79: Exhaust Elbow


Figure 3.80: Exhaust Elbow Assembly1

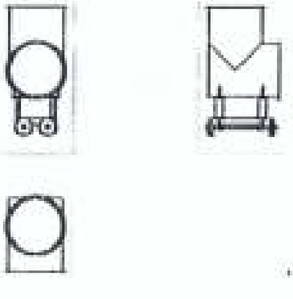
Figure 3.81: Exhaust Elbow Assembly2

3.3.5 GEOMETRIC MODELS DEVELOPED FOR THE RECUPERATOR ASSEMBLY

Figures 3.82 to 3.101 represent the various geometric models developed for all the essential components of the recuperator assembly, the purpose of which is to preheat the air needed for fuel combustion.

Figure 3.82: Blower Support

Figure 3.83: Recuperator Duct

Figure 3.84: Recuperator Stand Leg

Figure 3.85: Recuperator Top Brace1

Figure 3.86: Recuperator Top Brace2

Figure 3.87: Recuperator Top Brace3

Figure 3.88: Recuperator Stand1

Figure 3.89: Recuperator Stand2

Figure 3.90: Blower Outlet Flange

Figure 3.91: Inlet-Outlet Casing

Figure 3.92: Inlet-Outlet Casing Cover

Figure 3.93: Recuperator Barrel

Figure 3.94: Recuperator Sprout

Figure 3.95: Hot Air Elbow Joint Assembly1

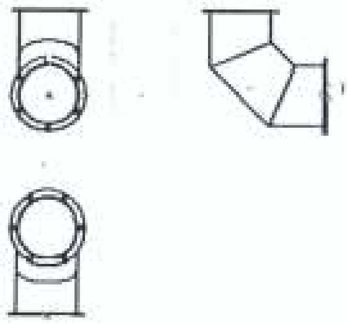

Figure 3.96: Hot Air Elbow Joint Assembly2

Figure 3.97: Recuperator Upper Assembly

Figure 3.98:Recuperator Blower Assembly1

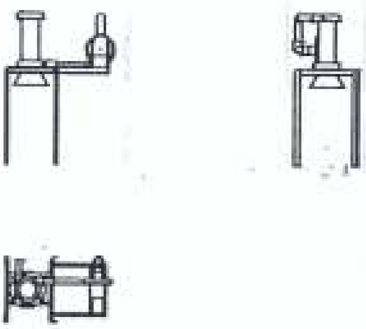

Figure 3.99: Recuperator Blower Assembly2

Figure 3.100: Air Ducting Assembly1

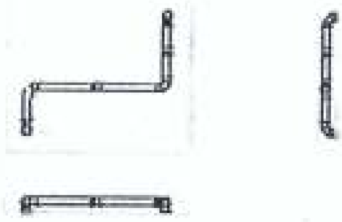

Figure 3.101: Air Ducting Assembly2

3.3.6 GEOMETRIC MODELS DEVELOPED FOR THE OVERALL RECUPERATOR-INCORPORATED ROTARY FURNACE ASSEMBLY

Figures 3.102 to 3.106 represent the various geometric models developed the overall recuperator-incorporated rotary furnace showing different views of interest.

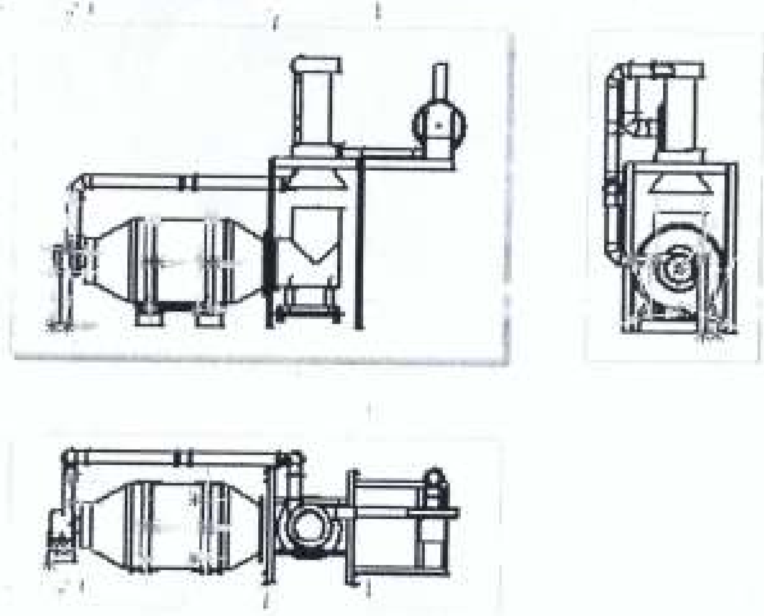

Figure 3.102: Overall Recuperator-Incorporated Rotary Furnace1

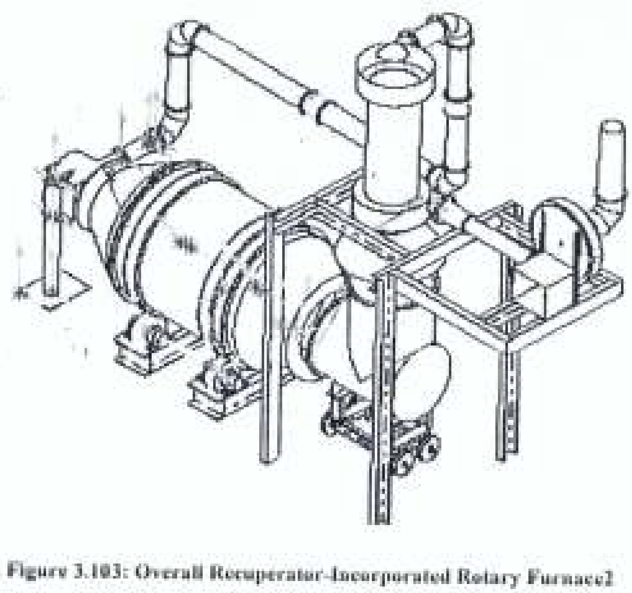

Figure 3.103: Overall Recuperator-Incorporated Rotary Furnace2

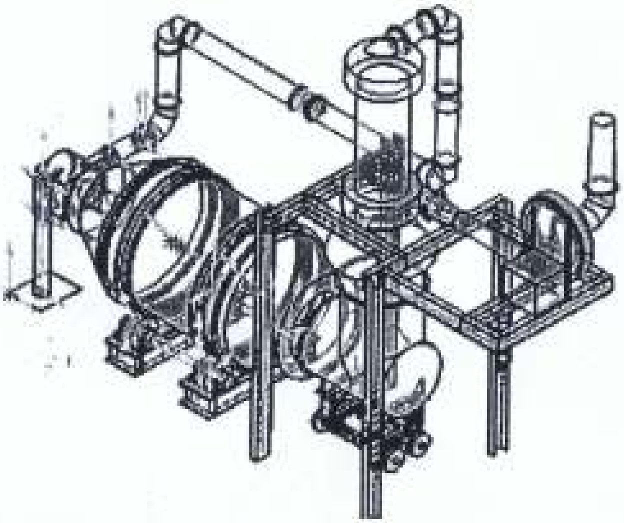

Figure 3.104: Overall Recuperator-Incorporated Rotary Furnace3

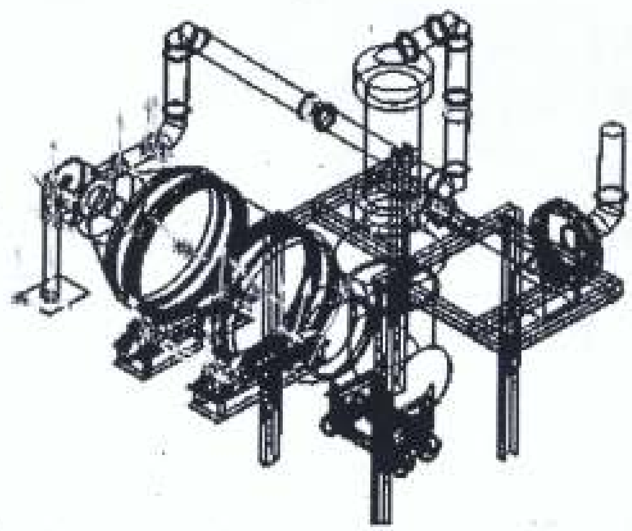

Figure 3.105: Overall Recuperator-Incorporated Rotary Furnace4

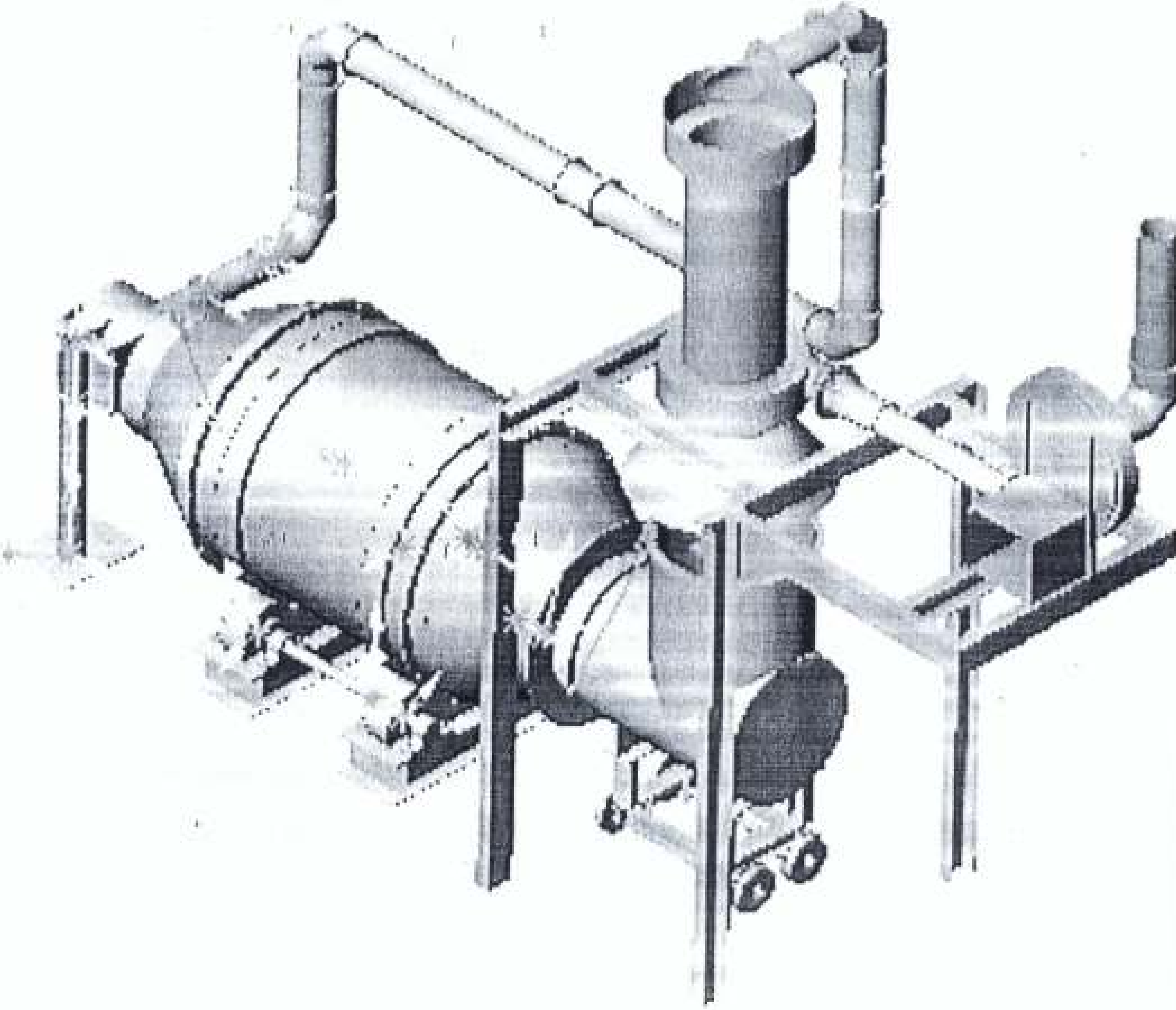

Figure 3.106: Overall Recuperator-Incorporated Rotary Furnace5

3.4 LINING OF THE FURNACE

As earlier stated, the following materials are used for the lining of the furnace:

- Silica Sand
- Kaolin
- Sodium Silicate
- Water

PROCEDURE FOR LINING OF THE FURNACE

The conical (frustum) parts of the furnace, the sprout of the recuperator and the furnace cylindrical parts are all lined. The lining procedure for all is very much similar and follows the pattern outlined by Akinsipe (2004) as enumerated below:

- Prepare mixture of silica sand, kaolin and water for the lining of the part(s);
- Wet the furnace with water and then pour the mixture of kaolin, and sodium silicate and then continue ramming;
- The ramming is done layer by layer round the furnace shell until the desired thickness layer is achieved;
- For the conical and sprout sections, the lining is done with the apex on the floor;
- For the cylindrical section, one-third of the furnace is lined first, and then the shell is rotated and rammed, the remaining two-thirds in two steps of one-third.

CHAPTER FOUR

4.0 RESULTS AND DISCUSSIONS

4.1 PERFORMANCE COMPARISON OF THE RECUPERATOR-INCORPORATED ROTARY FURNACE WITH A ROTARY FURNACE WITHOUT A RECUPERATOR

The performance of a rotary furnace without a recuperator will now be compared with that which is incorporated with a recuperator. The stages involve ascertaining the reliability of the developed models/software and then using the data generated from the reliable software to establish a significant improvement on the performance/efficiency of a recuperator-incorporated rotary furnace over that without a recuperator.

4.2 RELIABILITY OF THE DEVELOPED MODELS

As could be inferred from the preceding sections, the ultimate goal of comparison between a rotary furnace incorporated with a recuperator and the one without a recuperator is to establish a significant improvement in the performance cum efficiency as a result of the incorporation of the recuperator. In doing this, that is comparing different data generated from the various models, it is necessary to first of all establish the fact that the data so generated from the models compare favorably with those actual data obtained from an operating rotary furnace.

There are therefore three stages involved in testing the reliability of the data from the models. The first stage is to obtain actual data from an operating rotary furnace of known capacity. The second stage involves generating data from the developed models (without a recuperator). And, the third and last stage seeks to establish that there is no significant difference between the actual and the generated data using statistical analysis.

4.2.1 ACTUAL DATA FROM AN OPERATING 100KG-CAPACITY ROTARY FURNACE

Performance data in terms of temperature of combustion (bath temperature) from a functional 100kg capacity rotary furnace were obtained over a time interval of 90 minutes from Engineering Materials Development Institute, Akure. The data are as tabulated below:

Table 4.1: Actual Temperatures of combustion of a 100kg capacity rotary furnace

Time (minutes)	Inlet Temp. (^{0}C)	Outlet Temp. (^{0}C)	Average Temp. (^{0}C)
30	1087	1123	1105
40	1161	1177	1169
50	1191	1210	1201
60	1296	1302	1299
70	1383	1399	1391
80	1410	1426	1418
90	1448	1457	1453
100	1453	1464	1459
110	1468	1475	1472
120	1488	1496	1492

Source: Engineering Materials Development Institute, Akure (2004)

4.2.2 GENERATION OF DATA FROM THE DEVELOPED MODEL (WITHOUT A RECUPERATOR)

The temperature variations inside the rotary furnace with time, as established in section 3.2.2, is given by

$$t = t_i + C_h\tau + \frac{C_h R^2}{4a}\left(\frac{r^2}{R^2} - 1\right) \text{ --- } (4.1)$$

The above mathematical model was used to develop a program using C++ programming language which in turn was used to generate the needed data (see Appendix 1). The furnace capacity used is 100kg since this was the same capacity from which actual data to be compared were obtained.

The data generated from the developed program for a 100kg rotary furnace without a recuperator are as shown in the table below:

Table 4.2: Generated Temperatures of combustion of a 100kg capacity rotary furnace

Time (minutes)	Average Temp. (^0C)
30	1082.16
40	1150.5
50	1202.91
60	1290.29
70	1380.31
80	1427.62
90	1460.94
100	1476.81
110	1488.00
120	1517.97

4.2.3 ESTABLISHING THE DEPENDENCE OF THE COMBUSTION TEMPERATURE ON TIME

The dependence of the temperature of combustion on the time taken is done in order to establish that there is actually a correlation between combustion temperature and time, most especially before a steady state is reached. The stages involved are to calculate the coefficient of correlation of temperature on time; calculating the coefficient of determination and doing a test of significance.

- **COEFFICIENT OF CORRELATION**

The coefficient of correlation of average temperature t on time τ is given by

$$r_{t,\tau} = \frac{N\sum t\tau - \sum t \sum \tau}{\sqrt{(N\sum t^2 - (\sum t)^2)(N\sum \tau^2 - (\sum \tau)^2)}} \quad \text{-----------------------------------} (4.2)$$

(Adejuyigbe, 2002 and Grace 2000)

From table 4.1, the parameters needed in the above relation can be calculated thus

Time, τ (hour)	Furnace Temp., t (^{0}C)	$t\tau$	τ^2	T^2
0.50	1105	552.50	0.2500	1221025
0.67	1169	783.23	0.4489	1366561
0.83	1201	996.83	0.6889	1442401
1.00	1299	1299.00	1.0000	1687401
1.17	1391	1627.47	1.3689	1934881
1.33	1418	1885.94	1.7689	2010724
1.50	1453	2179.50	2.2500	2111209
1.67	1459	2436.53	2.7889	2128681
1.83	1472	2693.76	3.3489	2166784
2.00	1492	2984.00	4.0000	2226064
$\sum \tau$ =12.50	$\sum t$ =13459	$\sum t\tau$ =17438.76	$\sum \tau^2$ =17.9134	$\sum t^2$ =18295731

Substituting all the necessary values from the above table into equation (4.2)

$$r_{t,\tau} = \frac{10(17438.76) - (13459)(12.5)}{\sqrt{(10 \times 18295731 - 13459^2)(10 \times 17.9134 - 12.5^2)}}$$

$= 0.955$

- **COEFFICIENT OF DETERMINATION, R^2**

This is the square of the coefficient of correlation. It measures the proportion of total variance in t, which is due to linear association between r and t.

$$R^2 = \frac{\sum(\hat{t}-\bar{t})^2}{\sum(t-\bar{t})^2} = r^2 \quad \text{--} \quad (4.3)$$

where $\hat{t}$ = estimated t

$$\bar{t} = \frac{\sum t}{n} \quad \text{--} \quad (4.4)$$

$$R^2 = r^2 = 0.955^2$$

$$= 0.912$$

This shows that 91.2% of the variations shown by the temperature are as a result of time difference.

4.2.4 TEST OF SIGNIFICANCE

The procedure for testing hypothesis/significance involves six steps. These are to

- Set up the null hypothesis (H_0)
- Set up the alternative hypothesis (H_1)
- Choose a level of significance (α)
- Carry out the test using appropriate test statistic
- Determine the decision rule
- Draw up the conclusion **(Grace, 2000)**

Test of significance shall now be done for two purposes. The first purpose shall be to ascertain a significant association between temperature of combustion and time (reliability of r) while the second purpose is for showing that the developed models are reliable by proving that there is no significant difference between the means of the temperature of combustion from the developed models and the actual ones (obtained from industry)

- **RELIABILITY OF r**

Let the null hypothesis be that there is no correlation between the temperature and time. If ρ is the overall correlation coefficient for the entire population (all obtainable range of temperatures and time),

Null hypothesis $H_0 : \rho = 0;$

Alternative hypothesis $H_1 : \rho \neq 0;$

$$t_c = \frac{r - \rho}{s_r} \quad \text{------------------------} \quad (4.5)$$

where t_c = calculated t
r = sample's coefficient of correlation
ρ = Overall coefficient of correlation

$$s_r \text{ (standard error of r)} = \sqrt{\frac{1 - r^2}{n - 2}} \quad \text{------------------------} \quad (4.6)$$

$$t_c = \frac{r}{\sqrt{\frac{1 - r^2}{n - 2}}} \quad \text{------------------------} \quad (4.7)$$

$$t_c = \frac{0.955}{\sqrt{\frac{1 - 0.955^2}{10 - 2}}} = 9.107$$

Using 5% level of significance, $t_{0.95,8} = 2.306$ (from statistical table for t-distribution)

Since $t_c > t_{0.95,8}$

$\Rightarrow$ Null hypothesis is rejected while alternative hypothesis is accepted. Therefore there is a significant association between temperatures of combustion and time.

- **TEMPERATURES COMPARISON**

Let the temperatures obtained from an operating 100kg rotary furnace (without a recuperator) be represented by t_a and those generated from the developed program by t_g. The two sets of data are as tabulated below

S/N	t_a	t_g	t_a^2	t_g^2
1	1105	1082.16	1221025	1171070.27
2	1169	1150.5	1366561	1323650.25
3	1201	1202.91	1442401	1446992.47
4	1299	1290.29	1687401	1664848.28
5	1391	1380.31	1934881	1905255.7
6	1418	1427.62	2010724	2038098.86
7	1453	1460.94	2111209	2134345.68
8	1459	1476.81	2128681	2180967.78
9	1472	1488	2166784	2214144
10	1492	1517.97	2226064	2304232.92
	$\sum t_a = 13459$	$\sum t_g = 13477.51$	$\sum t_a^2 = 18295731$	$\sum t_g^2 = 18383606.21$

Assuming the population variance $\sigma_1 = \sigma_2$; $n_1 = n_2 = 10$

Null hypothesis H_0: $\mu_1 = \mu_2$

Alternative hypothesis H_1: $\mu_1 \neq \mu_2$

Test statistic is

$$t = \frac{\bar{t}_g - \bar{t}_a}{S\sqrt{\frac{1}{n_1} + \frac{1}{n_2}}} \quad \text{------------------------} \quad (4.8)$$

$$\bar{t}_a = \frac{\sum t_a}{n} = \frac{13459}{10} = 1345.9$$

$$\bar{t}_g = \frac{\sum t_g}{n} = \frac{13477.51}{10} = 1347.75$$

$$S_a^2 = \frac{\sum_{i=1}^{n}(t_{a_i} - \bar{t}_a)^2}{n-1} = \left[\frac{\sum_{i=1}^{n} t_{a_i}^2 - \left(\sum_{i=1}^{n} t_a\right)^2 / n}{n-1}\right] \quad \text{------------------------} \quad (4.9)$$

$$S_a^2 = \frac{18295731 - {13459^2}/{10}}{10-1} = 20140.32$$

$$S_g^2 = \frac{\sum_{i=1}^{n}(t_{g_i} - \bar{t}_g)^2}{n-1} = [\frac{\sum_{i=1}^{n} t_{g_i}^2 - (\sum_{i=1}^{n} t_g)^2 / n}{n-1}] \quad \text{-- (4.10)}$$

$$S_g^2 = \frac{18383606.21 - {13477.51^2}/{10}}{10-1} = 24364.29$$

$$S(\text{pooled variance}) = \sqrt{\frac{(n_1-1)S_a^2 + (n_2-1)S_g^2}{n_1+n_2-2}} \quad \text{-- (4.11)}$$

$$S(\text{pooled variance}) = \sqrt{\frac{(10-1)20140.32 + (10-1)24364.29}{10+10-2}} = 149.17$$

$$t = \frac{1347.45 - 1345.9}{149.17\sqrt{\frac{1}{10} + \frac{1}{10}}} = 2.323$$

At 1% level of significance, tabulated $t_{18,\,0.01} = 2.878$. Since the calculated t is less than the tabulated t, it shows that there is no significant difference between the two sets of temperatures. This implies that the values (data) generated from the developed program are reliable.

Graphs of the temperatures of combustion when plotted against time give the pattern shown in fig. 4.1 below

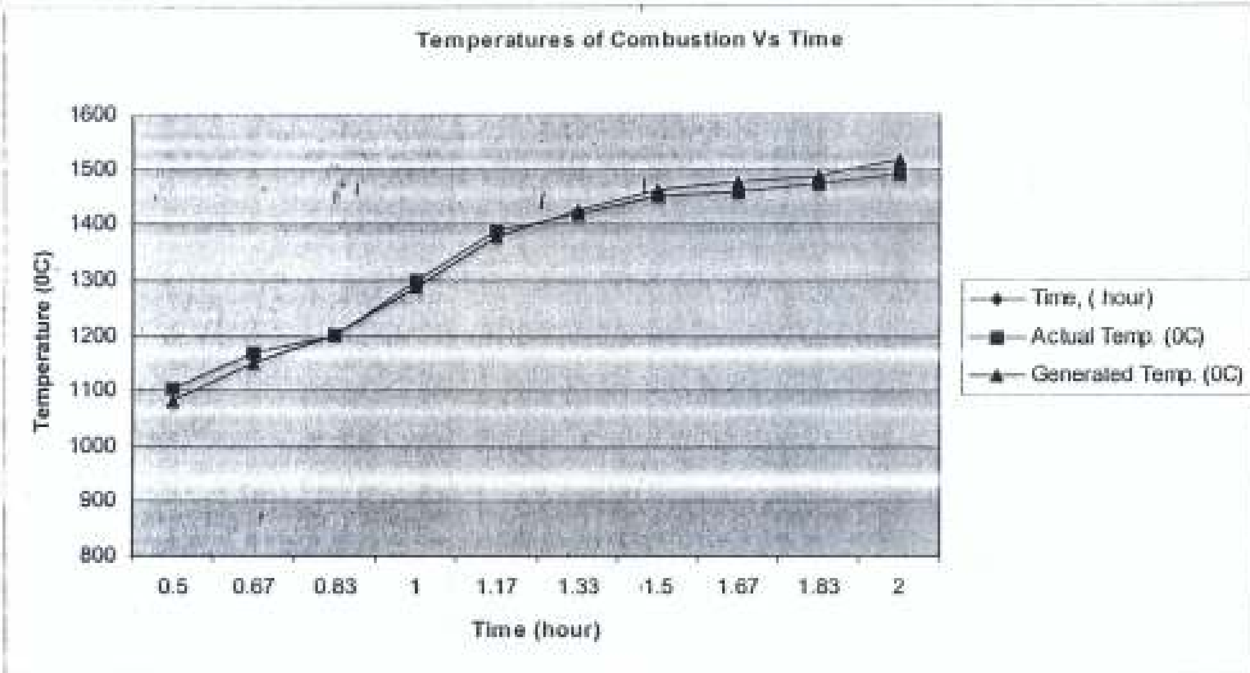

Figure4.1: Graphs of Combustion Temperatures (Actual and Generated) Vs Time

4.3 EFFECT OF WASTE HEAT RECUPERATION ON COMBUSTION TEMPERATURES

Now that the reliability of the developed program has been ascertained, data generated from it can now be reliably used. It is of interest here to compare the temperatures of combustion when a recuperator is incorporated in the design and when there is none. The temperatures generated for these two situations are as shown in table 4.3 below. The set of temperatures generated for an ordinary rotary furnace without a recuperator is represented by $\mathbf{t_o}$ while those generated for a rotary furnace incorporated with a recuperator is designated by $\mathbf{t_r}$.

Table 4.3: Temperatures of combustion of a 100kg capacity rotary furnace with and without a recuperator

Time, τ (hour)	Average Temp. t_a (^0C)	Average Temp. t_r (^0C)
0.50	1082.16	1222.84
0.67	1150.5	1372.92
0.83	1202.91	1451.6
1.00	1290.29	1564.97
1.17	1380.31	1627.79
1.33	1427.62	1681.32
1.50	1460.94	1726.14
1.67	1476.81	1737.97
1.83	1488.00	1749.76
2.00	1517.97	1754.48

- **COMPARISON OF THE TEMPERATURES USING t-TEST**

The two sets of temperatures generated from the developed program, having established the reliability of the program, shall now be compared using t-test to ascertain a significant improvement on combustion temperature as a result of waste heat recuperation. Table 4.3 can then be modified to accommodate other parameters needed in the analysis. The test of significance that shall be done will be similar to the process followed in section 4.3.4. This will involve setting of null and alternative hypotheses, choosing a level of significance, and interpreting the results by comparing the calculated t with the tabulated one (from the student t-table)

S/N	t_o	t_r	t_o^2	t_r^2
1	1082.16	1222.84	1171070	1495338
2	1150.5	1372.92	1323650	1884909
3	1202.91	1451.6	1446992	2107143
4	1290.29	1564.97	1664848	2449131
5	1380.31	1627.79	1905256	2649700
6	1427.62	1681.32	2038099	2826837
7	1460.94	1726.14	2134346	2979559
8	1476.81	1737.97	2180968	3020540
9	1488.00	1749.76	2214144	3061660
10	1517.97	1754.48	2304233	3078200
	$\sum t_o = 13477.51$	$\sum t_r = 15889.79$	$\sum t_o^2 = 18383606$	$\sum t_r^2 = 25553017$

sing sets of equations 4.8 to 4.11 and similar approach, we have

e population variance $\sigma_1 = \sigma_2$; $n_1 = n_2 = 10$

Null hypothesis $H_0: \mu_1 = \mu_2$

Alternative hypothesis $H_1: \mu_2 > \mu_1$

est statistic is

$$t = \frac{\bar{t}_r - \bar{t}_o}{S\sqrt{\frac{1}{n_1} + \frac{1}{n_2}}}$$

$$\bar{t}_o = \frac{\sum t_o}{n} = \frac{13477.51}{10} = 1347.751$$

$$\bar{t}_r = \frac{\sum t_r}{n} = \frac{15889.79}{10} = 1588.979$$

$$S_o^2 = \frac{\sum_{i=1}^{n}(t_{o_i} - \bar{t}_o)^2}{n-1} = [\frac{\sum_{i=1}^{n} t_{o_i}^2 - (\sum_{i=1}^{n} t_o)^2 / n}{n-1}]$$

$$S_o^2 = \frac{18383606 - 13477.51^2/10}{10-1} = 24364.27$$

$$S_r^2 = \frac{\sum_{i=1}^{n}(t_{r_i} - \bar{t}_r)^2}{n-1} = [\frac{\sum_{i=1}^{n} t_{r_i}^2 - (\sum_{i=1}^{n} t_r)^2 / n}{n-1}]$$

$$S_r^2 = \frac{25553017 - 15889.79^2/10}{10-1} = 33830.49$$

$$\text{S(pooled variance)} \pm \sqrt{\frac{(n_1-1)S_o^2 + (n_2-1)S_r^2}{n_1+n_2-2}}$$

$$\text{S(pooled variance)} = \sqrt{\frac{(10-1)24364.27 + (10-1)33830.49}{10+10-2}} = 170.58$$

$$t = \frac{1588.979 - 1347.751}{170.58\sqrt{\frac{1}{10}+\frac{1}{10}}} = 3.162$$

t 1% level of significance, tabulated $t_{18,\,0.01}$ is still 2.878. Since the calculated t is now greater ian the tabulated t, it shows that there is a significant improvement in the temperature of ›mbustion for a recuperator-incorporated rotary furnace over an ordinary furnace without a ecuperator.

Graphs of the temperatures of combustion (of the two situations) when plotted against time ive the pattern shown in figure4.2 below

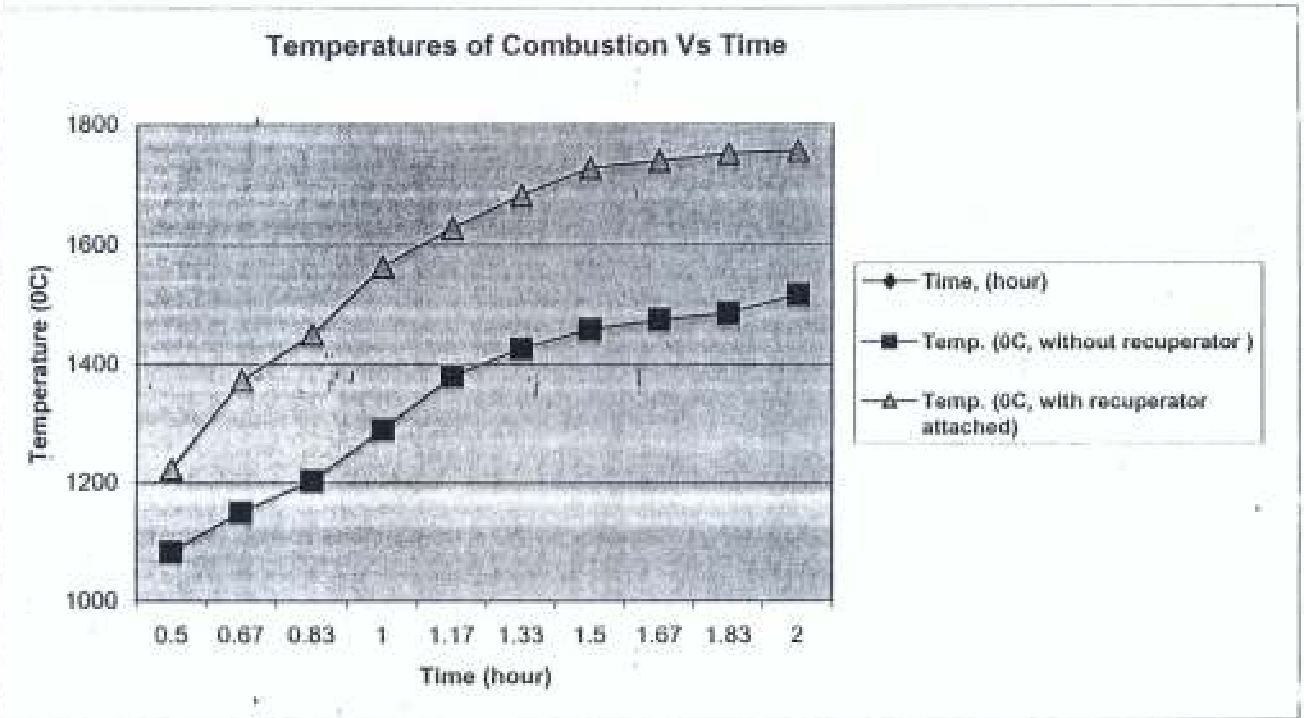

Figure4.2: Graphs of Combustion Temperatures (Generated) Vs Time

.4 GENERAL RESULTS AND DISCUSSION

'he general results obtained from this project work are divided into two categories. The first ategory is the quantitative resultant effect of recuperator-incorporation on the design of the rotary urnace system on the coefficient of heat utilization and on the rate of fuel consumption; while the econd category deals with the results obtained from the developed software for the design of the ecuperator-incorporated rotary furnace.

.4.1 INCREASE IN COEFFICIENT OF HEAT UTILIZATION

'here are two forms of coefficient of heat utilization. They are the coefficient of useful heat tilization (η_{uhu}) and coefficient of total heat utilization (η_{hu})

$$\eta_{uhu} = \frac{Q_{ch} + Q_{ph} - Q_{wg} - Q_{lo}}{Q_{ch} + Q_{ph}} \quad \text{.......} (4.12)$$

$$\eta_{hu} = \frac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}} \text{ -- } (4.13)$$

For the purpose of finding the effect of recuperator incorporation on the coefficient of heat utilization in the rotary furnace, the *coefficient of total heat utilization* is used.

Without recuperator incorporation, Q_{ph} in equation 4.13 is zero;

$$\eta'_{hu} = \frac{Q_{ch} - Q_{wg}}{Q_{ch}}$$

Using fuel consumption rate of 25litres/h ($= \frac{25 \times 10^{-3}}{10.44} = 2.395 \times 10^{-3}$ kg/h)

$$Q_{ch} = BQ'_w = 2.395 \times 10^{-3} \times 44000 \times 10^3 = 105.38 \times 10^3 \text{ J/h}$$

$$Q_{wg} = Bv_{wg}c_{wg}t_{wg} = 2.395 \times 10^{-3} \times 12.324 \times 1493.1 \times 1050 = 46.27 \times 10^3 \text{ J/h}$$

$$\eta'_{hu} = \frac{105.38 \times 10^3 - 46.27 \times 10^3}{105.38 \times 10^3} = 0.5609$$

With recuperator incorporation,

$$\eta_{hu} = \frac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}}$$

Using the same fuel consumption rate of 25litres/h ($= \frac{25 \times 10^{-3}}{10.44} = 2.395 \times 10^{-3}$ kg/h)

$$Q_{ch} = BQ'_w = 2.395 \times 10^{-3} \times 44000 \times 10^3 = 105.38 \times 10^3 \text{ J/h}$$

$$Q_{wg} = Bv_{wg}c_{wg}t_{wg} = 2.395 \times 10^{-3} \times 12.324 \times 1493.1 \times 693.44 = 30.56 \times 10^3 \text{ J/h}$$

$$Q_{ph} = Bc_at_anv_a = 2.395 \times 10^{-3} \times 1337.15 \times 450 \times 1.2 \times 11.481 = 19.86 \times 10^3 \text{ J/h}$$

$$\eta_{hu} = \frac{105.38 \times 10^3 + 19.86 \times 10^3 - 30.56 \times 10^3}{105.38 \times 10^3 + 19.86 \times 10^3} = 0.7560$$

The percentage increase in the coefficient of heat utilization is

$$\%Increase\,\eta_{hu} = \frac{\eta_{hu} - \eta_{hu}^{/}}{\eta_{hu}^{/}} \times 100\% = \frac{0.7560 - 0.5609}{0.5609} \times 100\% = 34.78\%$$

This signifies a significant improvement on the thermal efficiency of the rotary furnace system as a result of the recuperator incorporation.

4.4.2 REDUCTION IN FUEL CONSUMPTION

From equation 4.13, $\eta_{hu} = \dfrac{Q_{ch} + Q_{ph} - Q_{wg}}{Q_{ch} + Q_{ph}}$

Substituting $\eta_{hu} = 0.5609$

$$Q_{ch} = B^{/} Q_{w}^{l} = B^{/} \times 44000 \times 10^{3} = 44 \times 10^{6} B^{/} \text{ J/h}$$

$$Q_{wg} = 30.56 \times 10^{3} \text{ J/h}$$

$$Q_{ph} = B^{/} c_{a} t_{a} n v_{a} = B^{/} \times 1337.15 \times 450 \times 1.2 \times 11.481 = 8.29 \times 10^{6} B^{/} \text{ J/h}$$

$$0.5609 = \frac{44 \times 10^{6} B^{/} + 8.29 \times 10^{6} B^{/} - 30.56 \times 10^{3}}{44 \times 10^{6} B^{/} + 8.29 \times 10^{6} B^{/}} = \frac{52.29 \times 10^{6} B^{/} - 30.56 \times 10^{3}}{52.29 \times 10^{6} B^{/}}$$

From where $B^{/} = \dfrac{30.56 \times 10^{3}}{22.96 \times 10^{6}} = 1.331 \times 10^{-3}$ kg/h $= 1.331 \times 10^{-3} \times 10.44 \times 10^{3} = 13.90$litres/h

$$\%FuelConsumption\,\text{Re}\,duction = \frac{B - B^{/}}{B} \times 100\% = \frac{25 - 13.9}{25} \times 100\% = 44.4\%$$

This also signifies a significant reduction on the fuel consumption rate of the rotary furnace system as a result of the recuperator incorporation, thereby reducing the operational cost.

4.4.3 OUTPUTS FROM THE DEVELOPED SOFTWARE

The reliability of the developed program has been ascertained. All the basic parameters (dimensions and combustion temperatures) needed for the design of a rotary furnace (with and without a recuperator incorporation) are generated from the program. When the results are compared, it then clearly shows how significant the incorporation of recuperator could have on the performance of the eptire rotary-furnace setup. The program can find such parameters for a very wide range of capacities. There are six possible options to choose from, when the program is run depending on the desire of the user. Each of the options, with the raw results obtained from the program when run, is as shown in Appendix B.

CHAPTER FIVE

5.0 CONCLUSIONS AND RECOMMENDATIONS

5.1 CONCLUSIONS

Incorporation of recuperative heat exchanger into the design of a rotary furnace is very essential. A major reason being that a unit of physical heat recovered from waste gases and returned back into the furnace with air is much more useful than a corresponding unit of chemical heat produced in the furnace through combustion of fuel. Advantages of incorporation of recuperator into the design are

- Appreciable increase (34.78%) in the furnace thermal efficiency
- Considerable increase in the temperature of combustion (33.33%). This makes it possible to adapt the furnace for steel casting.
- Significant reduction (44.4%) in the fuel combustion thereby reducing operating cost and
- Aiding complete combustion.

The overall efficiency of the furnace system was shown to have increased considerably with the incorporation of a recuperator through the following procedure

- Necessary mathematical and geometric models are developed
- The developed models were used to develop a program using C++ programming language
- The reliability of the program developed was tested by
    - obtaining a set of actual data for a known capacity rotary furnace (without a recuperator) from the industry;
    - generating a corresponding set of data from the developed program;

- ascertaining the reliability of the developed program by establishing that there is no significant difference between the two set of data using the necessary statistical methods

- Establishing a significant increase in the overall efficiency of a recuperator-incorporated rotary furnace over the one without a recuperator using the data generated from the "now-reliable" program.

5.2 RECOMMENDATIONS

The use of a recuperator in the operation of a rotary furnace significantly increases the temperature of combustion. As such, the life span of the insulating refractory used with the furnace (when there is no recuperator) will reduce significantly. Subsequently, the number of melts before relining of the furnace will reduce because the temperature of combustion will now be very close (if not higher than) to the softening temperature of the refractory.

It is therefore recommended that a new refractory should be selected/developed, which has higher value of

- Softening Temperature
- Refractoriness
- Ultimate Compression Strength and
- Spalling Resistance

Also, the following possible sources of failure in the recuperator with the appropriate remedies should be taken cognizance of.

1) **Leakage of Air :** There are some units of the recuperator that are bolted together. Under the influence of repeated heating and cooling, these bolts tend to loosen, and

leakage may occur. Periodical checking of the bolted units is thus essential in order to re-tighten any loosening bolt. Also, usage of lock nuts can minimize such leakages.

2) **Cracks :** The main cause of cracking is jamming. This is a situation whereby free expansion is prevented somewhere in the system. This may be as a result of severe overheating causing the maximum allowable expansion to be exceeded;

3) **Corrosion :** There are three main forms of possible corrosion:

   a) **Oxidation :** Oxygen, water vapour, or carbon dioxide in the waste gases will cause oxidation to any part of the recuperator at a temperature of 500^{o}C and above. Although the alloys used could minimize the oxidation up to about 1100^{o}C, there will still be scaling and subsequent need to replace the hottest parts.

   b) **Sulphur Attack :** At the hot end of the recuperator, if the gases contain more than about 2% of Oxygen, and in the absence of vanadium and alkali oxides, the sulphur in the waste gases (in the form of SO_2 or SO_3) is a little more corrosive than CO_2 or H_2. Usage of low alloy steel could provide solution for this.

   c) **Carbon Monoxide :** There is danger of metal corrosion where the metal temperature is high and the waste gases contain large amount of carbon monoxide. The mechanism of this form of corrosion is that the carbon monoxide reduces the protective oxide film to spongy metal, which allows further oxidation of the underlying metal as soon as circumstances permit.

4) Blockage : This occurs in a situation where the waste gases are very dirty. Since the shell's diameter must be as small as possible for higher heat transfer coefficient, this on the other hand aids blockage from the dirt in the waste gases.

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APPENDIX A

```
#include<iostream.h>
#include<math.h>
#include<conio.h>
#include<fstream.h>
double Qch,Qa,Ta,Qf,tf,Tf,Qex,M,Qi,Q1,Q2,t1,Q3,Q4,Q5,r1,r2,D2,D3;
double T2,y,Q6,Q7,Q8,Qu,Qt,U,K1,K2,K3,tL,SA,SR,El,L,Ico2,In2,Io2,Ih2o,I0;
const double PI=3.142,K1=25.6,B=23.25,t=3.0,Qw=44000.00,Ca=1.455,n=1.2,Va=11.481;
const double Cm=0.5,To=295,Tm=1403,T1=1667,Lm=175.84,a=0.015,F=0.0452,tc=0.005;
const double twg=800,Cf=1.491,Ec=210e+9,Cwg=1.5836,Vwg=12.324,Co=5.768;
const double CO2=0.10686,N2=0.74429,O2=0.033,H2O=0.11576;
char response,response1,response2,response3;
int main(int argc, char*argv[])
{
        ofstream fout("result7");
        double SA=0.0877;SR=3e+9;double El=380e+9;
esc1:{cout<<"This program evaluates the essential parameters needed to design \n";
         cout<<"a recuperator-incorporated rotary furnace. The various options \n";
         cout<<"available to choose from are as outlined below, depending on \n";
         cout<<"what the user is interested in.\n";
         cout<<endl;
        cout<<"What parameters are you interested in?\n";
        cout<<endl;
        cout<<"\t\t(a) A rotary furnace without a recuperator? Enter 1\n";
        cout<<"\n\t\t(b) A rotary furnace with a recuperator? Enter 2\n";
        cout<<"\n\t\t(c) Temperatures of combustion of a rotary furnace \n";
        cout<<"\n\t\t    without a recuperator? Enter 3 \n";
        cout<<"\n\t\t(d) Temperatures of combustion of a rotary furnace \n";
        cout<<"\n\t\t    with a recuperator? Enter 4 \n";
        cout<<"\n\t\t(e) Interested in both (a) and (c)? enter 5\n";
        cout<<"\n\t\t(f) Interested in both (b) and (d)? enter 6\n";
        cout<<endl;
        cout<<"Make your choice by picking the appropriate number against the chosen option: ";
        cin>>response;
        cout<<endl;
        fout<<"This program evaluates the essential parameters needed to design \n";
        fout<<"a recuperator-incorporated rotary furnace. The various options \n";
        fout<<"available to choose from are as outlined below, depending on \n";
        fout<<"what the user is interested in.\n";
        fout<<endl;
        fout<<"What parameters are you interested in?\n";
        fout<<endl;
        fout<<"\t\t(a) A rotary furnace without a recuperator? Enter 1\n";
        fout<<"\n\t\t(b) A rotary furnace with a recuperator? Enter 2\n";
        fout<<"\n\t\t(c) Temperatures of combustion of a rotary furnace \n";
```

```
        fout<<"\n\t\t ,  without a recuperator? Enter 3 \n";
        fout<<"\n\t\t(d) Temperatures of combustion of a rotary furnace \n";
        fout<<"\n\t\t    with a recuperator? Enter 4 \n";
        fout<<"\n\t\t(e) Interested in both (a) and (c)? enter 5\n";
        fout<<"\n\t\t(f) Interested in both (b) and (d)? enter 6\n";
        fout<<endl;
        fout<<"Make your choice by picking the appropriate number against the chosen option: ";
        fout<<endl;
        if((response=='1'||response=='2'||response=='3')||(response=='4'||response=='5'||response=='
6'))
                goto esc;

        if((response!='1'||response!='2'||response!='3')||(response!='4'||response!='5'||response!='6'))
        {cout<<"YOU HAVE NOT MADE A VALID CHOICE!\n";
         cout<<"TRY TO MAKE A CORRECT CHOICE NEXT TIME YOU RUN THE
PROGRAM\n";
         fout<<"YOU HAVE NOT MADE A VALID CHOICE!\n";
         fout<<"TRY TO MAKE A CORRECT CHOICE NEXT TIME YOU RUN THE
PROGRAM\n";
         goto esc2;
        }

esc: {cout<<"\nEnter the capacity of the furnace, that is the maximum quantity of ";
        cout<<"\ncharges it is intended for (kg): ";
        fout<<"\nEnter the capacity of the furnace, that is the maximum quantity of ";
        fout<<"\n charges it is intended for (kg): 400";
        cin>>M;
        Qch=B*t*Qw;
        Qex=56.52*M*t;
        cout<<"\nDo you wish to supply the following parameters for your lining material";
        cout<<"\n(recommended data will be used otherwise): shrinkage allowance,";
        cout<<"\nCompressive strength and Young's Modulus? (Y/N)";
        cin>>response1;
        fout<<"\nDo you wish to supply the following parameters for your lining material";
        fout<<"\n(recommended data will be used otherwise): shrinkage allowance,";
        fout<<"\nCompressive strength and Young's Modulus? (Y/N) N";
        switch(response1)
        {case 'y':{cout<<"Supply the following parameters for the lining : "<<endl;
        cout<<"\t Shrinkage allowance  : ";
        cin>>SA;
        cout<<"\t Compressive strength (N/m2) : ";
        cin>>SR;
        cout<<"\t Young's Modulus (N/m2) : ";
        cin>>El;
        fout<<"Supply the following parameters for the lining : "<<endl;
        fout<<"\t Shrinkage allowance  : ";
```

```
fout<<"\t Compressive strength (N/m2) : ";
fout<<"\t Young's Modulus (N/m2) : ";} break;
case 'Y':{cout<<"Supply the following parameters for the lining : "<<endl;
cout<<"\t Shrinkage allowance  : ";
cin>>SA;
cout<<"t Compressive strength (N/m2) : ";
cin>>SR;
cout<<"t Young's Modulus (N/m2) : ";
cin>>El;
fout<<"Supply the following parameters for the lining : "<<endl;
fout<<"\t Shrinkage allowance  : ";
fout<<"\t Compressive strength (N/m2) : ";
fout<<"\t Young's Modulus (N/m2) : ";
} break;
default: {double SA=0.0877;
         SR=3e+9;
         double El=380e+9;};break;};
K1=SA*tc*Ec*El/SR;
Qi=Qch+Qex;
getch();
Q1=M*(Cm*(Tm-To)+Lm);
Q2=0.025*Cwg*t*Vwg*800;
Q3=B*Vwg*a*t*12142;
Q4=0.01*Qch;
L=(M>=100?(1150+0.875*(M-100)):1150);
y=0.8*PI*L*Kl*T1*3.6/1000;
Q6=Co*pow((T1/100),4)*F*t*0.67;
Q7=0.15*Qi;
Q8=0.2*Qi;
Qu=Q3+Q4+Q6+Q7+Q8;
Qt=Q1+Q2+Qu;
Q5=(Qi-Qt-0.2*Qu)/1.2;
U=(Q5-y)/(Q5+y);
K2=tc*Ec;
K3=sqrt((K2*K2)+4*K1*(El-U*El));
r2=(K3-K2)/(2*(El-U*El));
r1=U*r2;
tL=r2-r1;
D2=2*r2;
if(response=='1')
{D3=(M>=100?(D2+(M-75)/1000):(D2-(100-M)*2.5/1000));
cout<<endl;
cout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
cout<<"\n\tThickness of the lining is "<<(M>=100?(tL*1000):(tL*1000-0.7405*(100-
M)))<<"mm";
```

```
	cout<<"\n\tLength of the cylindrical shell is "<<(M>=100?(925+(M-100)/3):(925-(100-
M)*7))<<"mm";
	cout<<"\n\tSlant length of each of the two conical frustums is
"<<(M>=100?(250+0.875*(M-100)):(250+0.875*(M-100)-40))<<"mm"<<endl;
	cout<<"\n\tBase diameter of the conical frustum is "<<(4*D3/9)*1000<<"mm";
	cout<<"\n\tDiameter of the burner-end of the furnace flange is
"<<(3*D3/11)*1000<<"mm";
	cout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3*D3/10)*1000<<"mm";
	cout<<endl;
	fout<<endl;
	fout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
	fout<<"\n\tThickness of the lining is "<<(M>=100?(tL*1000):(tL*1000-0.7405*(100-
M)))<<"mm";
	fout<<"\n\tLength of the cylindrical shell is "<<(M>=100?(925+(M-100)/3):(925-(100-
M)*7))<<"mm";
	fout<<"\n\tSlant length of each of the two conical frustums is
"<<(M>=100?(250+0.875*(M-100)):(250+0.875*(M-100)-40))<<"mm"<<endl;
	fout<<"\n\tBase diameter of the conical frustum is "<<(4*D3/9)*1000<<"mm";
	fout<<"\n\tDiameter of the burner-end of the furnace flange is
"<<(3*D3/11)*1000<<"mm";
	fout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3*D3/10)*1000<<"mm";
	fout<<endl;};
	if(response=='2')
	{D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
	cout<<endl;
	cout<<"For the Furnace design parameters :  \n";
	cout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
	double tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
	cout<<"\n\tThickness of the lining is "<<tL2<<"mm";
	double L2=(M>=100?(925+1.1*(M-100)/3):(925-0.9*(100-M)*7));
		L2=(M==100?(925+1.1*M/3):L2);
	cout<<"\n\tLength of the cylindrical shell is "<<L2<<"mm";
	cout<<"\n\tSlant length of each of the two conical frustums is "<<(M>=100?(250+0.9*(M-
100)):(250+0.9*(M-100)-40))<<"mm";
	cout<<"\n\tBase diameter of the conical frustum is "<<(4000*D3/9)<<"mm";
	cout<<"\n\tDiameter of the burner-end of the furnace flange is "<<(3000*D3/11)<<"mm";
	cout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3000*D3/10)<<"mm\n";
	cout<<"\nFor the Recuperator design parameters :  \n";
	cout<<"\n\tDiameter of the inner cylinder of the recuperator is "<<(M<=300?(5000*D3/9-
28.85):(5000*D3/9-115))<<"mm";
	cout<<"\n\tDiameter of the outer cylinder of the recuperator is
"<<(M<=300?(5000*D3/9+tL2/5-4.314):(5000*D3/9+tL2/5-25))<<"mm";
```

```
cout<<"\n\tLength  of the inner cylinder of the recuperator is "<<(M<=300?(1.750*L2-94.92):(1.750*L2+453.08))<<"mm";
cout<<"\n\tLength  of the outer cylinder of the recuperator is "<<(M<=300?(1.750*L2-3000*D3/5+200.08):(1.750*L2-3000*D3/5+973.08))<<"mm";
cout<<"\n\tRecuperator sprout's base diameter is "<<(750*D3-1.447)<<"mm";
cout<<"\n\tRecuperator sprout's slant length is "<<(750*D3-6.447)<<"mm";
cout<<endl;
fout<<endl;
fout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
fout<<"\n\tThickness of the lining is "<<tL2<<"mm";
fout<<"\n\tLength of the cylindrical shell is "<<L2<<"mm";
fout<<"\n\tSlant length of each of the two conical frustums is "<<(M>=100?(250+0.9*(M-100)):(250+0.9*(M-100)-40))<<"mm"<<endl;
fout<<"\n\tBase diameter of the conical frustum is "<<(4000*D3/9)<<"mm";
fout<<"\n\tDiameter of the burner-end of the furnace flange is "<<(3000*D3/11)<<"mm";
fout<<"\n\tDiameter of the charging-end of the furnace flange is "<<(M<=300?(3*(5000*D3/9-25)):(3*(5000*D3/9-115)))<<"mm\n";
fout<<"\n\tDiameter of the inner cylinder of the recuperator is "<<(M<=300?(5000*D3/9-28.85):(5000*D3/9-115))<<"mm";
fout<<"\n\tDiameter of the outer cylinder of the recuperator is "<<(M<=300?(5000*D3/9+tL2/5-4.314):(5000*D3/9+tL2/5-25))<<"mm";
fout<<"\n\tLength  of the inner cylinder of the recuperator is "<<(M<=300?(1.750*L2-94.92):(1.750*L2+320))<<"mm";
fout<<"\n\tLength  of the outer cylinder of the recuperator is "<<(M<=300?(1.750*L2-3000*D3/5):(1.750*L2-3000*D3/5+790))<<"mm";
fout<<"\n\tRecuperator sprout's base diameter is "<<(750*D3-1.447)<<"mm";
fout<<"\n\tRecuperator sprout's slant length is "<<(750*D3-6.447)<<"mm";
fout<<endl;};
if(response=='3')
{D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
double tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
double t1,t2,t3,t4,t5,t6,t7,t8,t9,t10;
double l1,l2,l3,l4,l5,l6,l7,l8,l9,l10;
double c1,c2,c3,c4,c5,c6,c7,c8,c9,c10;
double a1,a2,a3,a4,a5,a6,a7,a8,a9,a10;
double r=D3/2,R=r+0.005;
const double ch=4017.5,t0=37.0,e=1850.0;
r=(M>=100?(r-0.000965):r);
R=(M>=100?(R-0.003465):R);
l1=0.835+0.00058*t0;
c1=0.88+0.00023*t0;
a1=l1/(e*c1);
t1=t0+(0.5*ch)+(ch*R*R)*((r*r)/(R*R)-1)/(40*a1);
t1=t1-t1/2.5;
t1=(M>=450?(t1+10):t1);
l2=0.835+0.00058*t1;
```

```
c2=0.88+0.00023*t1;
a2=l2/(e*c2);
t2=t1+0.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a2);
t2=t2-t2/2.4;
t2=(M>150?(t2+150):t2);
if(M>=230) t2=t2+210;
t2=(M>=350?(t2+200):t2);
t2=(M>=450?(t2+70):t2);
t2=(M>500?(t2+300):t2);
l3=0.835+0.00058*t2;
c3=0.88+0.00023*t2;
a3=l3/(e*c3);
t3=t2+0.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a3);
t3=t3-t3/1.8;
t3=(M>150?(t3+150):t3);
if(M>=230) t3=t3+130;
t3=(M>=350?(t3+200):t3);
t3=(M>450?(t3+130):t3);
t3=(M>500?(t3+150):t3);
l4=0.835+0.00058*t3;
c4=0.88+0.00023*t3;
a4=l4/(e*c4);
t4=t3+1.0*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a4);
t4=t4-t4/1.595;
t4=(M>150?(t4+150):t4);
t4=(M>=350?(t4+200):t4);
t4=(M>450?(t4+80):t4);
l5=0.835+0.00058*t4;
c5=0.88+0.00023*t4;
a5=l5/(e*c5);
t5=t4+1.17*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a5);
t5=t5-t5/1.48;
t5=(M>150?(t5+70):t5);
if(M>=230) t5=t5+50;
t5=(M>=350?(t5+140):t5);
t5=(M>=450?(t5+70):t5);
l6=0.835+0.00058*t5;
c6=0.88+0.00023*t5;
a6=l6/(e*c6);
t6=t5+1.33*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a6);
t6=t6-t6/1.398;
t6=(M>150?(t6+65):t6);
if(M>=230) t6=t6+80;
t6=(M>=350?(t6+80):t6);
t6=(M>=450?(t6+70):t6);
l7=0.835+0.00058*t6;
```

```
c7=0.88+0.00023*t6;
a7=l7/(e*c7);
t7=t6+1.5*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a7);
t7=t7-t7/1.34;
t7=(M>150?(t7+M-150):t7);
t7=(M>=350?(t7-60):t7);
l8=0.835+0.00058*t7;
c8=0.88+0.00023*t7;
a8=l8/(e*c8);
t8=t7+1.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a8);
t8=t8-t8/1.295;
t8=(M>=150?(t8+M/3+10):t8);
if(M>=230) t8=t8+30;
t8=(M>=350?(t8+28):t8);
t8=(M>=450?(t8+30):t8);
l9=0.835+0.00058*t8;
c9=0.88+0.00023*t8;
a9=l9/(e*c9);
t9=t8+1.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a9);
t9=t9-t9/1.263;
t9=(M>=150?(t9+M/3):t9);
if(M>=230) t9=t9+20;
t9=(M>=350?(t9+27):t9);
t9=(M>=450?(t9+22):t9);
l10=0.835+0.00058*t9;
c10=0.88+0.00023*t9;
a10=l10/(e*c10);
t10=t9+2*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a10);
t10=t10-t10/1.24;
t10=(M>=150?(t10+M/3-30):t10);
if(M>=230) t10=t10+20;
t10=(M>=350?(t10+20):t10);
t10=(M>=450?(t10+20):t10);
cout<<endl;
cout<<"The required temperatures of combustion for a "<<M<<"kg";
cout<<"\ncapacity rotary furnace (without a recuperator) are\n";
cout<<"\n\t\t  Time (hr)                          Temperature (0C) ";
cout<<"\n\t\t  0.50                                 "<<t1;
cout<<"\n\t\t  0.67                                 "<<t2;
cout<<"\n\t\t  0.83                                 "<<t3;
cout<<"\n\t\t  1.00                                 "<<t4;
cout<<"\n\t\t  1.17                                 "<<t5;
cout<<"\n\t\t '1.33                                 "<<t6;
cout<<"\n\t\t  1.50                                 "<<t7;
cout<<"\n\t\t  1.67                                 "<<t8;
cout<<"\n\t\t  1.83                                 "<<t9;
```

```
cout<<"\n\t\t 2.00                                    "<<t10;
fout<<endl;
fout<<"The required temperatures of combustion for a "<<M<<"kg";
fout<<"\ncapacity rotary furnace (without a recuperator) are\n";
fout<<"\n\t\t  Time (hr)                          Temperature (0C) ";
fout<<"\n\t\t  0.50                                 "<<t1;
fout<<"\n\t\t  0.67                                 "<<t2;
fout<<"\n\t\t  0.83                                 "<<t3;
fout<<"\n\t\t  1.00                                 "<<t4;
fout<<"\n\t\t  1.17                                 "<<t5;
fout<<"\n\t\t  1.33                                 "<<t6;
fout<<"\n\t\t  1.50                                 "<<t7;
fout<<"\n\t\t  1.67                                 "<<t8;
fout<<"\n\t\t  1.83                                 "<<t9;
fout<<"\n\t\t  2.00                                 "<<t10;
};
if(response=='4')
{D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
double tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
double t1,t2,t3,t4,t5,t6,t7,t8,t9,t10;
double l1,l2,l3,l4,l5,l6,l7,l8,l9,l10;
double c1,c2,c3,c4,c5,c6,c7,c8,c9,c10;
double a1,a2,a3,a4,a5,a6,a7,a8,a9,a10;
double r=D3/2,R=r+0.005;
const double ch=4017.5,t0=37.0,e=1850.0;
r=(M>=100?(r-0.000965):r);
R=(M>=100?(R-0.003465):R);
l1=0.835+0.00058*t0;
c1=0.88+0.00023*t0;
a1=l1/(e*c1);
t1=t0+(0.5*ch)+(ch*R*R)*((r*r)/(R*R)-1)/(40*a1);
t1=1.13*(t1-t1/2.5);
t1=(M>=450?(1.13*t1+10):t1);
l2=0.835+0.00058*t1;
c2=0.88+0.00023*t1;
a2=l2/(e*c2);
t2=t1+0.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a2);
t2=1.09*(t2-t2/2.4);
t2=(M<100?(1.61*t2):t2);
t2=(M>150?(1.09*t2+150):t2);
if(M>=230) t2=1.09*t2+210;
t2=(M>=300?(0.76*t2+200):t2);
t2=(M>=400?(1.09*t2):t2);
t2=(M>=450?(1.09*t2+70):t2);
l3=0.835+0.00058*t2;
c3=0.88+0.00023*t2;
```

```
a3=l3/(e*c3);
t3=t2+0.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a3);
t3=1.09*(t3-t3/1.8);
t3=(M<100?(1.45*t3):t3);
t3=(M>150?(1.09*t3+150):t3);
if(M>=230) t3=1.09*t3+130;
t3=(M>=300?(0.78*t3+200):t3);
t3=(M>=400?(1.09*t3+30):t3);
l4=0.835+0.00058*t3;
c4=0.88+0.00023*t3;
a4=l4/(e*c4);
t4=t3+1.0*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a4);
t4=1.11*(t4-t4/1.595);
t4=(M<100?(1.2*t4):t4);
t4=(M>150?(1.11*t4+150):t4);
t4=(M>=400?(t4+50):t4);
t4=(M>450?(1.11*t4+30):t4);
l5=0.835+0.00058*t4;
c5=0.88+0.00023*t4;
a5=l5/(e*c5);
t5=t4+1.17*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a5);
t5=1.09*(t5-t5/1.48);
t5=(M<100?(1.19*t5):t5);
t5=(M>150?(1.09*t5+70):t5);
if(M>=230) t5=1.09*t5+50;
t5=(M>=350?(1.09*t5-210):t5);
t5=(M>=450?(1.09*t5+70):t5);
l6=0.835+0.00058*t5;
c6=0.88+0.00023*t5;
a6=l6/(e*c6);
t6=t5+1.33*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a6);
t6=1.109*(t6-t6/1.398);
t6=(M<100?(1.19*t6):t6);
t6=(M>150?(1.09*t6+65):t6);
if(M>=300) t6=1.09*t6-20;
t6=(M>=350?(1.09*t6-200):t6);
t6=(M>=450?(1.09*t6+50):t6);
l7=0.835+0.00058*t6;
c7=0.88+0.00023*t6;
a7=l7/(e*c7);
t7=t6+1.5*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a7);
t7=1.12*(t7-t7/1.34);
t7=(M<100?(1.2*t7):t7);
t7=(M>150?(1.12*t7+M-150):t7);
t7=(M>=350?(1.12*t7-330):t7);
t7=(M>=450?(1.09*t7-200):t7);
```

```
l8=0.835+0.00058*t7;
c8=0.88+0.00023*t7;
a8=l8/(e*c8);
t8=t7+1.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a8);
t8=1.12*(t8-t8/1.295);
t8=(M<100?(1.2*t8):t8);
t8=(M>=150?(1.12*t8+M/3+10):t8);
if(M>=300) t8=1.12*t8-170;
t8=(M>=350?(1.12*t8-190):t8);
t8=(M>=450?(1.12*t8-223):t8);
l9=0.835+0.00058*t8;
c9=0.88+0.00023*t8;
a9=l9/(e*c9);
t9=t8+1.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a9);
t9=1.125*(t9-t9/1.263);
t9=(M<100?(1.2*t9):t9);
t9=(M>=150?(1.125*t9+M/3):t9);
if(M>=230) t9=1.125*t9-210;
t9=(M>=350?(1.125*t9-261):t9);
t9=(M>=450?(1.125*t9-199.8):t9);
l10=0.835+0.00058*t9;
c10=0.88+0.00023*t9;
a10=l10/(e*c10);
t10=t9+2*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a10);
t10=1.11*(t10-t10/1.24);
t10=(M<100?(1.2*t10):t10);
t10=(M>=150?(1.11*t10+M/3-30):t10);
if(M>=200) t10=1.11*t10-117;
t10=(M>=350?(1.11*t10-247):t10);
t10=(M>=450?(1.11*t10-180):t10);
cout<<endl;
cout<<"The required temperatures of combustion for a "<<M<<"kg";
cout<<"\ncapacity rotary furnace (with a recuperator) are\n";
cout<<"\n\t\t Time (hr)                    Temperature (0C) ";
cout<<"\n\t\t 0.50                           "<<t1;
cout<<"\n\t\t 0.67                           "<<t2;
cout<<"\n\t\t 0.83                           "<<t3;
cout<<"\n\t\t 1.00                           "<<t4;
cout<<"\n\t\t 1.17                           "<<t5;
cout<<"\n\t\t 1.33                           "<<t6;
cout<<"\n\t\t 1.50                           "<<t7;
cout<<"\n\t\t 1.67                           "<<t8;
cout<<"\n\t\t 1.83                           "<<t9;
cout<<"\n\t\t 2.00                           "<<t10;
fout<<endl;
fout<<"The required temperatures of combustion for a "<<M<<"kg";
```

```
fout<<"\ncapacity rotary furnace (with a recuperator) are\n";
fout<<"\n\t\t  Time (hr)                          Temperature (0C) ";
fout<<"\n\t\t  0.50                                "<<t1;
fout<<"\n\t\t  0.67                                "<<t2;
fout<<"\n\t\t  0.83                                "<<t3;
fout<<"\n\t\t  1.00                                "<<t4;
fout<<"\n\t\t  1.17                                "<<t5;
fout<<"\n\t\t  1.33                                "<<t6;
fout<<"\n\t\t  1.50                                "<<t7;
fout<<"\n\t\t  1.67                                "<<t8;
fout<<"\n\t\t  1.83                                "<<t9;
fout<<"\n\t\t  2.00                                "<<t10;};
if(response=='5')
{D3=(M>=100?(D2+(M-75)/1000):(D2-(100-M)*2.5/1000));
cout<<endl;
cout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
cout<<"\n\tThickness of the lining is "<<(M>=100?(tL*1000):(tL*1000-0.7405*(100-
M)))<<"mm";
cout<<"\n\tLength of the cylindrical shell is "<<(M>=100?(925+(M-100)/3):(925-(100-
M)*7))<<"mm";
cout<<"\n\tSlant length of each of the two conical frustums is
"<<(M>=100?(250+0.875*(M-100)):(250+0.875*(M-100)-40))<<"mm"<<endl;
cout<<"\n\tBase diameter of the conical frustum is "<<(4*D3/9)*1000<<"mm";
cout<<"\n\tDiameter of the burner-end of the furnace flange is
"<<(3*D3/11)*1000<<"mm";
cout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3*D3/10)*1000<<"mm";
cout<<endl;
fout<<endl;
fout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
fout<<"\n\tThickness of the lining is "<<(M>=100?(tL*1000):(tL*1000-0.7405*(100-
M)))<<"mm";
fout<<"\n\tLength of the cylindrical shell is "<<(M>=100?(925+(M-100)/3):(925-(100-
M)*7))<<"mm";
fout<<"\n\tSlant length of each of the two conical frustums is
"<<(M>=100?(250+0.875*(M-100)):(250+0.875*(M-100)-40))<<"mm"<<endl;
fout<<"\n\tBase diameter of the conical frustum is "<<(4*D3/9)*1000<<"mm";
fout<<"\n\tDiameter of the burner-end of the furnace flange is
"<<(3*D3/11)*1000<<"mm";
fout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3*D3/10)*1000<<"mm";
fout<<endl;
D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
double tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
double t1,t2,t3,t4,t5,t6,t7,t8,t9,t10;
double l1,l2,l3,l4,l5,l6,l7,l8,l9,l10;
```

```
double c1,c2,c3,c4,c5,c6,c7,c8,c9,c10;
double a1,a2,a3,a4,a5,a6,a7,a8,a9,a10;
double r=D3/2,R=r+0.005;
const double ch=4017.5,t0=37.0,e=1850.0;
r=(M>=100?(r-0.000965):r);
R=(M>=100?(R-0.003465):R);
l1=0.835+0.00058*t0;
c1=0.88+0.00023*t0;
a1=l1/(e*c1);
t1=t0+(0.5*ch)+(ch*R*R)*((r*r)/(R*R)-1)/(40*a1);
t1=t1-t1/2.5;
t1=(M>=450?(t1+10):t1);
l2=0.835+0.00058*t1;
c2=0.88+0.00023*t1;
a2=l2/(e*c2);
t2=t1+0.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a2);
t2=t2-t2/2.4;
t2=(M>=450?(t2+70):t2);
t2=(M>150?(t2+150):t2);
if(M>=230) t2=t2+210;
t2=(M>=350?(t2+200):t2);
t2=(M>450?(t2+50):t2);
l3=0.835+0.00058*t2;
c3=0.88+0.00023*t2;
a3=l3/(e*c3);
t3=t2+0.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a3);
t3=t3-t3/1.8;
t3=(M>150?(t3+150):t3);
if(M>=230) t3=t3+130;
t3=(M>=350?(t3+200):t3);
t3=(M>450?(t3+130):t3);
l4=0.835+0.00058*t3;
c4=0.88+0.00023*t3;
a4=l4/(e*c4);
t4=t3+1.0*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a4);
t4=t4-t4/1.595;
t4=(M>150?(t4+150):t4);
t4=(M>=350?(t4+200):t4);
t4=(M>450?(t4+80):t4);
l5=0.835+0.00058*t4;
c5=0.88+0.00023*t4;
a5=l5/(e*c5);
t5=t4+1.17*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a5);
t5=t5-t5/1.48;
t5=(M>150?(t5+70):t5);
if(M>=230) t5=t5+50;
```

```
t5=(M>=350?(t5+140):t5);
t5=(M>=450?(t5+70):t5);
l6=0.835+0.00058*t5;
c6=0.88+0.00023*t5;
a6=l6/(e*c6);
t6=t5+1.33*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a6);
t6=t6-t6/1.398;
t6=(M>150?(t6+65):t6);
if(M>=230) t6=t6+80;
t6=(M>=350?(t6+80):t6);
t6=(M>=450?(t6+70):t6);
l7=0.835+0.00058*t6;
c7=0.88+0.00023*t6;
a7=l7/(e*c7);
t7=t6+1.5*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a7);
t7=t7-t7/1.34;
t7=(M>150?(t7+M-150):t7);
t7=(M>=350?(t7-60):t7);
l8=0.835+0.00058*t7;
c8=0.88+0.00023*t7;
a8=l8/(e*c8);
t8=t7+1.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a8);
t8=t8-t8/1.295;
t8=(M>=150?(t8+M/3+10):t8);
if(M>=230) t8=t8+30;
t8=(M>=350?(t8+28):t8);
t8=(M>=450?(t8+30):t8);
l9=0.835+0.00058*t8;
c9=0.88+0.00023*t8;
a9=l9/(e*c9);
t9=t8+1.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a9);
t9=t9-t9/1.263;
t9=(M>=150?(t9+M/3):t9);
if(M>=230) t9=t9+20;
t9=(M>=350?(t9+27):t9);
t9=(M>=450?(t9+22):t9);
l10=0.835+0.00058*t9;
c10=0.88+0.00023*t9;
a10=l10/(e*c10);
t10=t9+2*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a10);
t10=t10-t10/1.24;
t10=(M>=150?(t10+M/3-30):t10);
if(M>=230) t10=t10+20;
t10=(M>=350?(t10+20):t10);
t10=(M>=450?(t10+20):t10);
cout<<endl;
```

```
cout<<endl;
cout<<"The required temperatures of combustion for a "<<M<<"kg";
cout<<"\ncapacity rotary furnace (without a recuperator) are\n";
cout<<"\n\t\t  Time (hr)                    Temperature (0C) ";
cout<<"\n\t\t  0.50                         "<<t1;
cout<<"\n\t\t  0.67                         "<<t2;
cout<<"\n\t\t  0.83                         "<<t3;
cout<<"\n\t\t  1.00                         "<<t4;
cout<<"\n\t\t  1.17                         "<<t5;
cout<<"\n\t\t  1.33                         "<<t6;
cout<<"\n\t\t  1.50                         "<<t7;
cout<<"\n\t\t  1.67                         "<<t8;
cout<<"\n\t\t  1.83                         "<<t9;
cout<<"\n\t\t  2.00                         "<<t10;
fout<<endl;
fout<<"The required temperatures of combustion for a "<<M<<"kg";
fout<<"\ncapacity rotary furnace (without a recuperator) are\n";
fout<<"\n\t\t  Time (hr)                    Temperature (0C) ";
fout<<"\n\t\t  0.50                         "<<t1;
fout<<"\n\t\t  0.67                         "<<t2;
fout<<"\n\t\t  0.83                         "<<t3;
fout<<"\n\t\t  1.00                         "<<t4;
fout<<"\n\t\t  1.17                         "<<t5;
fout<<"\n\t\t  1.33                         "<<t6;
fout<<"\n\t\t  1.50                         "<<t7;
fout<<"\n\t\t  1.67                         "<<t8;
fout<<"\n\t\t  1.83                         "<<t9;
fout<<"\n\t\t  2.00                         "<<t10;};
if(response=='6')
{D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
cout<<endl;
cout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
double tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
cout<<"\n\tThickness of the lining is "<<tL2<<"mm";
double L2=(M>=100?(925+1.1*(M-100)/3):(925-0.9*(100-M)*7));
        L2=(M==100?(925+1.1*M/3):L2);
cout<<"\n\tLength of the cylindrical shell is "<<L2<<"mm";
cout<<"\n\tSlant length of each of the two conical frustums is "<<(M>=100?(250+0.9*(M-
100)):(250+0.9*(M-100)-40))<<"mm";
cout<<"\n\tBase diameter of the conical frustum is "<<(4000*D3/9)<<"mm";
cout<<"\n\tDiameter of the burner-end of the furnace flange is "<<(3000*D3/11)<<"mm";
cout<<"\n\tDiameter of the charging-end of the furnace flange is
"<<(3000*D3/10)<<"mm\n";
cout<<"\nFor the Recuperator design parameters :  \n";
cout<<"\n\tDiameter of the inner cylinder of the recuperator is "<<(M<=300?(5000*D3/9-
28.85):(5000*D3/9-115))<<"mm";
```

```
	cout<<"\n\tDiameter of the outer cylinder of the recuperator is "<<(M<=300?(5000*D3/9+tL2/5-4.314):(5000*D3/9+tL2/5-25))<<"mm";
	cout<<"\n\tLength  of the inner cylinder of the recuperator is "<<(M<=300?(1.750*L2-94.92):(1.750*L2+453.08))<<"mm";
	cout<<"\n\tLength  of the outer cylinder of the recuperator is "<<(M<=300?(1.750*L2-3000*D3/5+200.08):(1.750*L2-3000*D3/5+973.08))<<"mm";
	cout<<"\n\tRecuperator sprout's base diameter is "<<(750*D3-1.447)<<"mm";
	cout<<"\n\tRecuperator sprout's slant length is "<<(750*D3-6.447)<<"mm";
	cout<<endl;
	fout<<endl;
	fout<<"\n\tDiameter of the cylindrical shell is "<<(D3*1000)<<"mm";
	fout<<"\n\tThickness of the lining is "<<tL2<<"mm";
	fout<<"\n\tLength of the cylindrical shell is "<<L2<<"mm";
	fout<<"\n\tSlant length of each of the two conical frustums is "<<(M>=100?(250+0.9*(M-100)):(250+0.9*(M-100)-40))<<"mm"<<endl;
	fout<<"\n\tBase diameter of the conical frustum is "<<(4000*D3/9)<<"mm";
	fout<<"\n\tDiameter of the burner-end of the furnace flange is "<<(3000*D3/11)<<"mm";
	fout<<"\n\tDiameter of the charging-end of the furnace flange is "<<(M<=300?(3*(5000*D3/9-25)):(3*(5000*D3/9-115)))<<"mm\n";
	fout<<"\n\tDiameter of the inner cylinder of the recuperator is "<<(M<=300?(5000*D3/9-28.85):(5000*D3/9-115))<<"mm";
	fout<<"\n\tDiameter of the outer cylinder of the recuperator is "<<(M<=300?(5000*D3/9+tL2/5-4.314):(5000*D3/9+tL2/5-25))<<"mm";
	fout<<"\n\tLength  of the inner cylinder of the recuperator is "<<(M<=300?(1.750*L2-94.92):(1.750*L2+320))<<"mm";
	fout<<"\n\tLength  of the outer cylinder of the recuperator is "<<(M<=300?(1.750*L2-3000*D3/5):(1.750*L2-3000*D3/5+790))<<"mm";
	fout<<"\n\tRecuperator sprout's base diameter is "<<(750*D3-1.447)<<"mm";
	fout<<"\n\tRecuperator sprout's slant length is "<<(750*D3-6.447)<<"mm";
	fout<<endl;
	D3=(M>=100?(D2+(M-70)/1000):(D2-0.88*(100-M)*2.5/1000));
	tL2=(M>=100?(tL*1000+D3/110):(tL*1000-0.67*(100-M)));
	double t1,t2,t3,t4,t5,t6,t7,t8,t9,t10;
	double l1,l2,l3,l4,l5,l6,l7,l8,l9,l10;
	double c1,c2,c3,c4,c5,c6,c7,c8,c9,c10;
	double a1,a2,a3,a4,a5,a6,a7,a8,a9,a10;
	double r=D3/2,R=r+0.005;
	const double ch=4017.5,t0=37.0,e=1850.0;
	r=(M>=100?(r-0.000965):r);
	R=(M>=100?(R-0.003465):R);
	l1=0.835+0.00058*t0;
	c1=0.88+0.00023*t0;
	a1=l1/(e*c1);
	t1=t0+(0.5*ch)+(ch*R*R)*((r*r)/(R*R)-1)/(40*a1);
	t1=1.13*(t1-t1/2.5);
	t1=(M>=450?(1.13*t1+10):t1);
```

```
l2=0.835+0.00058*t1;
c2=0.88+0.00023*t1;
a2=l2/(e*c2);
t2=t1+0.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a2);
t2=1.09*(t2-t2/2.4);
t2=(M<100?(1.61*t2):t2);
t2=(M>150?(1.09*t2+150):t2);
if(M>=230) t2=1.09*t2+210;
t2=(M>=300?(0.76*t2+200):t2);
t2=(M>=400?(1.09*t2):t2);
t2=(M>=450?(1.09*t2+70):t2);
l3=0.835+0.00058*t2;
c3=0.88+0.00023*t2;
a3=l3/(e*c3);
t3=t2+0.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a3);
t3=1.09*(t3-t3/1.8);
t3=(M<100?(1.45*t3):t3);
t3=(M>150?(1.09*t3+150):t3);
if(M>=230) t3=1.09*t3+130;
t3=(M>=300?(0.78*t3+200):t3);
t3=(M>=400?(1.09*t3+30):t3);
l4=0.835+0.00058*t3;
c4=0.88+0.00023*t3;
a4=l4/(e*c4);
t4=t3+1.0*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a4);
t4=1.11*(t4-t4/1.595);
t4=(M<100?(1.2*t4):t4);
t4=(M>150?(1.11*t4+150):t4);
t4=(M>=400?(t4+50):t4);
t4=(M>450?(1.11*t4+30):t4);
l5=0.835+0.00058*t4;
c5=0.88+0.00023*t4;
a5=l5/(e*c5);
t5=t4+1.17*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a5);
t5=1.09*(t5-t5/1.48);
t5=(M<100?(1.19*t5):t5);
t5=(M>150?(1.09*t5+70):t5);
if(M>=230) t5=1.09*t5+50;
t5=(M>=350?(1.09*t5-210):t5);
t5=(M>=450?(1.09*t5+70):t5);
l6=0.835+0.00058*t5;
c6=0.88+0.00023*t5;
a6=l6/(e*c6);
t6=t5+1.33*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a6);
t6=1.109*(t6-t6/1.398);
t6=(M<100?(1.19*t6):t6);
```

```
t6=(M>150?(1.09*t6+65):t6);
if(M>=300) t6=1.09*t6-20;
t6=(M>=350?(1.09*t6-200):t6);
t6=(M>=450?(1.09*t6+50):t6);
l7=0.835+0.00058*t6;
c7=0.88+0.00023*t6;
a7=l7/(e*c7);
t7=t6+1.5*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a7);
t7=1.12*(t7-t7/1.34);
t7=(M<100?(1.2*t7):t7);
t7=(M>150?(1.12*t7+M-150):t7);
t7=(M>=350?(1.12*t7-330):t7);
t7=(M>=450?(1.09*t7-200):t7);
l8=0.835+0.00058*t7;
c8=0.88+0.00023*t7;
a8=l8/(e*c8);
t8=t7+1.67*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a8);
t8=1.12*(t8-t8/1.295);
t8=(M<100?(1.2*t8):t8);
t8=(M>=150?(1.12*t8+M/3+10):t8);
if(M>=300) t8=1.12*t8-170;
t8=(M>=350?(1.12*t8-190):t8);
t8=(M>=450?(1.12*t8-223):t8);
l9=0.835+0.00058*t8;
c9=0.88+0.00023*t8;
a9=l9/(e*c9);
t9=t8+1.83*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a9);
t9=1.125*(t9-t9/1.263);
t9=(M<100?(1.2*t9):t9);
t9=(M>=150?(1.125*t9+M/3):t9);
if(M>=230) t9=1.125*t9-210;
t9=(M>=350?(1.125*t9-261):t9);
t9=(M>=450?(1.125*t9-199.8):t9);
l10=0.835+0.00058*t9;
c10=0.88+0.00023*t9;
a10=l10/(e*c10);
t10=t9+2*ch+ch*R*R*((r*r)/(R*R)-1)/(4*a10);
t10=1.11*(t10-t10/1.24);
t10=(M<100?(1.2*t10):t10);
t10=(M>=150?(1.11*t10+M/3-30):t10);
if(M>=200) t10=1.11*t10-117;
t10=(M>=350?(1.11*t10-247):t10);
t10=(M>=450?(1.11*t10-180):t10);
cout<<endl;
cout<<"The required temperatures of combustion for a "<<M<<"kg";
cout<<"\ncapacity rotary furnace (with a recuperator) are\n";
```

```
cout<<"\n\t\t  Time (hr)                          Temperature (0C) ";
cout<<"\n\t\t  0.50                                  "<<t1;
cout<<"\n\t\t  0.67                                  "<<t2;
cout<<"\n\t\t  0.83                                  "<<t3;
cout<<"\n\t\t  1.00                                  "<<t4;
cout<<"\n\t\t  1.17                                  "<<t5;
cout<<"\n\t\t  1.33                                  "<<t6;
cout<<"\n\t\t  1.50                                  "<<t7;
cout<<"\n\t\t  1.67                                  "<<t8;
cout<<"\n\t\t  1.83                                  "<<t9;
cout<<"\n\t\t  2.00                                  "<<t10;
fout<<endl;
fout<<"The required temperatures of combustion for a "<<M<<"kg";
fout<<"\ncapacity rotary furnace (with a recuperator) are\n";
fout<<"\n\t\t  Time (hr)                          Temperature (0C) ";
fout<<"\n\t\t  0.50                                  "<<t1;
fout<<"\n\t\t  0.67                                  "<<t2;
fout<<"\n\t\t  0.83                                  "<<t3;
fout<<"\n\t\t  1.00                                  "<<t4;
fout<<"\n\t\t  1.17                                  "<<t5;
fout<<"\n\t\t  1.33                                  "<<t6;
fout<<"\n\t\t  1.50                                  "<<t7;
fout<<"\n\t\t  1.67                                  "<<t8;
fout<<"\n\t\t  1.83                                  "<<t9;
fout<<"\n\t\t  2.00                                  "<<t10;};};};
cout<<"\nDo you wish to continue? (Y/N) : ";
cin>>response3;
cout<<endl;
esc2:{if(response3=='Y'||response3=='y')
        goto esc1;
return 0;
};}
```

APPENDIX B

OUTPUTS FROM THE DEVELOPED SOFTWARE

This program evaluates the essential parameters needed to design a recuperator-incorporated rotary furnace. The various options available to choose from are as outlined below, depending on what the user is interested in.

What parameters are you interested in?

(a) A rotary furnace without a recuperator? Enter 1

(b) A rotary furnace with a recuperator? Enter 2

(c) Temperatures of combustion of a rotary furnace

without a recuperator? Enter 3

(d) Temperatures of combustion of a rotary furnace

with a recuperator? Enter 4

(e) Interested in both (a) and (c)? enter 5

(f) Interested in both (b) and (d)? enter 6

Make your choice by picking the appropriate number against the chosen option: 1

Enter the capacity of the furnace, that is the maximum quantity of charges it is intended for (kg): 200
Do you wish to supply the following parameters for your lining material (recommended data will be used otherwise): shrinkage allowance, Compressive strength and Young's Modulus? (Y/N) N

Diameter of the cylindrical shell is 577.536mm
Thickness of the lining is 132.895mm
Length of the cylindrical shell is 958.333mm
Slant length of each of the two conical frustums is 337.5mm

Base diameter of the conical frustum is 256.683mm
Diameter of the burner-end of the furnace flange is 157.51mm
Diameter of the charging-end of the furnace flange is 173.261mm

Do you wish to continue? (Y/N) : Y

Make your choice by picking the appropriate number against the chosen option: 2

```
Enter the capacity of the furnace, that is the maximum quantity of
 charges it is intended for (kg): 250
Do you wish to supply the following parameters for your lining material
(recommended data will be used otherwise): shrinkage allowance,
Compressive strength and Young's Modulus? (Y/N) N

      Diameter of the cylindrical shell is 623.545mm
      Thickness of the lining is 135.65mm
      Length of the cylindrical shell is 980mm
      Slant length of each of the two conical frustums is 385mm

      Base diameter of the conical frustum is 277.131mm
      Diameter of the burner-end of the furnace flange is 170.058mm
      Diameter of the charging-end of the furnace flange is 964.242mm

      Diameter of the inner cylinder of the recuperator is 317.564mm
      Diameter of the outer cylinder of the recuperator is 369.23mm
      Length  of the inner cylinder of the recuperator is 1620.08mm
      Length  of the outer cylinder of the recuperator is 1340.87mm
      Recuperator sprout's base diameter is 466.212mm
      Recuperator sprout's slant length is 461.212mm
```

Do you wish to continue? (Y/N) : Y

```
Make your choice by picking the appropriate number against the chosen
option: 3

Enter the capacity of the furnace, that is the maximum quantity of
 charges it is intended for (kg): 300
Do you wish to supply the following parameters for your lining material
(recommended data will be used otherwise): shrinkage allowance,
Compressive strength and Young's Modulus? (Y/N) N
      Supply the following parameters for the lining :
      Shrinkage allowance   : 0.0877
      Compressive strength (N/m2) :3e+9
      Young's Modulus (N/m2) : 300e+9
The required temperatures of combustion for a 300kg
capacity rotary furnace (without a recuperator) are
            Time (hr)                 Temperature (0C)
            0.50                         1035.34
            0.67                         1131.82
            0.83                         1216.29
            1.00                         1235.64
            1.17                         1294.22
            1.33                         1382.07
            1.50                         1456.56
            1.67                         1491.64
            1.83                         1499.22
            2.00                         1506.1
```

Do you wish to continue? (Y/N) : Y

```
Make your choice by picking the appropriate number against the chosen
option: 4

Enter the capacity of the furnace, that is the maximum quantity of
 charges it is intended for (kg): 350
Do you wish to supply the following parameters for your lining material
(recommended data will be used otherwise): shrinkage allowance,
Compressive strength and Young's Modulus? (Y/N) y
Supply the following parameters for the lining :
        Shrinkage allowance   : 0.0877
        Compressive strength (N/m2) : 3e+9
        Young's Modulus (N/m2)  : 300e+9
The required temperatures of combustion for a 350kg
capacity rotary furnace (with a recuperator) are

                Time (hr)                  Temperature (OC)
                0.50                           1156.26
                0.67                           1259.51
                0.83                           1390.93
                1.00                           1533.74
                1.17                           1708.12
                1.33                           1803.33
                1.50                           1880
                1.67                           2055.7
                1.83                           2058.14
                2.00                           2052.42
```

Do you wish to continue? (Y/N) : Y

```
Make your choice by picking the appropriate number against the chosen
option: 5

Enter the capacity of the furnace, that is the maximum quantity of
 charges it is intended for (kg): 400
Do you wish to supply the following parameters for your lining material
(recommended data will be used otherwise): shrinkage allowance,
Compressive strength and Young's Modulus? (Y/N) N

       Diameter of the cylindrical shell is 743.967mm
       Thickness of the lining is 143.764mm
       Length of the cylindrical shell is 1025mm
       Slant length of each of the two conical frustums is 512.5mm

       Base diameter of the conical frustum is 330.652mm
       Diameter of the burner-end of the furnace flange is 202.9mm
       Diameter of the charging-end of the furnace flange is 223.19mm

The required temperatures of combustion for a 400kg
```

capacity rotary furnace (without a recuperator) are

Time (hr)	Temperature (0C)
0.50	1010.73
0.67	1129.96
0.83	1280.67
1.00	1359.53
1.17	1395.85
1.33	1419.68
1.50	1436.55
1.67	1481.45
1.83	1497.37
2.00	1503.94

Do you wish to continue? (Y/N) : Y

Make your choice by picking the appropriate number against the chosen option: 6

Enter the capacity of the furnace, that is the maximum quantity of charges it is intended for (kg): 450
Do you wish to supply the following parameters for your lining material (recommended data will be used otherwise): shrinkage allowance, Compressive strength and Young's Modulus? (Y/N) N

Diameter of the cylindrical shell is 791.472mm
Thickness of the lining is 146.44mm
Length of the cylindrical shell is 1053.33mm
Slant length of each of the two conical frustums is 565mm

Base diameter of the conical frustum is 351.766mm
Diameter of the burner-end of the furnace flange is 215.856mm
Diameter of the charging-end of the furnace flange is 974.121mm

Diameter of the inner cylinder of the recuperator is 324.707mm
Diameter of the outer cylinder of the recuperator is 443.995mm
Length of the inner cylinder of the recuperator is 2163.33mm
Length of the outer cylinder of the recuperator is 2158.45mm
Recuperator sprout's base diameter is 592.157mm
Recuperator sprout's slant length is 587.157mm

The required temperatures of combustion for a 450kg capacity rotary furnace (with a recuperator) are

Time (hr)	Temperature (0C)
0.50	1284.9
0.67	1490.5
0.83	1563.21
1.00	1561.5
1.17	1808.97

1.33	1952.06
1.50	1940.79
1.67	2049.15
1.83	2060.13
2.00	2057.92

Do you wish to continue? (Y/N) : Y

```
Make your choice by picking the appropriate number against the chosen
option: 7

YOU HAVE NOT MADE A VALID CHOICE!
TRY TO MAKE A CORRECT CHOICE NEXT TIME YOU RUN THE PROGRAM
```