Preliminary Design of a Heat Recovery Boiler for 100-Man Houseboat

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ABSTRACT: This research work presents a detailed work on the preliminary design, of a heat recovery boiler for a 100-man houseboat. The results obtained from this work shows that the following factors must be duly considered; the operating temperatures and pressure, the quality of steam, layout of heating surfaces, heating surface requirements, circulation of steam and water, capacity of boiler drum, materials and method of construction were obtained. To design an economically sound boiler, proper evaluation of the factors influencing the design and installation of steam generating equipment, to produce the required quantity of steam at the designed temperatures and pressure were taken into consideration. The design a boiler looked into the "must fit easily" property and the minimum engine room space factor. Also considered in the design is the accessible, safe operation, inspection and maintenance of the boiler. A mathematical model software was used to validate model equations and also check the effectiveness of the boiler at the various loads and designed parameters.

KEYWORDS: Boiler, House Boat, Temperature, Pressure, Steam, Tubes, Pump.

I. INTRODUCTION

The boiler is an equipment that generates steam on board ships. The steam generated is mostly used for heating of cargoes and cabins, to generate distilled water and even to power the vessel if the prime mover is a steam turbine.All marine heat recovery boilers are made of at least five major components which includes the burner, the steam drum, the steam generating or bank tubes, the feed water pump and the super heater. The burner which is located in the furnace is the heat source of the boiler. The steam drum or water drum may also be installed. The water drum is usually connected to the steam drum by the water screen tube and the generating tube provides a reservoir for holding water in the boiler while the steam drum which contains water and provide a space where steam is accumulated then drained into the super heater or output line. The feed water pump provides water for regulation at a rate that match with the production of steam. The use of air heaters and economizers are also possible to further improve the overall efficiency of the boiler [1].

Many factors influence the design and maintenance of steam generating equipment to produce the required quantity of steam at the design pressure and temperature for a particular boiler installation. Also, another factor of great concern is the fuel cost per shaft horse power. In establishing the characteristics of the boiler installation and whether or not the installation is economically sound, the fuel consumption can be decreased by the use of higher steam pressure and temperature or a more sophisticated cycle can be employed by the use of reheating, economizer etc.As a designer of marine heat recovery boiler, you must consider factors such as the initial cost, maintenance cost, and weight and space requirements within the engine room versus the savings resulting from increased thermal efficiency. During boiler operation as steam pressures increases, it is essential to use additional heat reclaiming equipment in the boiler unit, this is because of the corresponding increase in saturated team temperature, which can result in higher gas temperature leaving the boiler blank thereby reducing the boiler efficiency at a given firing rate. To determine the required quantities of steam and feed water flow, there must be a detailed heat balance prepared[2].

Most marine boilers are oil fired. At sea tankers designed to carry liquefied natural gas LNG may use the natural boil off from their cargo gas tanks as supplemental fuel, then the cargo gas boil- off is collected and pumped to the boiler where it is burnt either by itself or in combination with oil. The quantity of boil off available from the liquefied natural gas is a function of ambient sea and air temperatures, the ships motion and the cargo loading, among other things and may vary from day to day.

The fundamental boiler design problem helps to determine the proportion of the various heats absorbing surface to utilize the maximum heat available in the product of combustion. This research is aimed at designing a heat recovery boiler for a 100-man houseboat that will provide the needed steam services for the crew.The overall objective of this study is to examine the factors influencing the design

and installation of steam generating equipment to produce the required quantities of steam at the design temperature and pressure. Arising from the overall objective are the following specific objectives; to design a boiler that must fit easily and conveniently within a minimum of engine room space, to design a boiler that will be accessible for operation, inspection and maintenance though light in weight, to design a boiler that will be sufficiently dependable under adverse sea conditions and finally, the heat recovery boiler must meet the rules and regulations of the regulatory bodies.

Boilers have become a source of many serious injuries and property destruction due to poorly understood engineering principles. Thin and brittle metal shell can rupture, while poorly welded or riveted seams could open up, leading to a violent eruption of the pressurized steam. When water is converted to steam, it expands to over 1,000 times its original volume and travel down the steam pipes at over 100 kilometres per hour. If a leakage occurs in the steam supply lines hat is larger than what the make-up water supply could replace, then a violent explosion can take place.Therefore, this design work is aimed at addressing certain mishaps in the design of boilers. Further, integrated mathematical models and engineering software such as MAT LAB etc. is used to determine how much space a boiler will need and the type of material to be used in the design work which must conform to safety regulations [3].

1.1. Boiler Terms and Definitions

For an understanding of marine heat recovery boiler technology, a review of the applicable terms and definitions of various essential boiler parts may be helpful. The following terms and definitions are based on the standards of the American Boiler Manufacturers Association and on day-to-day usage.

- **Air (per) heater:** The heat-transfer apparatus through which air is passed and heated by a medium of higher temperature**,** such as the products of combustion or steam.
- **Apparatus:** For reducing and controlling the temperature of a super heater vapour.
- **Brick pan:** The plate and structural steed work which supports the furnace floor.
- **Brick work:** The refractory linings of the furnace.
- **Casing**: The covering of metal plates and structure used to enclose all or a portion of a steam generator unit.
- **Chemical feed pipe:** A pipe inside boiler drum through which chemicals for treating the boiler water are introduced.

- **Circulation ratio:** The ratio of water entering a circuit to the steam generated within that circuit.
- **Down-comer:** A tube in a boiler or water wall system through which fluid flows downward.
- **Dry pipe:** A perforated or slotted pipe or box inside the steam drums which is connected to the steam outlet
- **Economizer**: A heat recovery device designed to transfer heat from the products of combustion to a fluid, usually feed water.
- **Feed pipe:** The pipe used to distribute the feed water inside the boiler steam
- **Fire tube:** A tube in a boiler having water on the outside and carrying the products of combustion on the inside.
- **Floor tubes:** Those tubes in the furnace floor which is exposed to the products of combustion are generating to the products of combustion are generated tubes but if arranged beneath refractory are used as supply tubes to supply water to a drum or header.
- **Forced circulation:** Circulation in a boiler by mechanical means external to the boiler
- **Furnace screen:** One or more rows of tubes arranged across the furnace gas outlet.
- **Furnace volume:** The volume contents of the furnace combustion chamber
- **Generating tube:** A tube in which steam is generated
- **Reader**: A drum too small to permit entry through a manhole.
- **Heat release:** The total quantity of thermal energy above a fixed datum introduced into a furnace by the fuel. It is considered to be the product of the fuel delivered per hour and the higher heating value in Btu per volume
- **Heated down-comer:** Any tube in a loss generation bank which water may be from the steam drum the water drum or header.
- **Heating surface:** That surface which exposed to the heating medium for absorption and transfer of heat the heated medium including any fins, studs, etc. attached to the outside of the tube for the purpose of increasing the heating surface per unit length of tube.
- **Ligament (tube):** The minimum distance between two adjusted tubes.
- **Moisture-in-steam:** Particles of eater carried in steam, usually expressed as the percentage by weight.
- **Mud, lower, or water drum:** A pressure chamber is a drum or header type located at the lower end of a water feed boiler convection has which is normally provided with a blow valve for

periodic removal of sediment collecting in the bottom of the drum.

- **Natural circulation:** Circulation of water in a boiler caused by the difference in density between the water in the down comers and the water-steam mixture in the generating tubes.
- **Radiant heat absorbing surface (RHAS):** The projected area of tubes and extended metallic surfaces as steam baffling viewed from the furnace. Included are the planes of the furnace exit screen.
- **Reheater:** Heat transfer apparatus for heating steam after it has given up some of its original heat in doing work.
- **Riser**: Tube through which steam and water passes from an upper water wall header to the steam drum.
- **Steam baffling:** The plate's centrifugal separators or baffles arranged to remove entrained water. **Steam or steam-and water drum:** A pressure chamber located at the upper extremity of a boiler circulatory system in which the steam generated in the boiler is separated from the water and from which steam is discharged at a position above a water level maintained therein.
- **Super heater:** A group of tubes which absorbs heat from the products of combustion to raise the temperature of the vapour passing through the tubes above the saturation temperature corresponding to its pressure.
- **Tangent-tube wall:** A water wall in which the tubes are substantially adjacent to each other with practically no space between the tubes

Starting from the dawn of industrial revolution the attention of many individuals focused on the advantage of powering ships by steam. A careful study of the history of early marine boilers reveals that early designers and engineers did not lack novel and ingenuous ideas for steam propulsion equipment. However, they did lack the material and machine tools with which to implement these ideas. The development of boats powered by steam began in the United States[4].

Robert Fulton and Clermont (1807) was the first person to inaugurate steam navigation and this idea was popularly called Fulton's folly. The success of this vessel prompted others to follow Fulton's lead and soon steam boats were navigating all the great waterways of the North American continent and a new industry was born.Main engine on large or ultra-large merchant ships are powerful and consume large amount of fuel oil. A slight increase of their thermal efficiency could bring round-sum benefit for ship owners. Meanwhile, with the rising fuel costs and global warming ongoing, energy saving and emission- reduction have been getting more and more highlighted. Based on this consideration [5]

Convectional waste heat recovery systems on-board are cogeneration system i.e. combined heat and power generation system, and usually use water as working fluid in Rankine Cycles (RCs) when analysing these systems based on thermodynamic laws, effect of pinch point are always paid much attention. Butcher and Reddyanalysed the effects of different specific heat models, gas inlet temperature and other operating parameters on entropy generation rate and the second thermodynamics law efficiency for a waste heat recovery-based power generation system. They found that, approximating the exhaust gas underestimation of power plant performance on both first thermodynamic law and second thermodynamic law point of view. Actual gas composition models and specific heat models should be used to predict second thermodynamic law performance more accurately [6][7].

Compared with convectional waste heat recovery systems on-board, Organic Rankine Cycles (ORC) employed organic working fluid in steam power cycles, ORCs possess several notable advantages for economical utilization of energy resources, small system applications and environmental impacts and can give better performance to recover low grade waste heat on-board. Yalcin Durmusoglu et al, designed an ORC cogeneration system for a container ship and employed three performance criterions, i.e., energy utilization factor, artificial thermal efficiency and exergy efficiency, to evaluate system performance. Thermodynamic results indicated that ORC cogeneration systems. [8]

Ships are always the most efficient transportation means when important environment aspects are taken into account. Ships normally have enough waste heat (both energy quality) potential to be utilized and cover almost all demands for thermal energy on board. Cogeneration systems adapt to large and ultra-large ships. However, for small ships and other waste heat sources, thermoelectric generators (TEGs) represent a promising solution in that direction.Kristiansen et al analysed the waste heat sources on a median size bulker ship equipping a main engine of 7.8MW and found that incinerators were most promising application for TEGs. Moreover, important factor for successful TEGs applications were commercialization of material

research and reduced price/watt power output TEGs can bring [9].

In available literatures, the minimum value of heat recovery boiler working pressure for different types of ships, 0.7MPa, was normally selected to calculate advantages of waste heat recovery systems. Optimal working pressure of exhaust gas recovery boiler is seldom analyzed when installing a waste heat recovery system designed for a given ship. Superheated steam yield, total electric power yield, first law efficiency, second law efficiency at different exhaust gas boiler working pressure are seldom analyzed. As well, effects of feed water temperature and steam turbine back pressure on system exergy efficiency are seldom studied. In this paper, a WHRS is proposed to recover waste heat of main engine 9K98ME-C7. Superheated steam yield, total electric power yield, first law efficiency, second law efficiency at different heat recovery boiler working pressure are thermodynamically analyzed.

In addition, exergy efficiency is modeled under different feed water temperature and steam turbine back pressure. For a better understanding of such WHR solution, a comparison to results in published literatures is also done by an exhaust gas bypass system which could increase exhaust gas temperature by approximately 50° and specific fuel oil consumption (SFOC) slightly by about2g/(kW/h).When choosing waste heat recovery systems, payback time, electricity yield, system size, maintenance cost should be taken into account since waste heat recovery systems are rather expensive and appropriate only for large merchant ships. MANB&W Dieselrecommends single-pressure steam turbine system when installing combined turbines. [10]

Currently, there are three different types of exhaust gas waste heat recovery systems.Power Turbine Generator (PTG), PTG system is small, simple and can produce power approximately 4% of the main engine Specified Maximum Continuous Rating (SMCR). The exhaust gas expands in the power turbine while its temperature decreases. The enthalpy drops of the exhaust gas transforms into turbine's kinetic energy. The generator connects to the turbine via a gearbox. PTG system has been applied on some container ships and LNG ships.Steam turbine generator (STG), STG system is relatively complicated and can produce power equivalent toapproximately5%to7%ofthemainengineSMCR.Steam turbine is driven by superheated steam. Steam is

produced by a large heat recovery boiler installed on the main engine exhaust gas piping system. The enthalpy drops of the superheated steam transforms into turbine's kinetic energy. The generator connects to the turbine via a gearbox.Combined Turbines system is a combination of the above two systems. The power turbine is connected to the steam turbine via a gearbox and the steam turbine is further connected to a large generator, which absorbs the power from both turbines. Such system is most complicated and can produce power equivalent to approximately 10% of the main engine SMCR.

Modern heat recovery boilers currently are still in use today with the same principle of operation as the first-generation boilers, but differ markedly in their design. With advancement in technology various features and modification have been introduced to name a few, water cooled brick walls have been introduced to cut down maintenance of refractory materials, were complex cycles such as reheat cycles are employed by the ships power plant, reheater sections have been built into boilers to meet this need, even attemperators or desuperheaters have been built into boiler to enable same boiler provide steam for both the prime mover and for auxiliary usage. In an effort to cut down size, bulk forced circulation has been built and even supercharging of the furnace have been made by utilising combustion air at pressures greater than atmospheric [11]. These modifications have led to a variety of boilers such as the two drum D type boiler, the reheat boiler and the super charged or Velux boiler in modern times whose principle of operation and pattern of design will be duly considered in this research work.

II. METHOD
The design of marine boiler is a series of complex and interrelated step which commences with setting assumptions on a desired efficiency (based on owners requirements bearing in mind the practicability of achieving such an efficiency) and working towards achieving this efficiency, while ensuring that the first cost and total operating cost are kept at a minimum level without compromising the safety of the equipment or that of the personnel who may operate it, and without infringing upon the rules of the classification society.In the series of steps taken for the design of a boiler, the following components are generally considered:
- Fuel burning equipment
- Furnace (volume housing the burner in which combustion takes place)
- Boiler generating bank

- Super heater (and preheater where applicable)
- Economizer and air heater where applicable
- Attemperators (or control) and auxiliary desuperheater
- Circulatory and steam separating system
- Casing and setting
- Cleaning equipment
- Safety valve and other mounting
- Feed water treatment
- Foundation and support
- Combustion and supply system
- Uptake gas duct system and stack

In the research considerations are made at the fuel burning equipment, furnace, boiler tube bank, super heaters (and preheater where applicable) and the economizer (air heater where applicable) with broad emphasis on the performance/efficiencies of these steam generator systems using analytical method.

2.1. Boiler Design Steps
These considerations require many interrelated steps; in most cases a good number of assumptions must be made in other to initiate the design

The first step is the selection of the basic type of boiler, super heaters and economizer or air heater (or both) to be used. This selection is based in part on preference and in the space available for the installation and its operating requirements. The quantity of fuel required is determined from the steam generator efficiency, the given steam pressure, temperature and the heating value of fuel.

The fuel characteristics and quantities determine the right burning equipment to be employed. This in turn sets the air requirements. Combustion calculations are made to determine the hourly quantity of flue gas passing through the unit. The exit or stack gas temperature to which the flue gas must be cooled to achieve the desired efficiency is determined.

The furnace exit gas temperature is next calculated, its value is dependent on the radiant and convectional heat transfer surface installed in the water walls, floor, roof and screen (radiant only) as well as the extent of refractory present.

Next the gas temperature drops and the heat absorbed by the screen and super heaters are determined. The size and the spacing of tubes and the amount of surface are calculated. The boiler burner, economizer, and air heater, surface is then sized to provide the final uptake gas temperature required. Also, choice of

thickness and types of material for tubes, headers and drums are made.

Pressure drop of water and steam through all component from the economizer feed water inlet to the super heater outlet are next computed. They in turn, establish the required boiler and economizer design pressures and the safety valve settings. The foregoing steps are followed for each preliminary design of a marine heat recovery boiler. [4][11].

2.2. Heat Transfer Characteristics of the Boiler
All the three modes of heat transfer: radiation, convention and conduction enter into the equation either alone or in a combination during theoretical calculations of water-walls, super heaters, economizers and air heaters.Radiation heat transfer is prevalent in the cone (hottest part) of the furnace and the transfer of radiant energy to the boiler tubes is dependent on the mass flow rate of the exhaust gas and the amount of the heat absorbing surface exposed [12]. The radiant heat absorbed by the water walls can be calculated from;

$$q_{rad} = E\, A\, F\, \sigma\left[(T_1)^4 - (T_2)^4\right] \quad (1)$$

$$= 5.67\,(E)\, AF\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] 10$$

Where:

E = emissivity of the flame

A = area of cross-section of radiant heat absorbing surface

F = view-factor

σ = Stefan-boltmann constant = 5.67×10^{-8} $10/m^2k^4$

T_1 = absolute temperature of the flame ^0k

T_2 = absolute temperature of radiant heat absorbing surface

The coefficient of radiative heat transfer is given by

$$h_{rad} = q_{rad}/_{A\Delta\theta} = \frac{5.67.E.F\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]}{\Delta\theta} \quad (2)$$

- **Radiative Resistance**

$$R_{rad} = \frac{1}{h_{rad}A} = \frac{\Delta\theta}{5.67.E.F.A}\left[\left(\frac{T_1}{100}\right)^4 - T_2100\right]4^O KW \quad (3)$$

Heat transfer in the second vertical shift and the horizontal duct of the furnace takes place entirely by convention. The rate of the heat transfer from the hot exhaust gases to the heat absorbing surfaces is given by

$$q_{conv} = h_{conv} A \Delta\theta, W$$
(4)

where;

h_{conv} = coefficient of convective heat transfer w/m^2 ^{0}k

A = area of heat absorbing surface, m^2

$\Delta\theta$ = temperature difference between hot gaseous products of combustion and the tube walls ^{0}k

- **Convective Resistance**

$$R_{conv} = R_{conv} = \frac{1}{h_{conv}A} \qquad ^0k/w$$
(5)

Heat transfer by mode of conduction takes place through the wall thickness of the tubes as well as across the scale or depositions on both the inside and upside of the tube surface. The rate of conductive heat transfer through a wall is given by

$$q_{cond} = KA\frac{\Delta\theta}{\Delta x} \qquad (6)$$

For composite wall

$$q_{cond} = K_1 A \left[\frac{\Delta\theta}{\Delta x}\right]_1 + K_2 A \left[\frac{\Delta\theta}{\Delta x}\right]_2 + K_3 A \left[\frac{\Delta\theta}{\Delta x}\right]_3$$
(7)

And the conductive resistance is

$$R_{cond} = \frac{1}{A}\sum\frac{\Delta x}{K}$$
(8)

Where;

K_1 = thermal conductivity of the scale W/m ^{0}k

K_2 = thermal conductivity of the tube walls W/m ^{0}k

K_3 = thermal conductivity of slag deposition, W/m ^{0}k

$\left[\frac{\Delta\theta}{\Delta x}\right]_1$ = temperature gradient across the seale ^{0}k/m

$\left[\frac{\Delta\theta}{\Delta x}\right]_2$ = temperature gradient across the tube wall ^{0}k/m

$\left[\frac{\Delta\theta}{\Delta x}\right]_1$ = temperature gradient across the slag ^{0}k/m

If all the three processes of heat transfer take place simultaneously then overall resistance to the heat flow.

$$R = R_{rad} + R_{conv} + R_{cons}$$
(9)

Therefore, heat flow rate,

$$q = \frac{\Delta\theta}{\sum R} = UA\Delta\theta$$
(10)

Where, U is called overall heat transfer coefficient

As a boiler designer, I must consider the internal and external fouling factors while designing the super heaters, water walls, economizers etc.
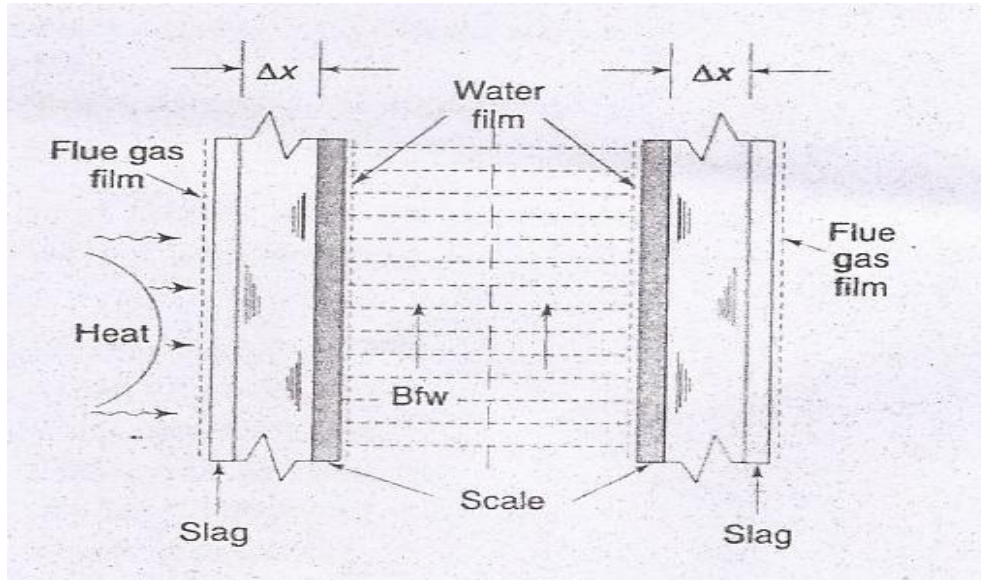
Figure 1: Internal and external fouling factors of a boiler [6]

The overall heat transfer coefficient is given by;

$$U = \frac{1}{\frac{1}{h_0} + \left[\frac{\Delta\theta}{K} + \frac{1}{h_{si}} + \frac{1}{h_{so}}\right] + \frac{1}{h_i}} \quad W/m^{2 0}k$$

(11)

Where;

h_o = coefficient of heat transfers across the flue gas film on the tube surface, $W/m^{2 0}k$

h_i = coefficient of heat transfers across the water or air or steam film on the inner surface of the tubes, $W/m^2\ k$

K = thermal conductivity of the tube wall, $W/m\ ^0k$

Δx = thickness of tube wall, m

h_{is} = coefficient of heat transfers of the scale inside the tube $W/m^{2\ 0}k$

h_{so} = coefficient of heat transfers of the slay outside the tube $W/m^{2\ 0}k$

Now if the hot modes of heat transfer across the hot flue gas film on the outer surface of the tube is both radiative and convective type than the rate of heat transfer.

$$Q = q + q = (K_r + K_{cv})\Delta\theta$$

(12)

Where, K_r = radioactive conductance of hot gases

$$= \sigma AE\left(T_1{}^4 - T_2{}^4\right)\Big/\Delta\theta$$

(13)

Also, K_{cr} = convective conductance of hot gas film

$$= (h_{cv})_o{}^A$$

(14)

$$h_O = \sigma EF\left[\left(T_1{}^4 - T_2{}^4\right)\Big/\Delta\theta\right] + (h_{cv})_0$$

(15)

2.3 Design of Steam Generating Equipment

1) Air Preheater Design

• **Heat load**

The amount of heat transferred from the hot exhaust gas to the air stream is given by

$$Q = \dot{M}aCp\ (\Delta\theta)a \quad (Kw)$$

Where;

$\dot{M}a$ = mass flow rate of air

$(Cp)a$ = specific heat of air at mass temperature

$(\Delta\theta)a$ = temperature differences between the air at inlet and outlet

• **Heating surface of the air heater.**

This is given by;

$$A = \frac{Q}{U\Delta\theta} \ (m^2)$$

(16)

Where;

Q = amount of heat

U = overall heat transfer coefficient

$\Delta\theta$ = air temperature difference at inlet and outlet

- **Total number of tubes**
This is denoted by

$$n = \frac{\dot{M}fg}{\frac{\pi}{4}d_i{}^2 \ Vfg \ \rho \ fg}$$

(17)

Where;

f_g, v_{fg}, P_{fg} are constant values gotten from Reynolds number for flue gas at mean temperature

- **Heights of tubes in one pass**

$$Li = \frac{A}{2\pi din}$$ (18)

- **Number of tubes arranged across the flow**

$$Ni = \frac{[A]deer}{Li \ (p_i - d_o)}$$

(19)

2) Economizer Design
The following data must be duly considered

- **Steel tube specification**
ID = Intent diameter of tube

OD = Outer diameter of tube

- **Tube layout**
Transverse pitch: Longitudinal pitch = 1.05

Longitudinal pitch = twice the diameter of the tubes.

- **Boiler feed water**
Flow rate = 200 t/h

Mean velocity of flow = 0.75m/s

Mode of flow: vertical, from bottom to top

- **Hot exhaust gas**
Composition; $CO_2 - 13\%$; $H_2O - 11\%$

Mean velocity of flow in the narrowest cross section of tube-banks = 15m/s

Flow rate = 480 t/h

Temperature inlet to economizer = 860°C

3) Evaporator Design
- **Calculation of log Mean Temperature Difference (LMTD)**

$$LMTD = \frac{\Delta\theta_1 - \Delta\theta_2}{In\left(\Delta\theta_1/\Delta\theta_2\right)}$$

(20)

Where;

$\Delta\theta_1$ = temperature difference of the evaporator inlet

$\Delta\theta_2$ = temperature difference at the evaporator outlet

- **Heat load of evaporator**

$$q = \dot{M}. \ Cp. \ \Delta\theta \ Kw$$

(21)

- **Heat transfer surface of the evaporator**

$$A = \frac{q}{U.LMTD} \ m^2$$

(22)

- **Number of loops**
If "n" is the number of loops connected in parallel

$$\pi/4. \ d_i{}^2. v. p. n = \frac{230 \times 10^3}{3600}$$

(23)

- **Determination the length of each loop**
If "l" is the length of an individual section (loop)

$$\pi. \ d.n. \ l = A$$

4) Combustor Design
A heat balance over the combustor is given by the following equation

$$[1.1 \ m_1 \ T_2 - m_1 \ T_1]C_p + \ m_2 \ [T_2 - T_1]C_p = 3100 \frac{m_1}{\Omega}$$

(24)

Where:

O.1m_1 = assumed exhaust gas mass flow rate

M_1 = mass flow rate of fluidizing air, kg/s

M_2 = mass flow rate of circulating air, kg/s

Ω = excess air factor

C_p = specific heat of gas or air KJ/ (kg. ⁰k)

5) Super Heater Design
The following parameters must be duly considered in the designing of a super heater;

- **Heating surface of super heater**

Recall that;

$$Q = UA\Delta\theta$$
(25)

$$A = \frac{Q}{U\Delta\theta 1_m}$$

Where;

A = surface area of the super heater

Q = quantity of steam generated

U = overall heat transfer coefficient

Olm = temperature difference of the hot exhaust gases

- **Number of loops**

To determine the number of loops (N) the following equation are considered

$$G_s = \left(\frac{\pi}{4} d_1{}^2 N\right)(V_s)(\ell_s)$$
(26)

Where:

G_s = steam flow rate

V_s = average velocity of steam

ℓ_s = density of steam

d_i = internal diameter of the tubes

N = number of loops

- **Length of each loop**

If L be the length of each loop, then;

$$A = (\pi d_0 L)N$$
(27)

$$L = \frac{A}{(\pi d_0)N}$$
(28)

Where;

H = length of each loop

A = surface area of each loop

d_0 = outer diameter of the tube

N = total number of loops *(Chattopadhyay, 2000)*

6) **Steam Heating Loads**

- **Fuel heating:** The quantity of steam used for heating of fuel oil is given by;

$$Q_F \frac{R_t \times C \times \Delta T}{h_s - h_d} \text{ (kg/h)}$$
(29)

Where;

Q_F - Steam consumed for heating of fuel, kg/h

C - Specific heat of fuel KJ/kg ^{0}c

ΔT - Temperature rise of fuel ^{0}c

h_s - Specific enthalpy of heating steam KJ/kg

h_d - Specific enthalpy of condensate KJ/kg

- **Steam consumption for hotel services:**

When hotel services are supplied by steam, the following allowance should be made:

For heating of domestic water Q = 0.4 X N

For galley   Q = 0.25 X N

For laundry   Q = 0.05 X N

Total hotel services Q_{TH} = 0.70 X N
(30)

Where;

Q_{TH} - Steam consumption for hotel services

N - Total complement

- **Heating steam for air conditioning system:**

The quantity of steam consumed by the air conditioning system may be assessed from the formula

$$QA = 0.01 \times N \times C \text{ (kg/h)}$$
(31)

Where;

Q - Steam consumption

N - Total complement of the heat

C - 100 for houseboat

- **Steam for houseboat heating in cold weather:** The quantity of steam consumed for houseboat heating in cold weather may be assessed from the formula below

$$Q_{EW} = A (T_2 - T_1) \times H \times 0.00081 \text{ (kg/h)}$$
(32)

Where;

Q_{EW} - Steam consumption for houseboat heating

A - 100 for houseboats

T_1 - outside temperature

T_2 - inside temperature

N - total complement of the boat.

- **Steam consumption for distilling plant:** At a service load of distilling plant when using steam from the heat recovery boiler, the quantity of steam may be calculated from the equation.

$$Q_{DP} = \frac{LPD \times f_D}{24(h_1 - h_d)} \quad (kg/h)$$
(33)

Where;

Q = mass of distiller heating steam, (kg/h)

LPD = distiller output (kg/24h)

f_D = heat consumption of distillate KJ/kg

h_1 = specific enthalpy of heating steam

h_d = specific enthalpy of condensate KJ/kg

- **Total steam consumption within the houseboat**

The total steam consumption of the heat recovery boiler at a given exhaust gas and steam condition may be calculated from

$$Q_H = Q_F + Q_{TH} + Q_A + Q_{CW} + Q_{DP}$$
(34)

Where;

Q_H = quantity of steam generated

Q_F = steam consumption for fuel heating

Q_{TH} = steam consumption for hotel services

Q_A = steam consumption for air conditioning

Q_{CW} = steam consumption during cold weather

Q_{DP} = steam consumption for distilling plant

TABLE 1: PARAMETERS CONSIDERED FOR PROPER ANALYSIS

For Air Preheater	
Inside/outer diameter of the tubes	50/55mm
Air Temperature difference	30°c to 274°c
Thermal conductivity of the tube material	$47W/m^{2o}$c
The tubes are in staggered arrangement	$p_1 = p_2 = 2.5d$
Mean velocity of air flow	10m/s
Flue gas composition	CO_2- 13%, H_2O- 11%
For the Economizer	
Inside/outer diameter of the tubes	45/50mm
Thermal conductivity	$26W/m^{2O}$c
Longitudinal pitch	twice the mean diameter of tubes
Boiler feed water	
Inlet/outlet temperature	140°c to 300°c
Flow Rate	200t/h
Mean Velocity of flow	0.75m/s
Flue Gas; Composition	$CO_2 – 13\%\ H_2O – 11\%$
Mean velocity of flow in the narrowed cross section of tube	15 m/s
Flow Rate	480 t/h
The gas temp inlet to economizer	800°c
At The Super heater	
Inside/outer diameter of tubes	35/30mm
Thermal conductivity of the material of the tubes	$25W/m^{2O}$c
The gas temperature at entry to super heater	1200°c
Mass Flow Rate of the gas	72 t/h
Steam to be super-heated	250 t/h
Superheated steam temp	525°c
Superheated steam pressure	$10\ MN/m^2$

III. RESULTS AND DISCUSSION

3.1. General Analysis on the Major Boiler Heat Exchanger Components

The Following Parameters Will Be Determined.

- **Average temperature of air preheater**

The average temperature of air in the air preheater is given by;

$\Theta_a = ½ (30 + 274) = 152^oc$

At the temperature (152^oc), air has the following physiological properties.

K = 3.58 x 10^{-2}W/m^{2O}c

Y = 29.174 x 10^{-5} m^2/s

Pr = 0.6792

μ = 24.18 x 10^{-6}Prs

C_p = 1.0154 KJ/Kg ^{o}c

P = 0.830 Kg/m^3

- **Heat load at the air preheater.**

The amount of heat transferred from the hot flue gas to the air stream.

From equation (21) the heat load Q is given as;

$$Q = M_a C_p (\Delta\Theta)$$
$$= \left(\frac{90 \times 100}{3600}\right)(1.0154)(274 - 39)$$
$$= 6193$$

- **Heat transfer coefficient from tube wall to the air.**

$$h_{fg} = \frac{N_{ufg} \times K_{fg}}{d_1}$$
$$= \frac{41.122(4.40 \times 10^{-2})}{50 \times 10^{-3}}$$
$$= 36.187 \ W/m^2{}^{\circ}C$$

- **Overall Heat Transfer Coefficient**

The overall heat transfer coefficient of the air preheater as stated in equation (11), is given by;

$$U = \frac{1}{\frac{1}{h_{fg}} + \frac{\Delta x}{K_W} + \frac{1}{h_a}}$$
$$= \frac{1}{\frac{1}{36.187} + \frac{2.5 \times 10^{-3}}{47} + \frac{1}{89.575}}$$

$= 25.739 W/m^2{}^{\circ}C$

- **Heating surface of Air Heater**

From equation (3.16), we can deduce that;

$$A = \frac{Q}{\mu\Delta\theta} = \frac{6193.94 \times 10^3}{(25.737)(80.257)}$$
$$= 2.998 \ m^2$$

- **Total Number of Tubes**

From equation (17), n is denoted as;

$$n = \frac{\dot{M}fg}{\frac{\pi}{4}d_i{}^2 \ Vfg \ \rho \ fg}$$
$$= \frac{72 \times 1000 \ /3600}{\pi/4 \ (50 \times 10^{-3})^2 (17)(0.6864)} = 872$$

- **Height of tubes in one pass**

$$L_i = \frac{A}{2\pi din}$$
$$= \frac{2998}{2(\pi)(50/103)(872)} = 10.94m$$

Since the tube height becomes abnormally large, I decided to halve the tube – height and increase the number of tubs two-fold, therefore the total number of tubes = 2 x 872 = 1744

Tube – height in one pass = ½ (10.94) = 5.47m

- **Mean Temperature of Boiler Feed water**

$$\Theta w = \Theta w_1 + \frac{\Theta w_2}{2} = 140 + \frac{300}{2}$$
$$= 220 \ ^{\circ}C$$

- **Heat Transfer Rate of Boiler Feed water @ 220°c**

P = 840 kg/m^3

C_p = 4.615 kJ/kg ^{o}c

K = 64.5 x 10^{-2} W/m^{o}c

V = 0.148 x 10^{-6} m^2/s

Pr = 0.89

The rate of heat transferred to the boiler feed water (BFW) in the economizer = Mass flow rate x Sp x heat x temp rise

$$= \frac{200 \times 103}{3600}(4.615)(300 - 140)$$
$$= 41022KW$$

- **Local Heat Transfer Coefficient for boiler feed water (BFW)**

$$h_1 = \frac{N\mu_1 K_1}{d_1} = \frac{386(64.5 \times 10^{-2})}{50 \times 10^{-3}}$$

$$= 5532.67 \; W/m^2{}^\circ C$$

- **Log Mean – Temperature – Difference**

Assuming the flue gas was in counter flow with water.

$$\frac{\Delta\Theta}{m} = \frac{500 - 411}{\ln(500/411)} = 454^\circ C$$

- **Heating surface of Economizer**

From equation (3.22), heating surface area is given as;

$$A = \frac{Q}{UA\Theta lm} = \frac{41022 \times 10^3}{(120)(454)} = 753 m^2$$

- **Number of loops in parallel**

Let the number of loops be n

$$\left(\frac{\pi}{4} d_1^2 V_{water}\right) \rho H_2 O(n) = \frac{200 \times 10^3}{3600}$$

$$= \left(\frac{\pi}{4}\right)(50 \times 10^{-3})^2 (0.75)(840)(n)$$

$$= \frac{1}{18} \times 10^3$$

$$n = 55.44 \; i.e \; 55 \; number \; of \; loops$$

- **Length of Individual Loop**

$$\pi d_2. n. L = A$$

$$\pi(55 \times 10^{-3})(55)L = A$$

$$L = 90.60 \; m$$

- **Heat Load at the Super heater**

The rate of superheated steam generation = 250 t/h

Heat load of the super heater

$$= \frac{250 \times 10^{-3}(3437 - 2725) \; kW}{300} = 49444.44 kW$$

- **Steam Parameters**

Average temperature of superheated steam

$$= \tfrac{1}{2}(311 + 525) = 418^\circ C$$

Superheated steam values, $418^\circ C$. 10 MN/m^2

$\rho = 0.259$ kg/m^3, v = 1 94 x 10^{-6} m^2/s

k = 11.64 x 10^{-2} W/m$^\circ$C

$P_r = 1.1$

- **Coefficient of Heat Transfer from Wall to Steam**

Reynolds Number for Steam

$$Re = \frac{dl \, v}{V} = \frac{(30x \, 10^{-3})(18)}{0.703x \, 10^{-6}} = 768136$$

- **Nusselt Number for Steam**

Nu = 0.021 $(768136)^{0.8}$ $(1.1)^{0.43}$ = 1117.81

Coefficient of Heat Transfer from tube wall to Steam

$$h_i = Nu \frac{k}{d_i} = \frac{(1117.81)(7.1 \, x \, 10^{-2})}{30x \, 10^{-3}}$$

$$= 2645 \; W/m^2{}^\circ C$$

- **Flue Gas Temperature at Outlet**

In order to determine the flue gas exit temperature, we must go through successive approximations.

Approximation: let $C_{pg} = 1.31$ kJ/kg$^\circ$C for flue gas.

Exit temperature of flue, $[\Theta_2]_g$

$$[\Theta_2]_g = [\Theta_1]_g - \frac{Q}{M_g \times C_{px}}$$

$$= 1200 - \frac{49444.44}{\left(600 \times \frac{103}{3600}\right) \times 1.31} = 974^\circ C$$

Therefore, average temperature of flue gas = (1/2) (1200 + 974) = 1087°C

- **The Coefficient of Heat Transfer by Radiation**

$$h_{rad} = \frac{q_{rad}}{\Theta_g - \Theta_w} = \frac{15848}{1087 - 460}$$

$$= 25.27 \; W/m^2{}^\circ C$$

The Coefficient of Heat Transfer from Flue Gas to tube walls

$h_o = h_g + h_{rad} = 83.2 + 25.27 = 108.47$ W/m$^{2\circ}$C

- **Heating Surface of Super heater**

From equation (28), heating surface area is given as;

$$Q = UA \, \Delta\Theta_{1m}$$

Therefore,

$$A = 49444.44 \times 10^3 = 717.55$$

$$\approx 718 \; m^2$$

- **Number of Loops**

Let N be the number of loops

$G_g = (\pi/4 \; d_i^2 \; N) \; (V_s) \; (\rho_s) = (\pi/4) \; (30 \times 10^{-3})^2 (N) \; (18) \; (37)$

Therefore, $N \approx 148$

- **Length of Each loop**

If L be the length of each loop, from equation (30), we obtain;

$A = (\pi d_o L) \; N$ or $718 = \pi \; (35 \times 10^{-3}) \; L \; (148)$
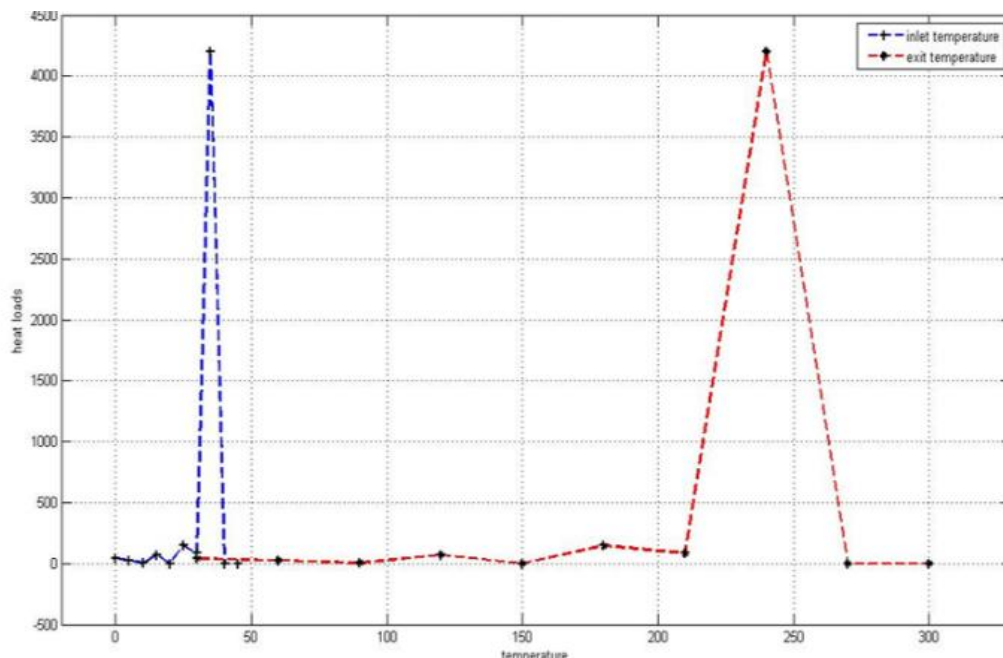
Therefore, L = 44.12 m

Figure 2: Graph of feed water inlet and exit temperatures against engines heat load, generated from MATLAB to ascertain the performance of the heat recovery boiler, assuming that the exhaust gas pressure is kept constant

IV. CONCLUSION

The mathematical model is used to determine the performance of the heat recovery boiler, to produce just saturated steam needed on board a 100 -Man houseboat at the various feed water inlet and exit temperatures with respect to the engines heat loads. The results obtain indicates the following deduction; that the higher the exhaust gas heat load, the higher the steam temperature and at constant mass flow rate of the exhaust gas heating. The feed water outlet temperature may vary due to total head losses in the system. The higher the heat transfer coefficient, the higher the fractional flow of the feed water, the coefficient of heat transfer by thermal radiation helps to determine the following parameters during boiler design stage: mean beam length, average emissivity of flue gas, average absorptivity of flue gas, effective emissivity of the boiler enclosure and the superheated steam yield is very low due to the relative low temperature and mass flow rate when the main engine operates under 50% SMCR. With main engine loads going up, the yield increases sharply after opening the exhaust gas bypass valve. The variation of ratios of superheated steam yield at other heat recovery boiler working pressures to the specific pressure of 0.7MPa with main engine loads. It is clear that there is not a positive correlation between the superheated steam yield and the exhaust gas boiler pressure, and the maximum of superheated steam yield occurs at the heat recovery boiler pressure of 0.6MPa.

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