# DETERMINATION OF RATE OF FUEL CONSUMPTION RATE OF A HEAT EXCHANGER (BOILER)

BY

*75%*

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# CERTIFICATION

This is to certify that the project was supervised, moderated and approved by the following under listed persons on behalf of the chemical engineering department, school of engineering and engineering technology, Federal University of Technology, Minna.


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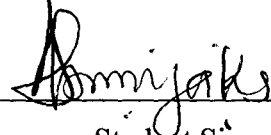


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# DECLARATION

I, Yakubu Abba Usman declare that this project report is the report of my work, and to the best of my knowledge have not been presented elsewhere for the award of certificate, diploma or degree.

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## DEDICATION

This research work is dedicated to Almighty God whose grace and mercy saw through to the successful completion of my first degree programme. And also to my parents and siblings who stood by me throughout my undergraduate days.

## ACKNOWLEDGEMENT

Rahila Bello, you guys are great. To my good family in Minna Mr. & Mrs. Toma Minti Thanks for your support. To my room – mate Ibrahim Yahaya staying with you for six years was a great priviledge. Finally to my friends Racheal Isah, Dauda Ishaku, Uwani Samaila, Joy Barde and my sister Patience Ibrahim, the Lord is your strength

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# ABSTRACT

The process of heat transfer occurring in a heat exchanger involves a number of operations. Analytical study of the heat exchanger applying the principles of thermodynamics and heat transfer gives a detailed account of the step-by-step processes involved in the generation of heat and its transfer within the heat exchanger.

The study carried out in this research work was done on a plate boiler having a combustion chamber where n-Decane fuel burns to provide the heat required for its operation. From the principle of thermodynamics applied, the amount of heat generated from the combustion chamber as a result of the combustion reaction (burning) of the n-Decane fuel with Air free of carbon dioxide $CO_2$, water $H_2O$ and rare gases was found to be 5287,540,573.2J. Also applying the principle of heat transfer on the boiler led to the determination of the various effectiveness of the boiler which are fin effectiveness 18.5752689 and extended surface effectiveness 2.75757269 and also the amount of heat that was dissipated by the boiler 39708.38736J. Relating the amount of heat dissipated by the boiler with the amount of heat generated by the combustion chamber gave the rate of fuel consumption of the boiler to be 92.1363kg/day.

Successfully the rate of fuel consumption of the boiler was determined. The values obtained will provide information if required for the modification of the boiler for better performance and also serve as a sort of standard for characterizing the boiler and its performance and also detecting any form of malfunction in the boiler.

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## 1.0    INTRODUCTION

## 1.1    BACKGROUND OF STUDY

In majority of chemical process heat is either given out or absorbed, and fluids must often be either heated or cooled in wide range of plant such as furnaces, evaporators, distillation units, dryers and reaction vessels where one of the major problems is that of transferring heat at the desired rate. In addition it may be necessary to prevent the loss of heat from a hot vessel or pipe system. The control of the flow of heat at the desired rate forms one of the most important areas of chemical engineering. Provided that a temperature difference exists between two parts of a system, heat transfer will take place in one or more of three different ways.

Conduction; In a solid, the flow of heat by conduction is the result of the transfer of vibrational energy from one molecule to another and it occurs in addition as a result of the transfer of kinetic energy. Heat transfer by conduction may also arise from movement of free electrons a process, which is particularly important with metals and accounts for their thermal conductivities.

Convection; Heat transfer by convection arises from the mixing of elements of fluid. If this mixing occurs as a result of density differences as for example when a pool of liquid is heated from below the process is known as natural convection it is important to note that convection requires mixing of fluid elements and is not governed by temperature difference alone as is the case in conduction and radiation.

Radiation; All materials radiate thermal energy in the form of electromagnetic waves when this radiation falls on a second body it may be partially reflected, transmitted, or absorbed. It is only the fraction that is absorbed that appears as heat in the body.

The plate heat exchanger was invented by Dr Richard Seligman in 1923 revolutionizing the methods for the indirect heating and cooling of fluids. A plate heat exchanger is a type of heat exchanger that uses metal plates to transfer heat between two fluids. This has a major advantage over a conventional heat surface area because the fluids spread out over the plate. This facilitates the transfer of heat and greatly increases the speed of the temperature change. It is not as common to see plate heat exchangers due to the fact that they need well-sealed gaskets to prevent the fluids from escaping although modern manufacturing processes have made them feasible.

The cost of production is now increasing due to increase in the cost powering the heat-exchanging unit. The heat exchanger been considered in this work is a boiler which can either be powered electrically or using burning fuel such as coal, diesel and other hydrocarbon fuel. The fuel used in this work is n-Decane fuel. Because of the continues increase in the cost of crude oil in the world market today, a device to reduce the fuel consumption rate of these boilers must be created so that production will not suffer set back due to burning fuel cost.

## 1.2    AIMS AND OBJECTIVE

The objective of this study is to determine the fuel consumption rate of a heat exchanger (boiler) by considering the basic principles of thermodynamics of the combustion chamber and heat transfer rate of the boiler which is aimed at determining the amount of n-decane fuel burnt per day to produce a certain amount of heat.

## 1.3    SCOPE OF STUDY

The research work carried out is the pure application of the principles of thermodynamics and heat transfer on a heat exchanger (boiler) to obtain the effectiveness of the boiler both fin and extended surface which in turn is used to determine the rate of fuel consumption and steam generation of the boiler.

# CHAPTER TWO

## 2.0 LITERATURE REVIEW

## 2.1 HEAT EXCHANGER

A heat exchanger is a device built for efficient heat transfer from one fluid to another, whether the fluids are separated by a solid wall so that they never mix, or the fluids are directly contacted. They are widely used in petroleum refineries, chemical plants, petrochemical plants, natural gas processing plants, refrigeration, power plant, air conditioning and space heating. One common example of a heat exchanger is the radiator in a car, in which a hot engine cooling fluid, like anti-freeze, transfers heat to air flowing through the radiator.

Heat exchangers may be classified according to their flow arrangement. Parallel flow heat exchangers where the two fluids enter the exchanger at the same end and travel in parallel to one another to other side.

Hot in     Cold in

Hot out     Cold out
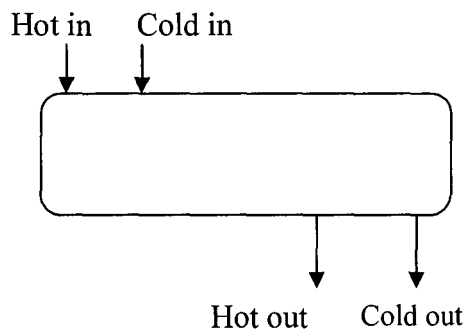
Fig 2.10 parallel flow heat exchanger

- In counter –flow heat exchangers the fluids enters the exchanger at opposite ends. The counter current design is most efficient in that it can transfer the most heat.

Hot in          Cold out

Hot out          Cold in
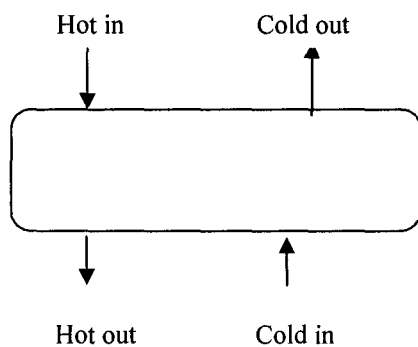
Figs 2.11 counter current flow heat exchanger

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- In a cross flows heat exchanger, the fluid travel roughly perpendicular to one another through the exchanger
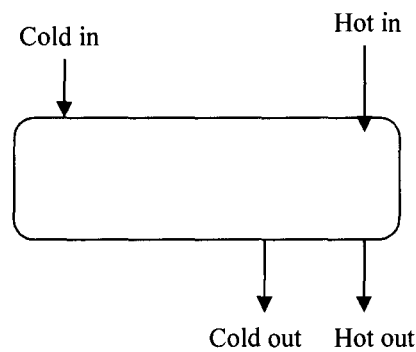


Fig 2.12 Cross flow heat exchanger

For efficiency heat exchanger are designed to maximize the surface area of the wall between the two fluids, while minimizing resistance to fluid flow through the heat exchanger. The exchanger's performance can also be affected by the addition of fins or corrugations in one or both direction which increase surface area and may channel fluid flow or induce turbulence. The driving temperature across the heat transfer surface varies with position, but an appropriate mean temp can be defined. In most simple systems this is the log mean temperature difference (LMTD)

## 2.1.1    TYPE OF HEAT EXCHANGER

There are about seven types of heat exchanger. They are:

1. Shell and tube heat exchanger

2. Plate heat exchanger

3. Regenerative heat exchanger

4. Adiabatic wheel heat exchanger

5. Fluid heat exchanger

6. Dynamic scraped surface heat exchanger

7. Phase change heat exchanger.

### SHELL AND TUBE HEAT EXCHANGERS

This is a typical heat exchanger usually used for higher-pressure applications; it consists of series of tubes through which one of the fluids runs the second fluid runs over the tube to be heated or cooled. The set of tubes is called tube bundle and may be composed of several types of tube' plain longitudinally finned etc. shell and tube heat exchanger are the most widely used form of heat transfer equipment because they have large heat transfer surface in a relatively small size

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(volume). They are fabricated form allow steels to resist corrosion and are used for heating cooling and for condensing a very wide range of fluids. The simplest type of unit shown in fig2.13 has fixed tubes plates at each end into which the tubes are expanded. The tubes are connected so that the internal fluid makes several passes up and down the exchanger thus enabling high velocity of flow to be obtained for a given heat transfer area and through put the fluid. The fluid flowing in the shell is made to flow first in one sense and then in the opposite sense across the tube bundle by fitting a series of baffles along the length. The baffles are frequently of the segmental form with about 25 percent cut array to provide the free space to increase the velocity of flow across the tubes thus giving higher rates of heat transfer.
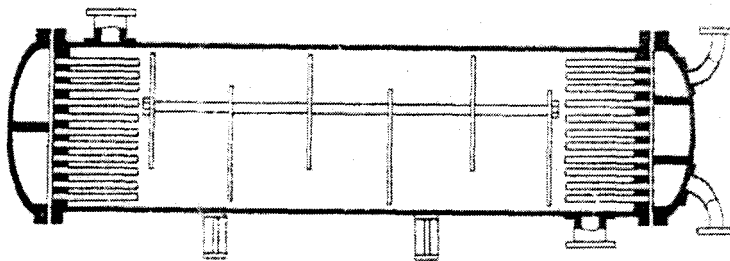


Figure 9.62.   Heat exchanger with fixed tube plates (four tube, one shell-pass)

Fig 2.13 Heat Exchanger with fixed tube plates

Many different mechanical arrangement of the shell are used it is convenient to use a basis for classification. The standard published by the Tubular Exchanger Manufacturer's Association TEMA is outlined.
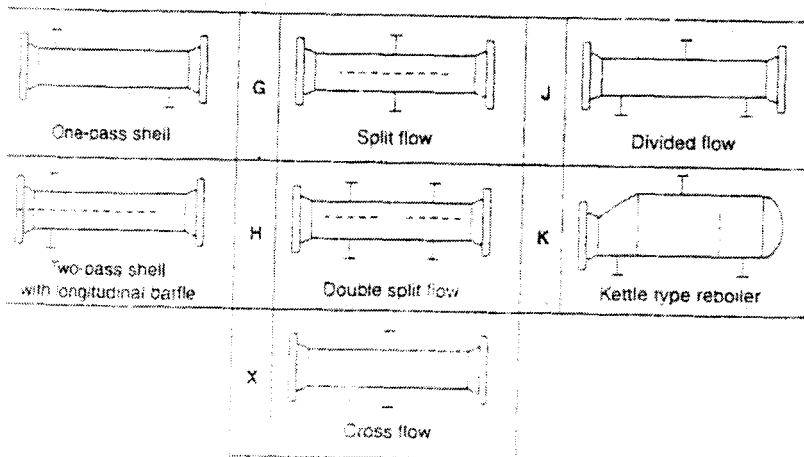


Figure 9.67.   TEMA shell types

Fig 2.14 TEMA Shell Type.

Of the various shell types shown in fig 2.14 the simplest with entry and exit nozzles at apposite ends of a single pass exchanger is the TEMA E-type on which most designs are based,

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although these may be adopted for other shell types by allowing for resulting velocity changes. The TEMA F-type has a longitudinal baffle giving two shell passes and this provides an alternative arrangement to the use of two shells required in order to cope with a close temp approach or lower shell-side flow rates. The pressure drop in two shell is some eight times greater than that encountered in the E-type design although any potential leakage between the longitudinal baffle and the shell in the F-type design may restrict the range application. The so-called "split-flow" type of unit with longitudinal baffle is classified as the TEMA G-type whose performance is superior although the pressure drop is similar to the E-type. This design is used mainly for reboilers and only occasionally for systems where there is no change of phase. The so-called "divided-flow" types. The TEMA J-type, has one inlet and two outlet nozzles and with a pressure drop some one-eighth of the E-type finds application in gas coolers and condenser operating at low pressures. The TEMA X-type shell has no cross baffles and hence the shell-side fluid is in pure counter flow giving extremely low pressure drops and again, this type of design is used for gas cooling and condensation at low pressures.

The shell of a heat exchanger is commonly made of carbon steel and standard pipes are used for the smaller sizes and rolled welded plate for the larger sizes (0.4-1.0m). The thickness of the shell may be calculated from the formula for thin-walled cylinders and a minimum thickness of 9.5mm is used for shells over 0.33m outer diameter and 11.1mm for shells over 0.9m outer diameter. Unless the shell is designed to operate at very high pressure, the calculated wall thickness is usually less than these values although a corrosion allowance of 3.2mm is commonly added to all carbon steel parts and thickness is determined more by rigidity requirements than simply internal pressure.

A shell diameter should be as such as to give as close a fit to the tube bundle as practical in order to reduce by passing round the outside of the bundle. In selecting the tube diameter, it may be noted that smaller tubes gives a larger heat transfer area for a given shell although 19mm outer diameter tubes are normally the minimum size used in order to permit adequate cleaning.

Heat exchanger tubes are usually in the range 16mm to 50mm outer diameter; the smaller diameter usually being preferred as these give more compact and therefore widely used for heavily

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fouling fluids. The tube thickness or garage must be such as to withstand internal pressure and also to provide an adequate corrosion allowance.

## PLATE HEAT EXCHANGER

Another type of heat exchanger is the plate exchanger. It composes of multiple thin, slightly separated plates that have very large surface areas and fluid flow passages for heat transfer. This stacked plate arrangement can be more effective in a given space, than the shell and tube heat exchanger. Advances in gasket and brazing technology have made the plate type heat exchanger increasingly practical. There are many types of permanently bonded plate heat exchanger such as often specified for close loop application such as refrigeration while the plate and frame are used in open loops, which allows periodic disassembly cleaning and inspection. Plate heat exchanger differ in the types of plates that are used, and the configurations of these plates some plates may be stamped with chevron or other patterns, where others may have machined fins and /or grooves. The major advantage of plate heat exchanger over other heat exchanger is that the fluids are exposed to a much larger surface area because the fluids spread out over plate; this facilitates the transfer of heat and greatly increases the speed of the temperature change.

A series of plate type heat exchangers was first developed by the APV company. The general construction is shown in fig. 2.15 which shows an Alfa-Laval exchanger and from which it is seen that the equipment consists of a series of parallel plates held firmly together between substantial head frames. The plates are one piece pressings frequently of stainless steel and are spaced by rubber sealing gaskets cemented into a channel of constantly changing direction and section with normal construction the gap between the plates is 1.3mm-1.5mm. Each liquid flows in alternate spaces and a large surface can be obtained in a small volume.

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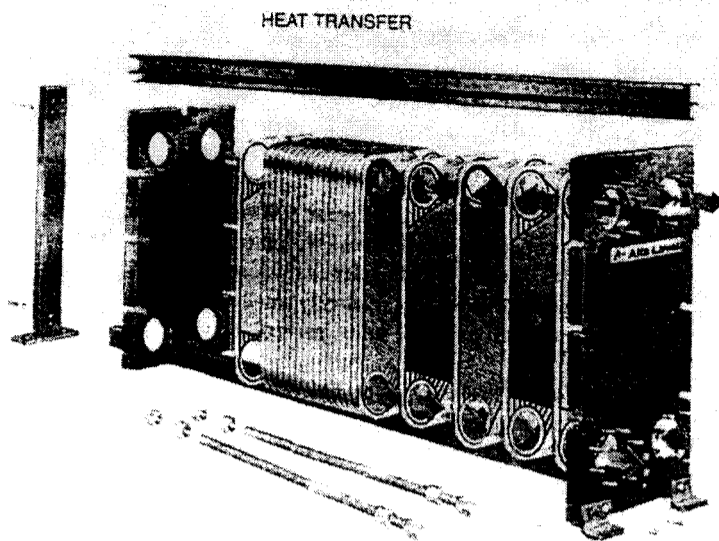
Fig 2.15 Alfa-Laval plate heat Exchanger

Because of the shape of plates, the developed area of surface is appreciably greater than the projected area. This is shown in the table below for four common sizes of plate.

| Plate Type | Projects Area (m$^2$) | Developed Area (m$^2$) |
|---|---|---|
| HT | 0.09 | 0.13 |
| HX | 0.13 | 0.17 |
| HM | 0.27 | 0.35 |
| HF | 0.36 | 0.43 |

Table 2.1

A high degree of turbulence is obtained even at low flow rates and the high heat transfer coefficients obtained are illustrated by the data in table. This refers to the heating of cold water by equal flow of hot water in the HF type exchanger (Aluminum or copper) at an average temperature of 310k.

| HEAT TRANSFER FOR PLATE | | WATER FLOW | | U. based on developed Area | |
|---|---|---|---|---|---|
| W/K | BTU/h$^o$F | L/s | gal/h | Kw/m$^2$k | BTU/hft$^{20}$F |
| 1530 | 3000 | 0.200 | 550 | 3.70 | 650 |
| 2100 | 4000 | 1.705 | 850 | 4.94 | 870 |
| 2640 | 1.580 | 1. 580 | 1250 | 6.13 | 1080 |

Table 2.2

The high transfer coefficient enables these exchangers to be operated with very small temperature. So that a high heat recovery is obtained. These units have been particularly successful

in the diary and brewing industries where the low liquid capacity and the close control of temperature have been valuable features, another advantage is that they are necessity for a long gasket is an inherent weakness but the exchanger have been worked successfully up to 423k at pressure of 930 kN/m$^2$. They are now being used in the processing and gas industries with solvents, sugar, acetic acid, ammoniacal liquor and so on.

## REGENERATIVE HEAT EXCHANGER

The third of heat exchanger is the regenerative heat exchanger. In this the heat from a process is used to warm the fluids to be used in the process and the same type of fluid is used either side of the heat exchanger. This exchanger can either be plate and frame or shell or tube construction. These exchangers are used only for gases and not for liquids. The major factor for this is heat capacity of the heat transfer matrix.

## ADIABATIC WHEEL HEAT EXCHANGER

The fourth type of heat exchanger uses an intermediate fluid or solid store to hold heat, which is then moved to the other side of the heat exchanger to be released. Two examples of this are adiabatic wheels, which consists of a large wheel with fine threads rotating through the hot and cold fluids and fluid heat exchanger. This type is used when it is acceptable for a small amount of mixing to occur between the two streams.

## FLUID HEAT EXCHANGER

This is a heat exchanger with a gas passing upwards through a shower of fluid (often water), and the fluid is then taken elsewhere before being cooled. This is commonly used for cooling gases whilst also removing certain, impurities thus solving two problems at once. It is widely used in espresso machines as an energy saving method of cooling super-heated water to be used in extraction of espresso.

## DYNAMIC SCRAPED SURFACE HEAT EXCHANGER

This is another type of heat exchanger often collect Dynamic heat exchanger or scraped surface heat exchanger. This is mainly used for heating or cooling with high viscosity products crystallization processes, evaporation and high foiling application. Long running times are achieved due to the continuous scraping of the surface, thus avoiding fouling and achieving a sustainable heat transfer rate during the process.

## PHASE-CHANGE HEAT EXCHANGERS

A typical example is the kettle reboiler used for industrial distillation towers. Typical water-cooled surface condenser, in addition to heating up or cooling down fluids in just a single phase heat exchanger can be used either to heat a liquid to evaporate (or boil) it or used as condensers to cool a vapour and condense it to a liquid. In chemical plants and refineries reboilers used to heat incoming feed for distillation towers are often heat exchanger. Distillation set-ups typically use condensers to condense distillate vapors back into liquid power plants which have steam driven turbines commonly use heat exchanger to boil water into steam. Heat exchangers or similar units for producing steam from water are often called boiler. In nuclear power plants called pressurized water reactors, special large heat exchanger which pass heat from the primary (reactor plant) system to the secondary (steam plant) system producing steam from water in the process are called steam generator. All fossil fueled and nuclear power from the turbines have surface condensers to convert the exhaust steam from the turbines into condensate (water) for re-use. In order to conserve energy and cooling capacity in chemical and other plants, regenerative heat exchangers can be used to transfer heat from one stream that needs to be cooled to another stream that needs be heated, such as distillate cooling and reboiler feed pre-heating. This term can also refer to heat exchangers that contain a liquid phase due to the small volume difference between these states. This change of phase effectively acts as a buffer because it occurs at a constant temperature but still allows for the exchanger to accept additional heat. One example where this has been investigated is for use in high power aircraft electronics.

## 2.1.2 HEAT EXCHANGER DESIGN

## PROCESS CONDITIONS

A first stage consideration in the design process is the allocation of fluids to either shell or tubes and by and large the more corrosive fluid is passed through the tubes to reduce the costs of expensive alloys and clad components. Similarly, the fluid with the greatest fouling tendency is also usually passed through the tubes where cleaning is easier. Further more velocities through the tubes are generally higher and more readily controllable and can be adjusted to reduce fouling. In selecting a design velocity it should be recognized that at high fouling is reduced, but pressure

drops are higher. Normally the velocity must not be as high as cause erosion which can be reduced at the tube inlet by fitting plastic inert and yet be such that any solids are kept in suspensions.


## 2.1.3 HEAT EXCHANGER PERFORMANCE.

Two methods can be used to evaluate the performance of a flat plat heat exchanger. They are;

1. Effectiveness of number of transfer unit NTU.

2. The correction factor F to the log mean temperature difference.

1. EFFECTIVENESS APPROACH; effectiveness of a heat exchanger is the ratio of the actual heat transferred Q to the maximum possible heat transfer, Qmax.

$$\varepsilon = \frac{Q}{Q_{max}} = \frac{C_h \left( T_{hi} - T_{ho} \right)}{C_{min} \left( T_{hi} - T_{ci} \right)} \qquad 2.1$$

$$= \frac{C_c \left( T_{co} - T_{ci} \right)}{C_{min} \left( T_{hi} - T_{ci} \right)} \qquad 2.2$$

Where T is the mixing cup mean temperature, the subscripts i and o denote inlet and outlet, h denotes the cold stream. C is the capacity rate.

$$C = mCp \qquad 2.3$$

Where m is the mass flow rate and Cp is the fluid specific heat capacity. $C_{min}$ is equal to $C_h$ or $C_c$ whichever is smaller.

2. F-FACTOR APPROACH; we can also compare the heat transferred in our system to the heat transfer one would expect from pure countercurrent flow. We write the heat flux, q as

$$q = UAF\Delta T_{LM} \qquad 2.4$$

Where F is the correction factor and $\Delta T_{LM}$ is the log-mean temperature difference,

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln \left[ \frac{\Delta T_2}{\Delta T_1} \right]} \qquad 2.5$$

Where, for pure counter- current flow,

$$\Delta T_1 = T_{hi} - T_{co} \qquad 2.6$$

and,

$$\Delta T_2 = T_{ho} - T_{cc} \qquad\qquad\qquad 2.7$$

Equation 2.4 is the definition of the correction factor, F and it uses $\Delta T_{LM}$ as defined for pure counter-current flow, regardless of whether the heat exchanger being analyzed is in diameter-current or concurrent operation.

Outlet temperatures are very important in the evaluation of the effectiveness factor E, and F- factor of a heat exchanger. The outlet temperatures are obtained by modeling the temperatures everywhere throughout the heat exchanger as a function of thermal and flow conditions.

## 2.2 PLATE AND FIN EXCHANGER

Plate and fin heat exchangers used in motor, air craft and transport industries for many years are finding increased application in the processing industries and in particular in natural gas liquefaction cryogenic air separation, the production of olefins and in separation of hydrogen and carbon monoxide. Potential applications include ammonia production, offshore processing nuclear engineering and syngas production. As described by GREGORY the concept is that of flat plates of metal usually aluminum with corrugated metal, which not only holds the plates together but also acts as a secondary surface for heat transfer. Bars at the edges of the plates retain each fluid between adjacent plates and the space between each pair of plate apportioned to each fluid according to the heat transfer and pressure drop requirements is known as a layer. The heights of the bars and corrugations are standardized in the UK at 3.8, 6.35 and 8.9mm and typical designs are shown in fig 2.16



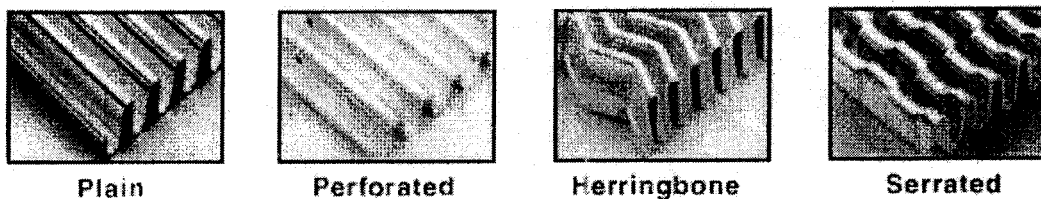| Plain | Perforated | Herringbone | Serrated |

Fig 2.16 typical fin designs.

There are four basic forms of corrugation; plan in which a sheet of metal is corrugated in the simplest way with fins at right angles to the plate fig.

Serrated where each cut is offset in relation to the proceeding fin fig.

Herringbone corrugation made by displacing the fin sideways every 9.5mm to give a zig-zig path fig.

And lastly perforated corrugation a term used for plain corrugation made from perforated materials.

Each stream to be heated or cool can have different corrugation heights, different corrugation types, different number of layers and different entry and exit points including part length entry and exit.

## 2.3 FINS (EXTENDED SURFACE)

In heat transfer processes involving resistances in series (such as in heat exchanger) there are situations where one of the resistances is entirely controlling due to a great in balance in the relatives magnitudes of the heat transfer co-efficient. This would occur for example, where there is heat exchange between a liquid (subcooled or undergoing phase change) and a gas. To eliminate such in balance, the surface on the high – resistance end could be 'extended' through the installation of fins, distributed so as to reduce the high resistance to a value that is approximately on a par with the other resistances.

In the study of heat transfer, a fin is a surface that extends from an object to increase the rate of heat transfer to or from the environment by increasing convection. The amount of conduction, convection or radiation of an object determines the amount of heat it transfers. Increasing the temperature difference between the object and the environment, increasing the convection heat transfer coefficient or in creasing the surface area of the object increases the heat transfer. Sometimes it is not economical or it is not feasible to change the first two options. Adding a fin to an object, however, increases the surface area and can sometimes be an economical solution to heat transfer problem.
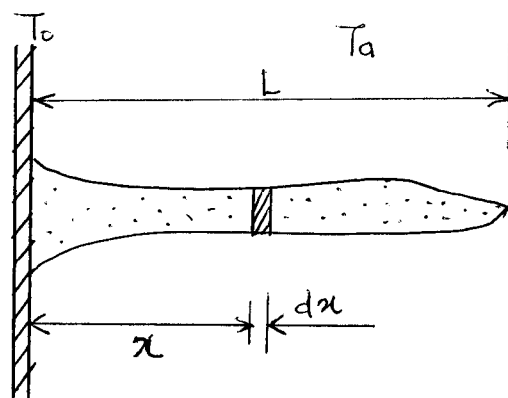


Fig 2.17 . SINGLE FIN.

Fig shows a single fin of arbitrary geometry and of length L, extending from a surface whose temperature is $T_0$ and "bathed" by a fluid at temperature $T_a$ Assuming constant temperature over

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any cross section of the fin and for steady-state operation, a heat balance over the differential element of the length dx yields

$$\frac{d}{dx}\left(kA_f \frac{dT}{dx}\right) = P\frac{ds}{dx}h_f(T - T_a)$$   2.18

Where $A_f$ is the local cross-sectional area of the fin, P is the local perimeter of the fin, K is the local thermal conductivity of the fin, $h_f$ is the local heat transfer coefficient between the fin and the surrounding fluid. Considering the situation where K and $h_f$ are held constant at some suitable values on which this research work is been carried out. Equation 2.18 then on rearrangement becomes,

$$\frac{d}{dx}\left(A_f \frac{dT}{dx}\right) = \frac{h_f}{k}P\frac{ds}{dx}(T - T_a)$$   2.19

Applying the boundary conditions.

1.  $T(x = 0) = T_o$

2.  $T(x = L) = T_L$

Or $\left.\frac{dT}{dx}\right|_{x=l} = 0$ , (insulated end)

Or $-k\left.\frac{dT}{dx}\right|_{x=l} = h_l(T_{x=l} - T_a)$

Where $h_L$ is the mean heat transfer coefficient over the end surface. The equations above can be solved through to obtain T (x) for any given fin geometry. The rate of heat dissipation, $Q_{Df}$ can then be determined from,

$$Q_{DF} = -kA_{fo}\left.\frac{dT}{dx}\right|_{x=0}$$   2.20

Or,

$$Q_{DF} = \int_{x=0}^{L} h_f(T - T_a)P\frac{ds}{dx}dx + h_l A_{fl}(T_l - T_a)$$   2.21

Fins are normally characterized by fin efficiency, $\eta_f$ and fin effectiveness $E_f$ the latter resulting in a corresponding extended surface effectiveness.

Fin efficiency is defined by,

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$$\eta_f = \frac{Q_{DF}}{Q_{DFO}} \qquad\qquad 2.22$$

Where $Q_{Dfo}$ is the heat dissipation rate that would be provided by the fin if its entire surface were at the base temperature, $T_o$.

$$Q_{DFO} = \left[h_f S_f + h_L A_{fL}\right]\left(T_o - T_a\right) \qquad\qquad 2.23$$

$S_f$ = total perimeter surface area of fin.

Fin effectiveness is defined by,

$$E_f = \underline{Q_{Df}} \qquad\qquad 2.24$$
$$Q_{Do}$$

Where $Q_{Do}$ is the heat dissipiation rate that would be obtained over the area ($A_{fo}$) covered by the fin on the surface if the fin were not present. $Q_{Do}$ is given by,

$$Q_{Do} = h_o A_{fo}(T_o - T_a) \qquad\qquad 2.25$$

$H_o$ is the heat transfer coefficient between the unfinned surface and the ambient fluid.

While the fin efficiency $\eta_f$ is always less than 1, fin effectiveness $E_f$ is normally greater than one (preferably much more greater than 1). Any design that makes $E_f < 1$ is undesirable as this would imply that the fin instead of reducing the resistance to heat flow, infact increases it due to an undesirable high conducive resistance. Fin effectiveness is given by,

$$\varepsilon_f = \frac{\left(h_f S_f + h_L A_{fL}\right)\eta_f}{h_o A_{fo}} \qquad\qquad 2.26$$

The extended surface effectiveness is defined by

$$Es = \underline{Q_D} \qquad\qquad 2.27$$
$$Q_o$$

Where $Q_D$ is the rate of heat dissipation over the entire extended surface, and $Q_o$ us the value of $Q_D$ if the surface were not extended. Having a surface whose total extended surface area is $A_o$ and contains $N_f$ fins. Then,

$$Q_D = [A_o + (E_f - 1) N_f A_{fo}]\, ho(T_o - T_a) \qquad\qquad 2.28$$

And

$$Q_o = A_o h_o (T_o - T_a) \qquad\qquad 2.29$$

Thus we have,

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$$\varepsilon_s = \frac{\left[A_o + \left(\varepsilon_f - 1\right)N_f A_{fo}\right]h_o\left(T_o - T_a\right)}{A_o h_o\left(T_o - T_a\right)} \qquad 2.30$$

Or $\qquad\qquad\qquad E_S = 1 = r_f(E_f - 1) \qquad\qquad\qquad 2.31$

Where, $\quad r_f = \dfrac{A_{fo} N_f}{A_o} \qquad\qquad\qquad\qquad 2.32$

is the fraction of the unextended surface area that is covered by the fins.

Fins of uniform cross section could be of various shapes triangular, circular or of any other shape. The circular gives the least heat dissipation area for a given volume of fin. For a maximum heat dissipation area, a flat fin of rectangular cross section is to be preferred. Introducing dimensionless temperature,

$$\theta = \frac{\left(T - T_a\right)}{\left(T_o - T_a\right)} \qquad\qquad 2.33$$

And dimensionless spatial coordinate,

$$\lambda = \frac{X}{L} \qquad\qquad\qquad 2.34$$

Since $A_f$ and $P$ are constant in this case and $\dfrac{ds}{dx} = 1$, equation 2.19 then becomes

$$\frac{d^2\theta}{d\lambda^2} - r^2\theta = 0 \qquad\qquad 2.35$$

Where, $\quad r^2 = \dfrac{Ph_f L^2}{kA_f} \qquad\qquad 2.36$

The boundary conditions becomes,

1. $\theta\,(\lambda = 0) = 1$

2. $\theta\,(\lambda = 0) = \theta_L$

Or $\quad \dfrac{d\theta}{d\lambda}\bigg|_{\lambda=1} = 0$

Or $\quad \dfrac{d\theta}{d\lambda}\bigg|_{\lambda=1} = \dfrac{-h_L L}{k}\theta_{\lambda=1}$

Integrating equation 2.35 we have,

$$\theta = C_1 \text{Sinhr}\lambda + C_2 \text{Coshr}\lambda \qquad\qquad 2.37$$

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Applying the boundary conditions given above,

$$Q_{DF} = \frac{h_f P(T_o - T_a)}{x \sinh \beta L + \cosh \beta L} \int_{x=0}^{L} [x \sinh \beta(L - x) + \cosh \beta(L - x)] dx + \frac{h_L A_f (T_0 - T_a)[x \sinh 0 + \cosh 0]}{x \sinh \beta L + \cosh \beta L}$$

Or,

$$Q_{DF} = \frac{h_f P \Phi (T_o - T_a)}{\beta} \qquad 2.49$$

$$\text{Where } \Phi = \frac{x \cosh \beta L + \sinh \beta L}{x \sinh \beta L + \cosh \beta L}$$

$$\Phi = \frac{x + \tanh \beta L}{1 + x \tanh \beta L} \qquad 2.50$$

$$\eta_f = \frac{\Phi}{\beta L_e} \qquad 2.51$$

$$\varepsilon_f = \frac{k \beta \Phi}{h_o} \qquad 2.52$$

FIN USES; Fins are most commonly used in heat exchanging devices such as car radiators, and heat exchanger in power plant. They are also used in newer technology such as hydrogen fuel cells. Nature has also taken advantage of the phenomena of fins, such as the ears of jack rabbit act as fins to release heat from the blood that flows through them.

## 2.4 BURNING FUEL (n-Decane)

| Formula | Molecular weight | Critical pressure | Critical temperature |
|---------|------------------|-------------------|----------------------|
| $C_{10}H_{22}$ | 142.29 | 304.21 psia | $652.1^0F$ |

Table 2.3

The hydrocarbon n- decane consists of ten carbons and twenty hydrogen. Classified as alkane, or paraffin hydrogen saturates the carbon atoms via covalent single bonds. Derived from the Latin term parum affinis; meaning "little affinity" for other compounds. Paraffin is known for its stability and resistance to reactivity. Alkanes such as decane are non-polar thus insoluble in polar solvent such as water. Alkanes may be separated into fractions via distillation. The lowest boiling point (3-4 carbons) is used as fuel in cigarette lighters and barbecues. Gasoline follows in the next distillation fraction (5-11 carbon) next kerosene and jet fuels (9-16 carbons), then diesel fuel (15-25 carbons) and the highest boiling point fractionation provides lubricants and greases

(26+carbons). The greater the branching of a hydrocarbon chain, the greater the stability of the molecule and the higher the boiling point. A colorless, odorless, non toxic, yet flammable gas, ethane is a constituent of natural gas and petroleum (75% methane, 25% Ethane, propane and butane). These "fossil fuels" were formed through the decomposition of organic matter over thousands of years and today provide a major energy source. Large amounts of the element may also be located in the atmospheres of Saturn and Jupiter

PROPERTIES.

The properties of n-Decane is shown in a tabular form in Table 2.4

| Pressure, psia | 14.7 | 50 | 100 |
|---|---|---|---|
| Temperature, $^0$F | 400 | 500 | 600 |
| Compressibility,(Z) | 0.949 | 0.884 | 0.841 |
| Enthalpy Btu/lb,(h) | 278.8 | 334.8 | 395.7 |
| Entropy Btu/lb-R(s) | 0.9756 | 1.0214 | 1.0734 |
| Specific heat Btu/mol-R(Cp) | 0.548 | 0.647 | 0.702 |
| K,(Cp/Cv) | 1.025 | 1.022 | 1.020 |
| Sonic velocity, ft/Sec | 528 | 520 | 520 |
| Specific volume, ft$^3$/lb | 4.19 | 1.28 | 0.0672 |
| Dynamic viscosity, lb/ft-Sec | $5.67 \times 10^{-6}$ | $6.46 \times 10^{-6}$ | $7.42 \times 10^{-6}$ |

Table 2.4.

## 3.0. METHODOLOGY

Material balance is carried out over the combustion chamber so as to obtain the amount in moles of the combustion reactants and product. After which an enthalpy balance was carried out so as to obtain the amount of heat accompanying the reaction and also the mount of heat generated by the combustion reaction in the combustion chamber ($\Delta H_R$ and Q respectively).

$$\sum_R \int_{298.15}^{298.15} nC_p dT + \Delta H_R + \sum_P \int_{298.15}^{550} nC_p dT + \Delta(kE) + \Delta(PE) = (-Q) - W_s$$

Since $\Delta(KE) = \Delta(PE) = Ws = 0$, as given by the process and the first integral term is also zero, the above equation then reduces to,

$$(-Q) = \Delta H_R + \sum_P \int_{298.15}^{550} nC_p dT$$

$\Delta H_R$ is given by, $\Delta H_R = \sum_P \Delta H_f - \sum_R \Delta H_f$    Applying

the heat transfer principle, fin effectiveness and extended surface effectiveness can be then be

calculated using, $\varepsilon_f = \dfrac{k\beta\Phi}{h_o}$, where $\Phi$ is obtained from $\Phi = \dfrac{x + \tanh \beta L}{1 + x \tanh \beta L}$, and $\beta = \sqrt{\dfrac{h_f P}{kA_f}}$ also

$x = \dfrac{h_L}{k\beta}$ extended surface effectiveness is then obtained from $\varepsilon_s = 1 + r_f(\varepsilon_f - 1)$,

where $r_f = \dfrac{A_{fo}N_f}{A_o}$. Rate of fuel consumption is obtained from relating amount of heat dissipated

by fin and then the heat generated from consumption chamber from the enthalpy balance. The heat dissipated by fin

is calculated from the expression $Es = \dfrac{Q_D}{Qo}$ , $Q_D = EsQ_{o\_}$ and $Q_{Do} = h_o A_{fo}(T_o - T_a)t$.

Equating both heats and eliminating Q gave the amount of fuel consumed by the boiler.

## 3.1. PROBLEM STATEMENT

The heat transfer surface of the combustion chamber of a boiler consists of 4  rectangular areas (2 pairs of dimensions 2mx1m  and 2mx2m) bearing 300  rectangular fins each of uniform thickness 2mm, length 2cm and running the entire 2m height of the chamber. The fins, whose end

are uninsulated, are made of aluminum (K=200w/mk). The combustion chamber is at or uniform temperature, equal to the temperature of the effluent gaseous products.

Saturated steam at a pressure of 20 bars is generated from saturated water feed at essentially the same pressure. The combustion fuel is n-decane and air (20mole% $O_2$ and free of $CO_2$, $H_2O$ and rare gases) is employed at a rate which is 20% in excess of the required for complete combustion both the oil and the air are fed in at $25^0C$. The combustion chamber operates at about atmospheric pressure and the product of combustion contains $CO_2$ and CO in a mole ratio of 9:1. Soot formation is negligible and the product of combustion contains a negligible quantity of n-decane vapour.

Careful measurement at steady state operation gave the temperature of the effluent gases to be $550^0k$ and that of the inside surface of the combustion chamber walls to be $490^0k$. Kinetic and potential energy changes are negligible for the process. Data on heats of formation and heat capacities are presented below in a tabular form. The following values of heat transfer co-efficient apply

Gas-to-wall = $20W/m^2K$

Gas-to-wall (perimetric surface) = $18 W/m^2K$

Gas-to-fin (end surface) = $16W/m^2K$.

Data on steam may be drawn from steam table

Data on Cp and $\Delta H_f$.

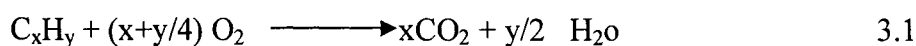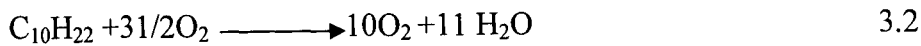| COMPOUND | Cp (Btu/1bmol$^0$F) {Note T=K} | $\Delta H_{f,298.15}$ (Cal/gmole) |
|---|---|---|
| $O_2$ | $6.148+0.003102T -903(10^{-7})T^2$ | 0.0 |
| $N_2$ | $6.524+001250T -1.0(10^{-9})T^2$ | 0.0 |
| CO | $6420+0.001665T-1.96(10^{-7})T^2$ | -29416 |
| $CO_2$ | $6.214+0.10396T -3.5(10^{-6})T^2$ | -94052 |
| $H_2O_{(g)}$ | $7.256+002298T-2.8(10^{-7})T^2$ | -57798 |
| $nC_{10}H_{22}$ | $10.357+0.173551T-5.4803(10^{-5})T^2$ | -59660 |

Table 3.1

Determine the fuel consumption rate.

## 3.2 THERMODYNAMICS OF THE COMBUSTION CHAMBER

Combustion of n-decane in the present of air free of $CO_2$, $H_2O$ and rare gases (Helium and neon) General combustion stoichiometry is given by.

$$C_xH_y + (x+y/4) O_2 \longrightarrow xCO_2 + y/2 \ H_2o \qquad\qquad 3.1$$

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$$C_{10}H_{22} + 31/2O_2 \longrightarrow 10O_2 + 11\,H_2O \qquad\qquad 3.2$$
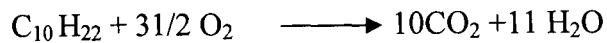
Equation 3.2 gives the stoichiometric equation of n-decane combustion.

## 3.2.1 MATERIAL BALANCE

Time basis: 1 second.

Enthalpy reference temperature: $25^0c = 298.15^0k$

Let fuel consumption rate be m kmole/S. According to the stoichiometry in equation 3.2.

$$C_{10}H_{22} + 31/2\;O_2 \longrightarrow 10CO_2 + 11\,H_2O$$

Reactant;  1 mole of n-decane

31/2 mole of oxygen is required for the combustion moles of oxygen in excess

(20%) =20/100 * 31/2 =3.1

Total number of oxygen present = 31/2+3.1 = 18.60

Moles of Nitrogen entering. Since the air contains 20%

Oxygen and is free from $CO_2$ and rare gases. It implies that 80% will be nitrogen.

Therefore moles of nitrogen entering will be 80/20 x18.6= 74.4    moles    of

nitrogen.

Products;  20 moles of $CO_2$, but since $CO_2$ is in mole ratio of 9:1 with CO, it

implies that there will be 9moles of $CO_2$ and 1 mole of CO

74.4  Moles of Nitrogen.

20/100x18.6 =3.6 moles of $O_2$.

Then the actual reaction is defined by.

$$mC_{10}H_{22} + 18.6mO_2 + 74.4mN_2 \longrightarrow \underline{9mCO_2 + mCO + 11mH_2O + 3.6mO_2 + 74.4mN_2} \quad 3.3$$
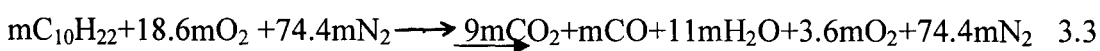
The diagrammatic representation of the process is shown in fig 3.2



Fig 3.2. BLOCK DIAGRAM OF MATERIAL BALANCE.

## 3.2.2 ENTHALPY BALANCE

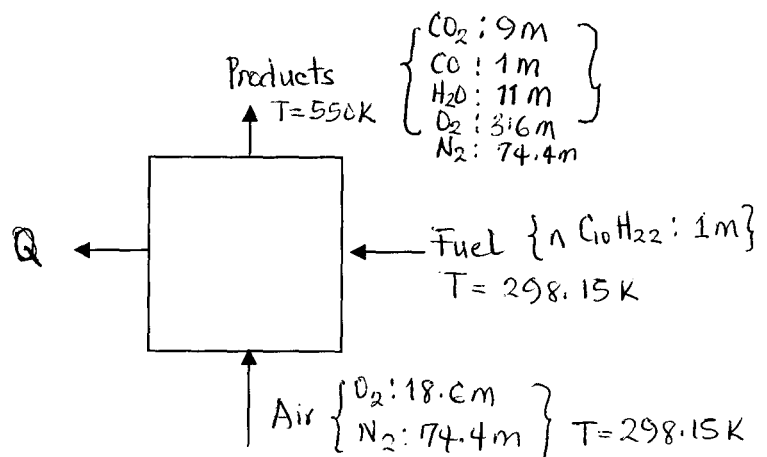The data on heat of formation were given at a temperature of $298.15^0K$ it will be convenient to choose the process of enthalpy change to occur along a path involving reaction at $298.15^0K$. If the accompanying heat of reaction be $\Delta H_R$. then, the enthalpy balance is given by,

$$\sum_R \int_{298.15}^{298.15} nC_p dT + \Delta H_R + \sum_P \int_{298.15}^{550} nC_p dT + \Delta(kE) + \Delta(PE) = \left(-Q\right) - W_s \qquad 3.4$$

Since $\Delta(KE) = \Delta(PE) = Ws = 0$, as given by the process and the first integral term is also zero, equation 3.4 then reduces to,

$$\left(-Q\right) = \Delta H_R + \sum_P \int_{298.15}^{550} nC_p dT \qquad 3.5$$

$\Delta H_R$ is given by,

$$\Delta H_R = \sum_P \Delta H_f - \sum_R \Delta H_f \qquad 3.6$$

$$= m\,[(9x-94052) + (1x-26416) + (11x\,-57798) - (1x-59660)]\,\underline{kmole.\,Cal}$$
$$gmole$$

$$= -1449002\,\underline{kmole.cal}\ x\ \underline{1000gmole}\ x\ \underline{4.1868J}$$
$$gmol \qquad\qquad kmole \qquad\quad cal$$

$$\Delta H_R = -6065533372m\,J$$

Let us write $Cp = a + bT + cT^2$, where $T = {}^0K$ and $Cp = Cal/gmoleK$.

Note $1Btu/lbmole^0F = 1\ Cal/gmole^0K$

Then, $\displaystyle\sum_P \int_{298.15}^{550} nC_p T = \sum_P \int_{298.15}^{550} (na + nbT + ncT^2)dT$

$$= (550 - 298.15)\sum_P na + \frac{1}{2}(550^2 - 298.15^2)\sum_P nb + \frac{1}{3}(550^3 - 298.15^3)\sum_P nc$$

$= (550 - 298.15)\,[(9x\,6.214) + (1x\,6.420) + (11x7.256) + (3.6x6.148) + (74.4x6.524)] + 0.5(550^2 - 298.15^2)\,[(9x0.010396) + (1x0.001665) + (11x0.002298) + (3.6x0.003102) + (74.4x0.01250)] + \frac{1}{3}(550^3 - 298.15^3)\,[(9x-3545(10^{-6})) + (1x-1.96(10^{-7})) + (11x-2.83(10^{-7})) + (3.6x-9.23(10^{-7})) + (74.4x-1.0(10^{-9}))]\,mCal/gmole.K\ x\ kMole.K/1\ x\ 1000gmole/kmole\ x\ 4.1868J/Cal$

$$= 77,981,798.8mJ$$
$$-Q = (-60065522372m + 777981798.8.)J$$

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$$Q = 5287540573.2mJ$$

## 3.3. HEAT TRANSFER ANALYSIS OF THE BOILER.

### 3.3.1. COMPUTATION OF HEAT EFFECTIVENESS.

Data Given.

Heat transfer coefficient of gas-to-fin (perimetric surface) $h_f$ = 18 W/m²K

Heat transfer coefficient of gas-to-fin (end surface) $h_L$ = 16W/m²K

Heat transfer coeffient of gas-to-wall ho =20 W/m²K

Thermal conductivity of Aluminum fin K = 200 W/mk.

Length of fin L= 2cm = 0.002m

Height of H = 2m

Perimeter of fin P = 2(2.0+0.002) = 4. 004m.

Area of fin $A_f$ = 2x 0.002 = 004m²

The relationship as shown in the previous chapter between fin effectiveness $E_f$ and $\Phi$, apply directly in spite of the fact that in this case, Ta> T≥ To this is in view of the fact that θ remains positive and the corrective boundary constraint is unchanged.

Therefore, $\Phi$ = $\dfrac{XCosh\beta L +Sinh\beta L}{XSinh \beta L +Cosh \beta L}$ = $\dfrac{X+Tanh \beta L}{1+XTanh BL}$

$$\beta^2 = \frac{h_f P}{KA_f} = \frac{18x4.004}{200x0004} = 90.09m^{-2}$$

$$\beta =\sqrt{90.09}m^{-2} = 9.4915752m^{-1}$$

$$X = \frac{h_L}{K\beta} = \frac{16}{200x9415752} = 0.0084285.$$

$$\beta L = 9.4915752 x0.02 =0.1898315.$$

$$\Phi = \frac{0.0084285+Tanh\ 0.1898315}{1+0.0084285xTanh\ 0.189815}$$

Evaluating Tanh 0.1898315, = $\dfrac{e0.1898515 -e-0.1898315}{e018983115 +e-018983115}$

= 0.18758364

$$Q = \frac{0.0084285+0.18758364}{1+0.0084285\{0.18758364\}} = 0.1957027$$

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$$E_f = \frac{K\beta\Phi}{h_o}$$

$$= \frac{200 \times 9.04915752 \times 0.1957027}{20} = 18.5752689.$$

$$E_f = 18.5752689.$$

## 3.3.2   COMPUTATION OF EXTENDED SURFACE EFFECTIVENESS

The fraction of the surface over laid as shown in the previous chapter is given by.

$$r_f = \frac{A_{fo} N_f}{Ao}, \qquad N_f = \text{number of fins} = 300$$

$$= \frac{0.004 \times 300}{2[2 \times 1 + 2 \times 2]} = 0.1$$

Hence extended surface effectiveness $E_s = 1 + r_f (E_f - 1)$

$$E_s = 1 + 0.1[18.5752689 - 1]$$

$$E_s = 2.7575269.$$

## 3.3.3. DETERMINATION OF RATE OF HEAT DISSIPATION

The rate of heat dissipation as shown in previous chapter is given by $Q_o = h_o A_o (T_a - T_o)t$ multiplying this equation by the time basis of 1 second gives the rate of heat dissipation in second of the surface assuming it were not extended. .

$$Q_o = h_o A_o (T_a - T_o)t$$

$$= 20 \times [2 \times 1 + 2 \times 2] \times [550 - 490] \times 1 \text{ J}$$

$$= 14400 \text{J}.$$

From the expression of extended surface effectiveness in the previous chapters equation. A relationship exists between the extended surface effectiveness, rate of heat dissipation of the surface assuming it were not extended and rate of heat dissipation of the extended surface given by

$$E_s = \frac{Q_D}{Qo}$$

From this expression rate of heat dissipation of the extended surface, this gives the rate of heat dissipated by the boiler.

$$Q_D = E_s \times Qo$$

$$= 2.7575269 \times 14400 \text{J}$$

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=39708.38736J.

## 3.4    DETERMINATION OF FUEL CONSUMPTION RATE

Amount of fuel consumed is obtained by equating the amount of heat generated in the combustion chamber and that dissipated by the boiler and eliminating Q. we have

$$5287540573.2m = 39708.38736$$

$$m = \frac{39708.38736}{5287540573.2} = 7.509803 \times 10^{-6} \frac{kmole}{S}$$

$$= 7.509803 \times 10^{-6} \frac{kmole}{S} \times 142 \frac{kg}{kmole} \times 3600 \frac{S}{hr} \times 24 \frac{hr}{day}$$

$$= 92.1363 kg/day$$

This gives the fuel consumption rate of the boiler to be 92.1363kg/day.

## 4.0 RESULTS

### 4.1 Results from thermodynamics of the combustion chamber.

### 4.1.1. Amount in moles of reactants and products of the combustion reaction obtained from the material balance.

| Reactants at $298.25^0K$ | Amount(moles) |
|---|---|
| Fuel (n-Decane) $C_{10}H_{22}$ | 1 |
| Air ; oxygen $O_2$ | 18.6 |
| Nitrogen $N_2$ | 74.4 |

Table 4.1

| Products at $550^0K$ | Amount(moles) |
|---|---|
| Carbon dioxide $CO_2$ | 9 |
| Carbon monoxide CO | 1 |
| Water $H_2O$ | 11 |
| Oxygen $O_2$ | 3.6 |
| Nitrogen $N_2$ | 74.4 |

Table 4.2.

### 4.1.2. Amount of heat obtained from enthalpy balance.

| Type of heat | Amount (J) |
|---|---|
| Accompanying heat of reaction $\Delta H_R$ | -6065522372m |
| $\sum\limits_{P} \int\limits_{298.15}^{550} nC_p dT$ | 777981798.8m |
| Heat generated from the combustion chamber | 5287540573.2m |

Table 4.3

### 4.2 Results obtained form the heat transfer analysis of the boiler.

Dimensions.

Height of fin, $H_f$=2m

Length of fin, L =0.002m

Thermal conductivity of fin, K =200W/mK

Number of fin, $N_f$=300

Heat transfer coefficient gas-to-fin (perimetric surface), $h_f$=18W/m$^2$K.

Heat transfer coefficient gas-to-fin (end surface), $h_L$=16W/m$^2$K.

Heat transfer coefficient gas-to-wall, ho=20 W/m$^2$K.

Perimeter of fin P=2 (2.0+ 0.002) = 4.004m.

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Area of fin, $A_f = 2 \times 0.002 = 0.004 m^2$.

Total unextended surface area, $A_0 = (2 \times 1 + 2 \times 2) = 6 m^2$.

### 4.2.1 Computation of fin effectiveness ($E_f$) results.

| Parameter | Value |
|---|---|
| $\beta = \sqrt{\dfrac{h_f P}{kA_f}}$ | $9.4915752 m^{-1}$ |
| $x = \dfrac{h_L}{k\beta}$ | 0.0084285 |
| $\beta L$ | 0.1898315 |
| $\Phi = \dfrac{x + \tanh \beta L}{1 + x \tanh \beta L}$ | 0.1957027 |
| $\varepsilon_f = \dfrac{k\beta\Phi}{h_o}$ | 18.5752689 |

Table 4.4

### 4.2.2 Results of computation of extended surface effectiveness Es.

| Parameter | $r_f = \dfrac{A_{fo} N_f}{A_o}$ | $\varepsilon_s = 1 + r_f(\varepsilon_f - 1)$ |
|---|---|---|
| Value | 0.1 | 2.7575269 |

Table 4.5

### 4.2.3. Rate of heat dissipation over the entire extended surface obtained.

Amount of heat dissipated if the surface were not extended is given by $Q_0 = h_o A_o (T_a - T_o)t$ as shown in the previous chapter. The reference time t is taken to be 1 Secs

| Parameter | Qo = hoAo (Ta-To)t  (J) | $Q_D$ =EsQo (J) |
|---|---|---|
| Value | 14400 | 39708.38736 |

Table 4.6

### 4.3. Fuel consumption rate of the heat exchanger (boiler).

Amount of fuel consumed per day is then obtained by eliminating Q from heat dissipated by the boiler and heat generated from the combustion chamber.

$$5287540573.2 mJ = 39708.38736 J$$

$$m = \frac{39708.38736}{5287540573.2} = 7.509803 \times 10^{-6} \text{ Kmole/S}$$

Amount of fuel consumed in Kg/day was the calculated to be

$$7.509803 \times 10^{-6} \frac{\text{Kmole}}{\text{S}} \quad \times \quad \frac{142 \text{Kg}}{\text{Kmole}} \times \frac{3600 \text{S}}{\text{hr}} \times \frac{24 \text{hr}}{\text{day}}$$

$$= 92.1363 \text{kg/day}.$$

Total amount of fuel consumed by the heat exchanger (boiler) per day is then 92.1363kg/day.

## 4.4 DISCUSSION ON RESULTS.

The amount of oxygen in the air that was used for the combustion reaction was not totally consumed in the reaction, 3.6 moles appears in the product this indicates incomplete combustion, the presence of 1 mole of carbon monoxide confirms incomplete combustion. Since it is in a small amount soot formation a product of incomplete combustion was neglected. The amount of nitrogen in the reactant was fully recovered in the product as shown in Table 4.1 and 4.2 ; indicating that did not take part in the combustion reaction hence no harmful nitrogenous compound were formed in the effluent that needs to be treated.

The accompanying heat of reaction obtained from the enthalpy balance gave a negative value -6065522372J indicating that heat was given out hence the reaction is exothermic and its summation with the total sum of the integral of the product of all specific heat capacities of their number of moles gave the total amount of heat generated to be 5287540573.2J by the combustion process which heats the heat exchanger.

Heat transfer analysis of the heat exchanger (boiler) gave the effectiveness of the fins within the boiler and the extended surface effectiveness to be 18.5752689 and 2.7575269 respectively which were used to compute the amount of heat dissipated by the heat exchanger (boiler) given as 39708.38736. Establishing a relationship between the amounts of heat generated from the combustion chamber and that dissipated by the heat exchanger (boiler) gave the amount or rate of fuel consumption in kg/day of the heat exchanger to be 92.1363kg/day.

# CHAPTER FIVE

## 5.0 CONCLUSION

The analytical study carried out in this research work reveals in detail the heating processes occurring within a heat exchanger of which a boiler was considered. The boiler has fin effectiveness $E_f$ of 18.5752689, extended surface effectiveness Es of 2.7575269, and amount heat dissipated to be 39708.38736J. The boiler consumes n-Decane fuel at a rat of 92.1363kg/day to produce a temperature of $550^0K$ for heating the content of the boiler.

## 5.1 RECOMMENDATION

Based on some observation in the carrying out of this research work the following recommendations were made.

1. This project work should be extended to chemical process industries where heat exchangers (boilers) are used so as to enable effective utilization of the research work in rating the performance of their boilers.

2. this research project can be used as a guide to evaluate and standardize heat exchangers (boilers) which values in turn can be used to detect any malfunction in the heat exchanger's (boiler) operation.

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