T.R.
ONDOKUZ MAYIS UNIVERSITY
INSTITUTE OF GRADUATE STUDIES
DEPARTMENT OF ENVIRONMENTAL ENGINEERING

ECONOMIC AND ENVIRONMENTAL EVALUATION OF HEAT RE-USE IN POWER PLANTS

Master Thesis

Wakjira TESFAYE

Supervisor
Prof. Dr. Bahtiyar ÖZTÜRK

SAMSUN

2020

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THESIS APPROVAL

The thesis prepared by Wakjira **Tesfaye MULETA** under the supervision of **Prof. Dr. Bahtiyar ÖZTÜRK** titled *Economic and Environmental Evaluation of Heat Re-Use in Power Plants* has been accepted as **Master's Thesis** by the jury members on the day of 18/11/2020.

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Chair **(Thesis Supervisor)**	Prof. Dr. Bahtiyar ÖZTÜRK Ondokuz Mayıs University Department of Environmental Engineering		☒ Accept ☐ Reject
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This thesis was approved by the jury members whose names are written above and determined by the Institute Administrative Board.

APPROVAL

__/__/ 2020

Prof. Dr. Ali BOLAT

Director of Institute

ETHIC DECLARATION

I declare that I have prepared all the information in this thesis according to Ondokuz Mayıs University Institute of Graduate studies thesis writing rules. The work provided in this thesis is correct, complete and unless otherwise referenced, is the researcher's own work, and has not been submitted elsewhere for any other degree or qualification.

21/12/2020
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STATEMENT OF THE THESIS STUDY ORIGINAL REPORT

Thesis Title: Economic and environmental evaluation of heat re-use in power plants

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ÖZET

ECONOMIC AND ENVIRONMENTAL EVALUATION OF HEAT RE-USE IN POWER PLANTS

Wakjira TESFAYE
Ondokuz Mayıs Üniversitesi
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Çevre Mühendisliği Ana Bilim Dalı
Yüksek Lisans, Kasım/2020
Danışman: Prof. Dr. Bahtiyar ÖZTÜRK

Termik santraller ve ağır sanayinin farklı kısımlarında üretilen atık gaz, buhar, yağ ve su gibi akışkanlarca yüksek sıcaklığa sahip büyük miktarda ısı taşınır ve bu ısının büyük çoğunluğu atılır. Mevcut durumda, bu ısının yarısı herhangi bir pratik kullanım olmaksızın çevreye salıverilir ve kaybolur. Kaybolan ısının buhar üretme, ön ısıtmada ve başka şekillerde kullanılmasıyla yakıt tasarrufu ve enerji verimliliği sağlamak mümkündür.

Farklı durumlarda çeşitli sistemlerden atılan ısının geri kazanılması için ısı değiştirici, atık ısı kazanı ve atık ısı geri kazanımı gibi çeşitli teknikler vardır. Termik santraller ve ağır sanayilerde atık ısı geri kazanılması daha az yakıt kullanımı, temiz enerji üretimi, daha az sera gazı üretimi (CO_2), hava kirletici gazların (NOx, SOx,..) emisyonunu azaltma ve küresel ısınma etkisini azaltma gibi avantajlar sunar.

Bu yüksek lisans tezi ağır sanayilerde atık ısı geri kazanımının ekonomik ve çevresel etkilerini değerlendirmeyi amaçlamaktadır. Bu değerlendirmeyi yaygın olarak kullanılan borulu ısı değiştiriciyi kullanarak yaptık. Bu değerlendirme yakma odaları, endüstriyel fırınlar, termik santraller ve ağır endüstrilerden atık gaz, buhar, yağ ve su gibi akışkanlarla atılan ısı problemini konu almaktadır. Çalışmanın ana hedefleri bu sistemlerden atılan ısıyı geri kazanarak sistemin kendi içinde veya komşu sistemlerde kullanışlı hale dönüştürmek ve enerji dönüşüm sistemini mümkün olduğunca verimli hale getirmektir. Atık ısının geri kazanılabilirliği, kaynaklarının ve transfer edileceği akımların karakterizasyonu da değerlendirilmiştir.

Yapılan çalışmada kömür kullanan bir termik santralde yakma havası borulu ısı değiştiricilerle ısıtılarak %1.87'ye kadar kömür tasarrufu sağlanabileceği ve buna bağlı olarak sera gazı (CO_2) emisyonunun %13.47 azaltılabileceği ve diğer hava kirletici gaz ve tozların emisyonunun da azaltılabileceği görülmüştür.

Anahtar Sözcükler: Termik santral, atık ısı kazanımı, borulu ısı değiştirici, yakıt tasarrufu, emisyon azaltma.

ABSTRACT

ECONOMIC AND ENVIRONMENTAL EVALUATION OF HEAT RE-USE IN POWER PLANTS

Wakjira TESFAYE
Ondokuz Mayıs University
Institute of Graduate studies
Department of Environmental Engineering
Master's Thesis, November/2020
Supervisor: Prof. Dr. Bahtiyar ÖZTÜRK

A large quantity of heat, which is carried by flue gases, steams, oils and water…etc., at high temperatures, is produced by power plants and different parts of the heavy industries, and much of this heat will be wasted. In the present condition, half of the produced heat is dumped into the environment without any practical use and is lost. It is possible to get efficient energy and fuel-saving by trapping the heat lost and thereby using it to produce steam, preheat air, or other.

There are various techniques such as economizer, waste heat boiler, and waste heat recovery generator in use for the recovery of waste heat from various systems under different situations. The main advantages offered by the waste heat recovery in power plants and other heavy industries can be less fuel consumption, clean power generation, minimum production of greenhouse gases (CO_2), minimizing of air polluting gases (NO_x, SO_x..), and reduction of a global warming effect.

This master thesis attempts to evaluate the recovery of waste heat in heavy industries on their economic and environmental impacts. We use the most common and widely used heat recovery of shell and tube heat exchanger to evaluate it. This evaluation addresses the problem of waste energy which is wasted out from combustion chambers, industrial furnaces, and other waste heat sources of power plants and heavy industries in the form of flue gases, steams, oils and water. The main objectives are to recover the wasted heat from those systems, transfer it to the source back into the useful work within its own or neighboring systems, and also to make the energy conversion process as efficient as possible. The achievability of waste heat recovery and characterizing the waste heat sources and the streams to which the heat will be transferred are also evaluated.

The results of the study showed that, depending on the combustion air temperature, it is possible to reduce greenhouse gas (CO_2) emission up to 13.47 % by saving coal consumption of 1.87 % and it is also possible to reduce emissions of pollutant gases and particulates by applying a shell and tube heat exchanger to heat up combustion air in a power plant which use coal.

Keywords: Power plant, waste heat recovery, shell and tube heat exchanger, fuel saving, emission reduction.

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Finally, my thanks go to all the people who have supported me to complete the research work directly or indirectly.

November 18, 2020, Samsun

Wakjira TESFAYE

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LIST OF ABBREVIATIONS

- A Heating area, m^2
- A_1 Area of the tube layout refers to a single tube, m^2
- A_c Heat transfer area without fouling, m^2
- A_f Heat transfer area with fouling, m^2
- A_o Heat transfer surface area (based on the outside diameter of the tube, m^2
- A_{of} Heat transfer area based on the outside with fouling, m^2
- A_{tp} Tube side heat transfer, m^2
- B Baffle spacing, m
- BWG Birmingham Wire Gage
- C Clearance between adjacent tubes, m
- CL Layout of the tubes constant
- C_p Specific heat at constant pressure, $J/kg \cdot K$
- CTP Tube count calculation constant
- C_v Specific heat at constant volume
- D_e Equivalent diameter on the shell side, m
- D_s Shell diameter, m
- d_o Tube outside diameter, m
- e Wall thickness in, m
- F Correction factor
- f friction factor
- GCV Gross calorific value
- GHG Greenhouse gas
- G_s Shell-side mass velocity, $Kg/m^2 \cdot s$
- H Hydrogen
- HRSG Heat recovery steam generator
- h_{fg} Latent heat of the phase change, J/kg
- h_i Shell-side heat transfer coefficient, $W/m^2 \cdot K$
- h_o Tube-side heat transfer coefficient, $W/m^2 \cdot K$
- ID Internal diameter of tubes, m
- k Thermal conductivity of the wall, $W/m \cdot K$
- k_f Thermal conductivity of flue gases at a specific temperature, $W/m \cdot K$
- kW Kilowatt
- L Tube length of the heat exchanger, m
- LMTD Logarithmic mean temperature difference
- MW Megawatt
- $\dot{m}$ Mass of the stream changing phase per unit time, Kg/s
- $\dot{m}_{act}$ Actual saved coal, Kg/s
- $\dot{m}_{coal}$ Saved coal, Kg/s
- N Nitrogen
- N_b Baffle number
- N_t Number of tubes
- N_p Number of tube passes
- Nu Nusselt number

- Nu_b Correlation between Gnielinski's
- O Oxygen
- OD Outside diameter of the tubes, m
- P Power (kW or kcal/h)
- PR Tube pitch ratio
- Pr Prandt number
- P_T Tube pitch, m
- Q Heat duty of the exchanger, kw
- Q_{act} Actual air feed amount, m^3/s
- R Resistance due to the wall fouling, $m^2 or\ k/w$
- Re Reynolds number
- Re_s Shell side Reynolds number
- R_{fi} Fouling resistance, $m^2 or\ k/w$
- S Sulphur
- STHE Shell and tube heat exchanger
- T Temperature, °C
- TEMA Tubular exchanger manufacturers association.
- U Heat transfer coefficient, $W/m^2 \cdot K$
- U_o Overall heat transfer coefficient, $W/m^2 \cdot K$
- U_c Overall heat transfer for a clean surface, $W/m^2 \cdot K$
- U_f Overall heat transfer coefficient, $W/m^2 \cdot K$
- u_m Average velocity inside tubes, m/s
- WHB Waste heat boiler
- W_i Mass of elements found in a coal W_i (i: 1-8)
- $\triangle P$ Pressure drop, Pa
- ΔP_s Shell side pressure drop, Pa
- ΔT Change in temperatures in, °C
- ϕ_s Viscosity correction factor for shell side
- μ_w Wall temperature viscosity at a specific temperature
- μ Dynamic viscosity
- ρ Density, kg/m^3
- η Overall surface efficiency

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1. INTRODUCTION

Wasted heat for recovery starts by evaluating the benefits regarding the environment and economy. The generation of waste heat takes place by the burning of fuels or chemical reactions in different power plants and industries and then discarded (leaked) to the atmosphere (ambient) without any task despite it may still be used in various helpful and for economical tasks (Jouhara, et al., 2018).

The energy of waste heat is linked with the waste streams of air, gases, and liquids. In waste heat, the ‘value’ of the heat quality is more essential than the ‘amount’ of heat. In the process of installation, the temperature of the waste heat gases and the economics are basics and the recovery of waste heat depends on it (Kuppan, 2013). For energy consumption, a large quantity of hot flue gases is produced in boilers, furnaces, kilns, and ovens that some amount of heat energy will be lost. If we recovered some of the wasted heat, a significant amount of fuels (energy) will be saved and we can use it for the process itself or exclusive uses (Jouhara, et al., 2018). However, we cannot recover fully and much of the heat will be recovered and reduce the loss in waste heat (Jelena, et al., 2018). We cannot discard recklessly this valuable product (heat energy) valued with the high cost and environmental impact of fossil fuels. For sizeable heat recovery of waste, any exhaust stream with a temperature above 250°F has the potential for some useful purposes in an economic routine (Suryawanshi and Pitale, 2017).

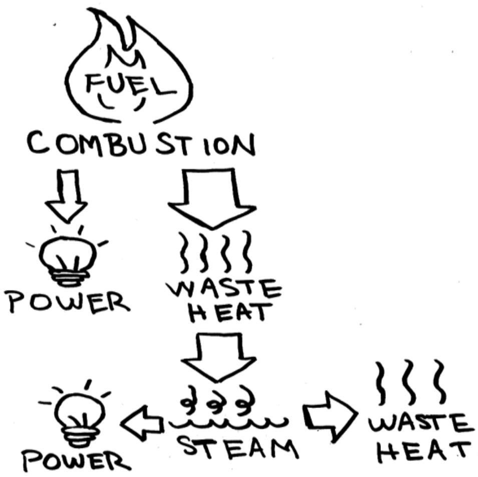

Figure 1.1. Schematic drawing of the system from combustion to waste heat (Anonymous.1, 2020).

2. RELATED LITERATURE REVIEW

2.1. Heat exchangers

A heat exchanger is a device that offers heat from a fluid ((gas) air) to transfer to a second fluid (another gas source) whereas the two fluids not having to mix along or obtain direct contact or a device that transfers heat between different two process fluids (air) without this two fluids mixing across the process (Çengel and Ghajar, 2015). The heat exchanger has a large benefit in power plants and some other big industries and additionally for domestic uses. Some of the types developed for the use in coal power plant, chemical processing plant, transportations, air conditioning building heat, refrigeration's and so on (Ramesh and Dušan, 2003).

Designing the actual heat exchanger will be complex and have some obstacles. Evaluation of heat transfer alone cannot make it happen rather we have to consider the cost of fabrication and installation, size, and weight. All this consideration has an advantage and plays an important role within the choice and ultimate design from a complete value of possession purpose (Khayal, 2018).

Usually convection in each fluid (air) and conduction through the wall separating the two fluids (air) involved in heat exchangers (Khayal, 2018).

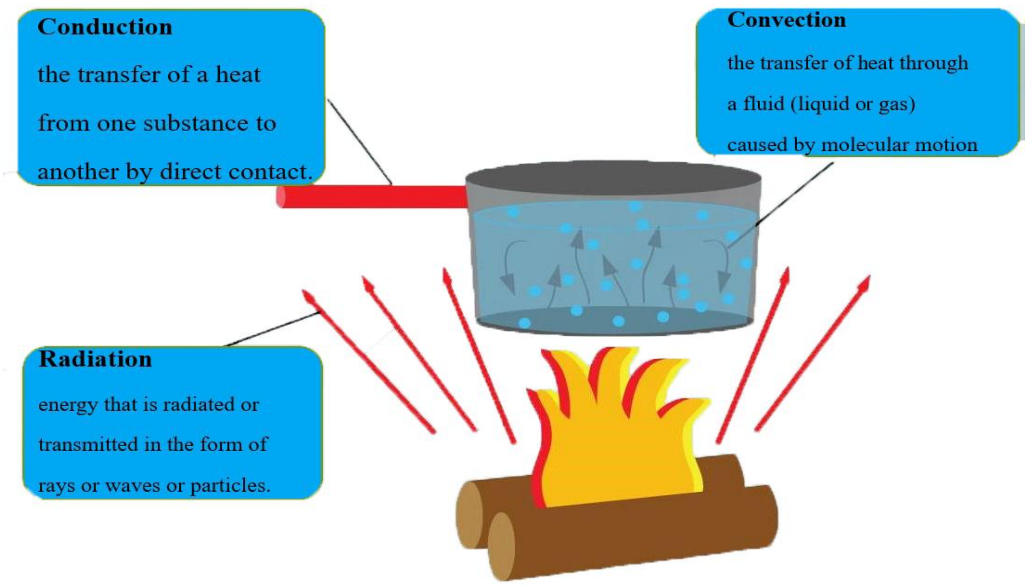

Figure 2.1. Heat transfer mechanisms (Anonymous.6, 2019)

2.1.1. A basic design method of heat exchangers

The design (layouts) of a heat exchanger addresses this essentially three phases:

A. The selection of technology
B. Thermal design
C. Heat exchanger pressure drop calculation.

2.1.1.1. The selection of technology

Finest technology selection is linked with the listed elements:

- The thermal program (Required temperatures, efficiency…)
- The fluids nature
- The application (where to use and others)
- The limitations of installations and maintenances.

Taking all these elements permits us to define which type of materials are used for heat exchanger and allows the type of heat exchanger we use. Sometimes using specific material is required to satisfy the choice and selection of the exchanger. (E.g. liquids like sea water cannot be used on titanium application jointly on all kinds of heat exchanger but we can use on some other exceptional exchangers) (Anonymous.1, 2020).

2.1.1.2. Thermal design

Validation of the thermal program

Before heading to the design of a heat exchanger, first we have to decide the choice of technology we use. The design of heat exchanger can determine its power, size, and geometry. The data validation of thermal program is vital and seen under this 3 formulas:

$$P_{hot} = P_{cold}$$
$$P_{hot} = Q \times C_p \times \Delta T_{hot}$$
$$P_{cold} = Q \times C_p \times \Delta T_{cold}$$

C_p: specific heat, $J/kg \cdot K$

P: power, kW or $kcal/h$

Q: Flow rate of heat, kg/h

ΔT*: Tin-Tout,* ℃ or K

Heating surface calculation

Logarithmic mean temperature difference (ΔT_{lm}), logarithmic average of the temperature variations at any end of the heat exchangers (Kuppan, 2013). First, ΔT_{lm} will be calculated using the formula given below.

$$\Delta T_{lm} = \frac{\left(T_{h_1} - T_{c_2}\right) - \left(T_{h_2} - T_{c_1}\right)}{\ln \frac{T_{h_1} - T_{c_2}}{T_{h_2} - T_{c_1}}}$$

- $T_{h_1} - hot\ in\ temp$
- $T_{h_2} - hot\ out\ temp$
- $T_{C_1} - cold\ in\ temp$
- $T_{C_2} - cold\ out\ temp$

Previously, log mean temperature difference (ΔT_{lm}) and Power has been calculated, and following this the heat exchanger calculation will be done as per the following formula;

$$\boldsymbol{Q = U \times A \times \Delta T_{lm}}$$

U: Heat Transfer Coefficient, $\boldsymbol{kW}/°\mathrm{C}/\boldsymbol{m^2}$. Specific for the materials and manufacturers give it.

A: Heating area

Calculation of heat transfer coefficient

The heat transfer coefficient (U) is a heat transferred per unit area per kelvin and the calculation for a heat exchanger will be as the following formulas given:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2} + \frac{e}{k} + R$$

e: thickness of the wall, $\boldsymbol{m}$

k: thermal conductivity of the wall, $\boldsymbol{W}/°\mathrm{C}/\boldsymbol{m}$

R:** resistance occurred due to the **wall fouling

And h_1 and h_2: Local heat exchange coefficients can calculated as per the local geometry as well as dimensionless numbers such as Reynolds (Re), Prandt (Pr), and Nusselt (Nu)

Getting the *Heat Transfer Coefficient* (U) enables us to find the heating area and hence sizing the heat exchanger (Çengel and Ghajar, 2015).

2.1.1.3. Calculation of the heat exchanger pressure drop

Because of friction at the walls (ordinary head loss) or accidents (singular head losses) a moving substance undergoes several power losses. This loss of energy is expressed in pressure drop (△P) and it needs to be compensated to allow the moving substance (air, fluid…) (Çengel and Ghajar, 2015).

2.1.2. Heat exchanger design characteristics

Entire heat exchangers are characterized under the same essential concepts and it can be labeled and classified in numerous exclusive methods primarily depending on their design characteristics (Kuppan, 2013). Heat exchangers can be categorized by using means of these main (fundamental) characteristics. It includes;

- Based on flow configuration
- Based on the construction method
- Based on the heat transfer mechanism

2.1.2.1. Based on flow configuration

This flow configuration is the directional motion of the fluids in the heat exchanger with each other. This flow configuration has 4 principal;

- Parallel flow principals
- Crossflow principles
- Hybrid flow principles
- Counterflow principles

Parallel flow principles

Parallel flow heat exchangers, in this type of heat exchanger both the fluids flow parallel throughout the process, and in the same direction to each other. This configuration will have lower efficiency than the counter-flow type of arrangement. But it has a high thermal uniformity throughout (across) the wall of the exchanger (Çangarli, 2008).

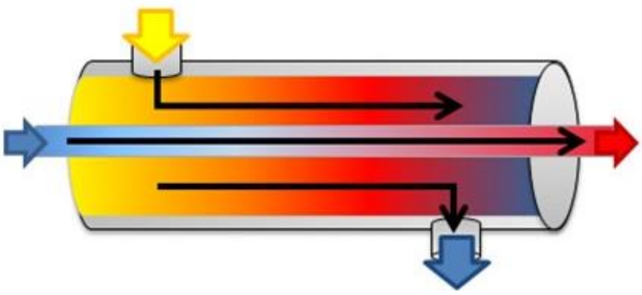

Figure 2.2. Parallel flow diagram (Çengel and Ghajar, 2015)

Counter flow principals

Counterflow heat exchangers are designed for the fluids to move reverse to each other within the heat exchanger. It is the most common and widely used of the flow configurations and this arrangement typically gives the highest efficiencies as it permits for the greatest amount of heat transference between fluids. Due to this, there will be the highest change in temperature (Çangarli, 2008).

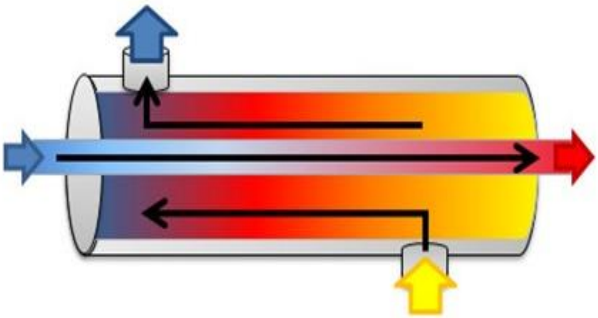

Figure 2.3. Counterflow diagram (Çengel and Ghajar, 2015)

Cross-flow principles

In *crossflow principles*, the flow of fluids is perpendicular to one another inside the heat exchanger. The efficiencies of this principle come across that of Counter and parallel flow heat exchangers (Çangarli, 2008).

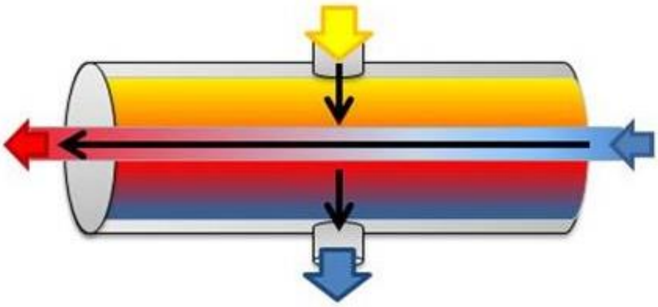

Figure 2.4. Cross-flow diagram (Anonymous.2, 2020)

Hybrid flow principles

Hybrid flow principle, this type of configuration is the combination of parallel and cross-flow principle. Some heat exchanger design can undertake arrangements of multiple principles to pass within a single heat exchanger. Mostly this kind of arrangement are typically used to put up the limitations of application, i.e. space, budget cost or temperature and pressure supply (Nitsche and Gbadamosi, 2016).

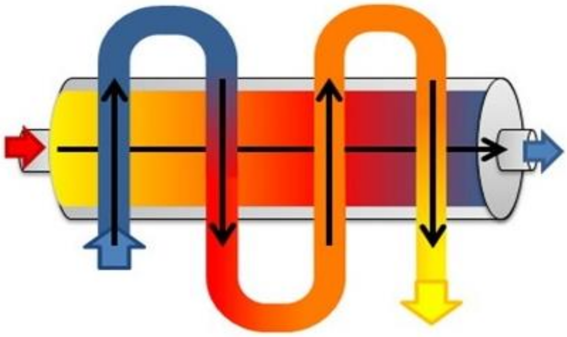

Figure 2.5. Hybrid flow diagram (Çengel and Ghajar, 2015)

2.1.2.2. Based on the construction method

Based on their construction characteristics, heat exchanger devices are classified by:

- Recuperative and regenerative
- Direct and indirect
- Static and dynamic.

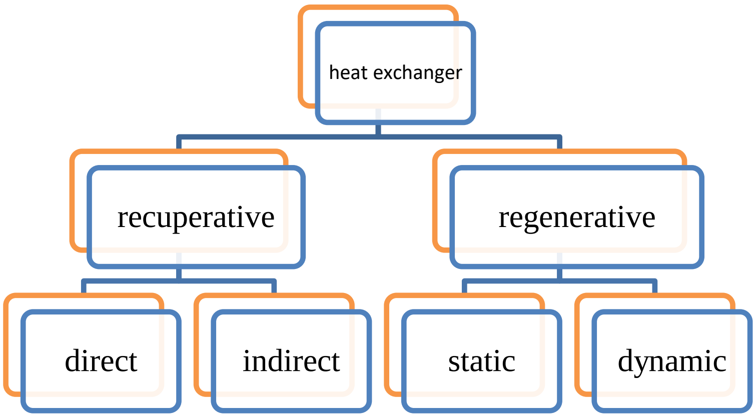

Figure 2.6. Heat Exchanger Classification by Construction

Recuperative and Regenerative

This Heat exchangers can be classified as recuperative and regenerative heat exchangers. There is difference in operation between these heat exchanging systems. *In recuperative heat exchangers*, through its channel inside the heat exchanger the fluids flow concurrently. On the other hand, *regenerative heat exchangers*, it allows through similar channel that the cooler and warmer fluids to flow. Additionally both of them could further divided into different categories, for instance direct or indirect and static or dynamic, respectively (BCS Incorporated, 2008). Among the two types showed, recuperative are most universally working throughout the industries (Ramesh and Dušan, 2003).

Direct and Indirect

Recuperative heat exchangers, to exchange heat between fluids (air), it applies either direct or indirect contact transfer processes (Ramesh and Dušan, 2003).

With *direct contact heat exchangers* system, the heat transfer from one fluid (air) to another through direct contact, and the fluids are not separated inside the device (Ramesh and Dušan, 2003).

On the contrary, in *indirect heat exchangers* system, all the way through the heat transfer processes the fluids (air) thermally conductive components will cause them to remain separated from one another and these components are tubes and plates. While the hotter fluids flow through the channel, it transfers its heat to the colder fluid which flow through another channel over the hot channel. Some devices use direct contact transfer process system steam injector and cooling towers are among them. Indirect contact transfer processes include tubular or plate heat exchanger (Ramesh and Dušan, 2003).

Static and Dynamic

- Static heat exchangers
- Dynamic heat exchangers

In *static regenerators* (fixed bed regenerators), the components and materials of the heat exchanger remain at the stationary while the fluids flow through the device (Çangarli, 2008).

Dynamic regenerators: the components and materials pass throughout the heat transfer process (Anonymous.3, 2020). During manufacturing, careful design consideration is necessary, because both types are at risk of cross-contamination between the flows (Çangarli, 2008).

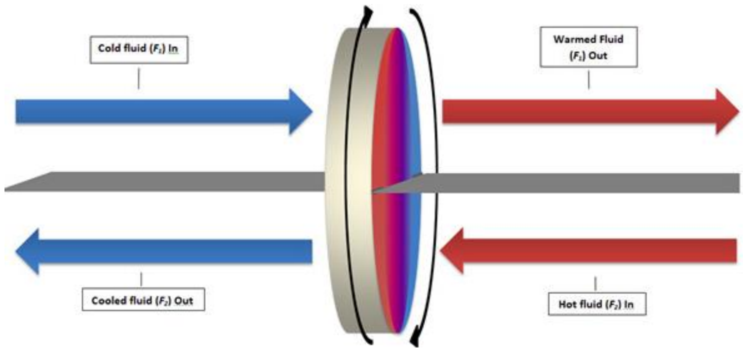

Figure 2.7. Heat Transfer in a Rotary-Type Regenerator (Çangarli, 2008)

2.1.2.3. Based on heat transfer mechanism

Heat transfer mechanisms employed by heat exchangers has 2 types;

- Single-phase transfer mechanism
- Two-phase heat transfer mechanism.

- In *single-phase transfer mechanisms*- In this process, both the warmer and the cooler fluids stand with the same state (without changing) of matter while they enter and exit the heat exchanger and during the process the fluids do not change to any other forms (phases). For instance, in applications of flue gas to air heat transfer, the flue gas gives heat which is transferred to the air and this will not lead to a change in phase like to fluids or solids (Kuppan, 2013).
- *Two-phase heat transfer mechanism*- In this process, the fluids (air) will experience a phase change during the transfer process. The change in phase can occur in either one or both of the fluids (air) involved causing in a change from a gas to a liquid or liquid to a gas. Typically, more than a single-phase heat transfer mechanism, two-phase heat transfer mechanisms require more complex design considerations (Kuppan, 2013). Boilers, condensers, and evaporators are some of the types of two-phase heat exchangers.

2.1.3. Heat exchanger components and materials

There are various types of components (materials) that can be involved in heat exchangers and a wide range of materials are used to construct them. These materials are dependent on the type of heat exchanger and its intended applications (Mukherjee, 1998). Some components used commonly to construct heat exchangers include shells, tubes, spiral tubes (coils), plates, fins, and adiabatic wheels. Metals are highly suitable and commonly used for the construction of heat exchangers because of their high thermal conductivity. Those metals are titanium, copper, and stainless. We can also use materials like graphite, ceramics, composites, and plastics depending on the requirements of the heat transfer applications.

2.1.4. Heat exchanger selection considerations

While there are a broad range of heat exchanger on the burden and available in the market, the suitability and its design in transferring heat between fluids is reliant on the specifications and requirements of the application. The main design of the wanted heat exchanger and influencing the equivalent rating and sizing calculations largely determined by those factors (Kuppan, 2013). Factors that professionals and design experts should keep in mind while designing are listed here;

- The type of fluids, the fluids stream, and their properties;
- The desired thermal outputs;
- Size limitations;
- And how much it costs.

2.1.4.1. Fluid type, stream, and properties

For particular heat transfer applications, the specific type of fluids involved and their physical, chemical, and thermal properties help to determine the best-suited flow configuration and construction (Anonymous.4, 2020).

For example, the heat exchanger design must be able to endure the high-stress conditions (corrosive, high-pressure fluids, or high temperatures) involved throughout the heating or cooling process. Selecting construction materials which hold the desired properties is the first method to fulfill these requirements:

- Graphite heat exchangers corrosion resistance and exhibit high thermal conductivity;
- Ceramic heat exchangers can endure (resist) temperatures higher than many commonly used metals melting points;
- Plastic heat exchangers can offer a thermal conductivity and a low-cost alternative which maintains a moderate degree of corrosion resistance.

Selecting a design suited for the fluid properties it the other method:

- A plate heat exchanger is capable of enduring (resisting) low to medium pressure fluids but with higher flow rates and;
- When using fluids that need a phase change in the transfer process, two-phase heat exchangers are essential.

The manufacturer or industry professionals may take into account other fluid and fluid stream properties, when choosing a heat exchanger include fluid viscosity, particulate matter content, fouling characteristics, and presence of water-soluble compounds.

2.1.4.2. Thermal outputs

The heat transfer amount between both fluids and the equivalent temperature change at the end of the transfer of heat process refers to the thermal output of a heat exchanger. As heat is removed, the temperature of one fluid will be lowered and as the other fluid heat is added the temperature will raised up. The conversion of heat within the heat exchangers will leads to the fluids temperature change. The optimal type and design of heat exchanger will be determined by the required thermal output and rate of heat transfer as some exchanger designs endure higher temperatures and provide greater heater transfer rates then other designs, even if at a higher cost (Anonymous.4, 2020).

2.1.4.3. Size limitations

Common mistakes will happen while purchasing one that is too big or too small for the given physical space after choosing the design and optimal types of heat exchanger. Oftentimes, Rather than choosing one which fully encompasses the space, purchasing a heat exchanging device with a size that leaves room for further expansion or addition is more prudent (Anonymous.4, 2020). Compact heat exchangers can offer high heat transfer efficiencies in smaller, more light-weight solutions for applications with limited space, such as in vehicles or aircraft (Ramesh and Dušan, 2003). For heat exchanging devices, several alternatives are accessible, including compact plate heat exchangers characterized by high heat transfer surface area to volume ratios. And these ratios are $\geq 400 m^2/m^3$ for the liquid to gas applications and $\geq 700 m^2/m^3$ (Khayal, 2018).

2.1.4.4. Costs

The cost of a heat exchanger includes the initial price of the equipment and also the maintenance, operational, and installation costs over the devises lifespan. Choosing a heat exchanger which effectively achieves the requirements of the applications are very necessary. The overall costs of the chosen heat exchanger to well determine whether the device is worthy of the investment is also very important (Anonymous.4, 2020). For instance, initially cheaper heat exchangers require several repairs and replacements of parts or completely within the same time, while the expensive heat exchangers may be initially expensive, but it’s more durable and may result in lower maintenance costs and, as a result, less overall spend over the courses of a few times.

2.1.5. Design optimization of heat exchangers

Determining the heat transfer coefficient, change of the temperature in the fluids, and the heat exchanger construction and relating them to the rate of heat transfer will be involved while designing the optimal heat exchanger for a given application (Anonymous.4, 2020). Calculating the device's rating and sizing are the two major problems which will arise in pursuing this objective.

- The rating refers to the calculation of the effectiveness of the thermal (i.e., efficiency and so on) of the exchanger of a given size and design, the rate of heat transfer, the amount of heat transferred between fluids (air), and their equivalent temperature change, and the total pressure drop throughout the device.

- The sizing refers to the calculation of the required total dimensions of the heat exchanger for an application with given process specifications and requirements. This sizing includes the length, height, width, thickness, components number and geometries, and arrangements, etc.,

The rating and sizing calculations will be affected by the design characteristics of a heat exchanger (material construction components, flow configurations, and geometry). The balance between the rating and sizing which fulfill the process specifications and requirements at the minimum necessary cost will be given by using the optimal heat exchanger design for specific applications (Anonymous.4, 2020).

2.2. Importance and benefits of waste heat capture

Today in most industries, more than half of the generated heat goes to waste of energy, whether it is a production industry or a power plant. There are few exceptions, whether its power plant or any other heavy industries, no heat engine is exactly efficient, automotive (vehicles) are among the worst achieving between 25 and 40 percent efficiency. In the case of coal and oil power plant, can be as low as 30 to 42 percent.

At the present system, for potential energy saving the prior alternative is waste heat recovery. For the reduction of fuel used to produce heat, heating efficiency increase, resulting in lower fuel use, waste heat recovery is so valuable. It serves as efficiency increasing and reduction of carbon emissions of the industries (Arzbaecher, et al., 2007).

As climate change makes energy efficiency an increasing urgency around the world, on each level of heat-producing industries and power plants, capturing waste heat is an important piece of not only boosting energy efficiency initiatives but ultimately of reducing greenhouse gas emissions. Improving energy efficiency without addressing waste heat is an effort that is inherently handicapped (Someone, et al., 2008).

Capturing waste heat empowers it to be sent to a function that would, otherwise be utilizing energy from the grid and prevents the consumption of power used to counteract the very effect of the waste heat itself (Someone, et al., 2008).

2.2.1. Economic benefits of waste heat recovery in power plants

The potential economy of waste heat recovery systems depends on capital recovery, which, in turn, depends on the annual fuel saving and cost. Fuel-saving can be challenging to predict because it is dependent on the heat load availability and time distribution of waste heat (Arzbaecher, et al., 2007).

Based on technical achievability, capital cost, and appropriate annual cost savings, most types of heat-recovery equipment will be determined. Using a simple payback period only can be dangerous. Instead, for an accurate comparison of alternatives proper discounted cash flow analysis should be in use (Sen, et al., 2019).

2.3. Types of heat exchangers

Regarding the design characteristics indicated above, there is a wide range of options to heat exchangers are available. Some of the regular options working throughout the industry include:

- Shell and tube heat exchangers
- Double pipe heat exchangers
- Compact heat exchanger
- Plate heat exchangers
- Condensers, evaporators, and boilers.

2.3.1. Shell and tube heat exchangers

There are several types of heat exchanger which are used in different industries and power plants. One of this heat exchanger is the shell and tube heat exchangers that was most widely used (Mukherjee, 1998). It is used widely in coal power plants (as pre-heater), petrochemical industries, and energy conservation and manufacturing industries. A shell and tube heat exchanger is a pressure vessel with many tubes (tube bundle) inside of a large tube (Thakore and Bhatt, 2007). Additionally, it has some baffles arranged according to their applications, tube sheets, and inlet channel and an outlet channel. There working principle is, the process fluids (air) flows through the tube tubes of the exchanger while the other flows outside of the tubes within the shell. The shell side and tube side fluids (air) are separated by a tube sheet. In this type of heat exchanger, one fluid (air) flows through the tubes while the other flows in the shell around the tube bundles (Khayal, 2018).

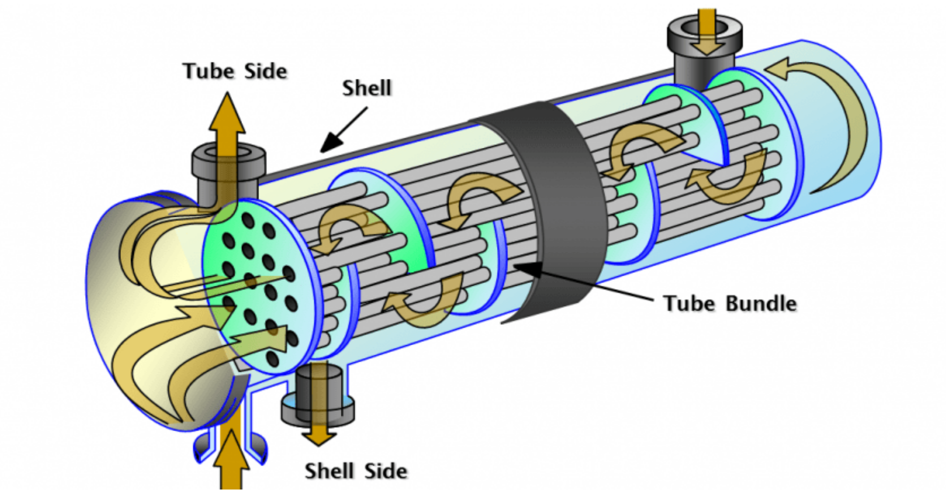

Figure 2.8. Shell and tube (Anonymous.5, 2020)

2.3.2. Double pipe heat exchangers

Double pipe heat exchangers are a form of shell and tube heat exchanger and use the simplest heat exchanger design and configuration, which includes two or more concentric, cylindrical pipes or tubes (one or smaller tubes and one larger tube). (Khayal, 2018) According to the designs of all shell and tube heat exchangers, in the smaller tube, the fluid flows through it while the one fluid flows within the larger tube and around the smaller tubes. The design requirements of double pipe heat exchangers are the fluids flow through their channels and remain separated across the heat transfer process (characteristics mentioned above in the indirect contact and recuperative types).

Despite that, in the design of double pipe heat exchangers, there is some flexibility. They can be designed with the parallel flow, counter-flow, cross-flow, or hybrid flow arrangements and to be used modularly in parallel, series or series-parallel configurations across the system. For instance, the transfer of the heat diagram below shows a parallel flow configuration within an isolated double pipe heat exchanger (Kuppan, 2013).

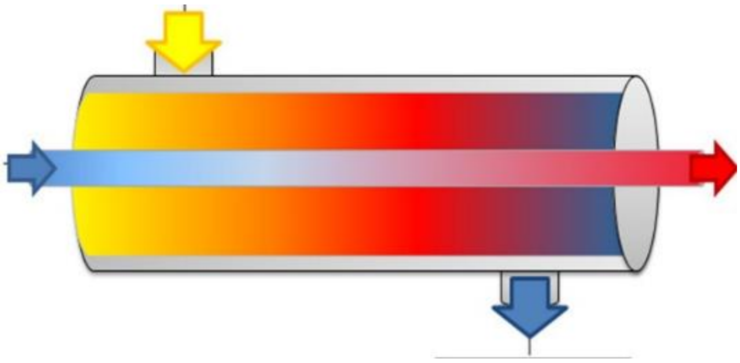

Figure 2.9. Parallel flow double Pipe Heat Exchanger (Çengel and Ghajar, 2015)

2.3.3. Compact heat exchangers

Compact heat exchangers are a device that offer high heat transfer efficiency within smaller, light-weighted and gives solution for employment within limited spaces, such as in all types of vehicles or aircraft. For heat exchanging devices, several options are available, including compact plate heat exchangers characterized by high heat transfer surface area to volume ratios (Zare, et al., 2016). And these ratios are ≥ 400m^2/m^3 for the liquid to gas applications and ≥ 700m^2/m^3 (Khayal, 2018). Pressure drops can be high because to increase the overall surface area they rely highly on the use of extended surfaces, while keeping the size to a minimum as possible (Thulukkunam, 2000). They are widely used in industrials and automotive such as oil coolers, intercoolers, automotive radiators, cryogenics, and electronics cooling applications.

Figure 2.10. Compact heat exchanger (Anonymous.6, 2019)

2.3.4. Plate heat exchangers

Also referred to as plate type heat exchangers and they are constructed of several thin, corrugated plates bundled together (Ruoxu, et al., 2014). Each pair of plates creates a channel. Through this channel, the fluid (air) will flow and the second passage is created between pairs through which the other fluid (air) can flow (Jouhara, et al., 2018). The pairs are stacked and attached via brazing, bolting, or welding.

Plate-fin or pillow-plate heat exchangers are also available with some variations in standard plate design.

- Plate-fin exchangers allow for multiple flow configurations and more than two fluid streams to pass across the device and employ spacers or fins between plates.
- Pillow-plate exchangers across the surface of the plate it applies pressure to the plates to amplify the heat transfer efficiency.

Some other types of plate heat exchangers available include plate and frame, plate and shell, and plate.

Figure 2.11. Plate type heat exchanger (Anonymous.7, 2020)

2.3.5. Boilers, Condensers, and Evaporators

Boilers, Condensers, and Evaporators employing a two-phase heat transfer mechanism heat exchangers. In this process, the fluids (air) will experience a phase change during the transfer process. The change in phase can occur in either one or both of the fluids (air) involved causing in a change from a gas to a liquid or liquid to a gas (Khayal, 2018).

Condensers are devices that take heated gas or vapor and cool it to the point of condensation, changing the gas or vapor into a liquid. Besides this, the heat transfer process changes the fluids from liquid form to gas or vapor form in boilers and evaporators (Khayal, 2018).

2.4. Typical examples of waste heat recoveries

2.4.1. Economizer

Stack economizers are the type of waste heat recovery that employs the simplest and most commonly used to heat water and air. This stack economizers use heat energy from the gas evicted in the heating process and send it to the stack where the economizer installed to heat boiler feed water and reduce the amount of energy required to make steam. The thermal efficiency of a boiler will rise and reduce fuel consumption by 5 - 10% when we install a boiler feed water economizer (Anonymous.2, 2020).

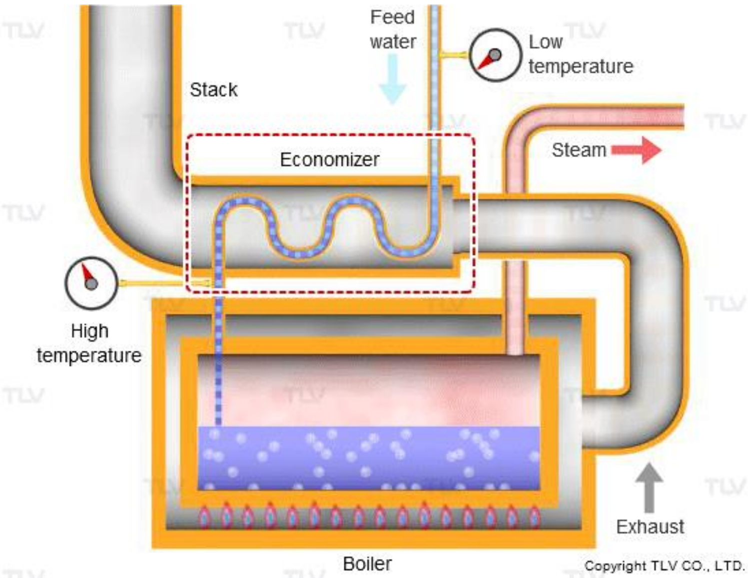

Figure 2.12. Economizer (Anonymous.2, 2020)

2.4.2. Waste heat boiler

The heat generated in the combustion system or exothermic chemical reactions at industrials and power plants can be recovered by waste heat boilers (WHB). These industries may contain significant energy that should not be wasted up a stack and left to the surrounding environment. Rather, to generate low-to-medium pressure steam in a waste heat boiler (WHB) we have to capture it. The steam generated in this process will be used for heating applications, or to drive turbines that generate electricity, compress vapors, or pump liquids. It is recommended that a high-efficiency separator and steam trap combination should be installed because of waste heat boiler steam may contain significant wetness. It will ensure that the waste heat boilers deliver optimal quality steam to the receiving process (Anonymous.2, 2020).

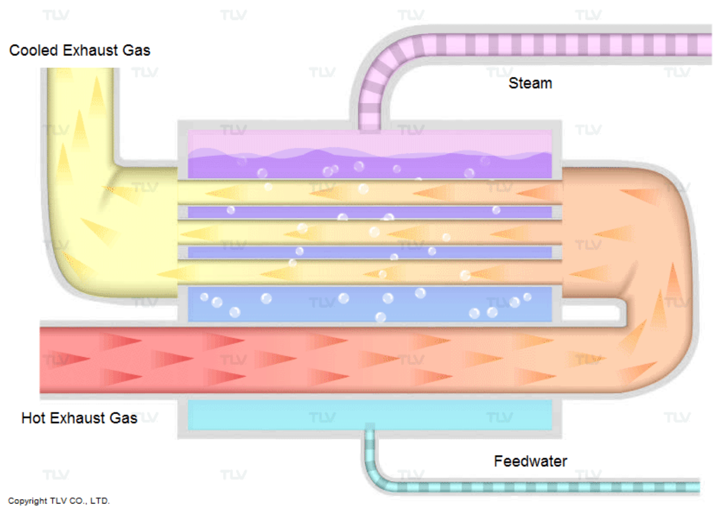

Figure 2.13. Waste heat boiler (Anonymous.2, 2020)

2.4.3. Heat Recovery steam generator

A heat recovery steam generator (HRSG) is an energy recovery heat exchanger that recovers heat from a hot flue gas streams produced in a combustion turbine or other waste gas streams of power plants. It produces steam that can be used to drive a steam turbine or used in a process. A heat recovery steam generator (HRSG) claims a high thermal efficiency and produces minimal CO_2 emissions. A heat recovery steam generator (HRSG) is a type of heat exchanger that recovers heat from the exhaust gasses or stack of industrial plants to an extreme degree. A gas turbine /combustion system is fired using coal or natural gasses and its exhaust contains extremely hot flue gases and hot vapor that would simply be expelled to the atmosphere (ambient) without capturing this kinetic energy (wasted heat energy). Many highly efficient power plants and industries (with cogeneration) uses a gas turbine to generate electricity then create steam from the wasted heat using a heat recovery steam generator (Jouhara, et al., 2018).

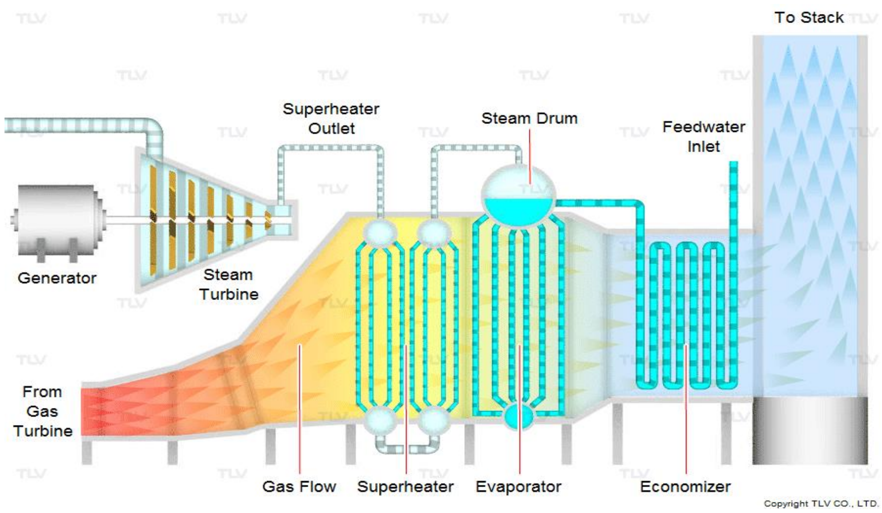

Figure 2.14. Heat recovery steam generator (Anonymous.2, 2020)

2.5. Utilization of waste heat

Utilization of waste heat from coal power plant helps in treating wasted heat and non-usable waste as well as recovering the valuable energy resource into air pre heat or other forms (Eboh, et al., 2019). Internally and/or externally are the two ways to utilize the recovered waste heat.

1. Utilizing waste heat internally can be conducted in 3 steps;
   - ✓ First, reducing the heat loss with thermal insulation improvements and process optimization, this process is before starting the utilization of waste heat
   - ✓ Secondly, using as a fuel supply for the production process or the building's heat supply, we have to reintegrate the waste heat into production processes by waste heat recovery
   - ✓ Thirdly, transforming the wasted heat energy into other useful energy forms to electricity, thermal cooling, or other forms.

Those processes can be carried out internally.

2. The second option is the cross-company waste heat utilization (external utilization),

This option allows us to share the wasted heat that can't be used internally will be used by other parties, in residential or commercial, and so on.

At present, the spatial proximity of the waste heat source and demand is required for the most economically achievable utilization of waste heat. The easiest and flexible form of energy conversion seems to be electricity, because of its easy transportable and distributed to the utility company or other parties. However, there is still an obstacle that needs to be addressed better with low technology efficiency in electricity generation from waste heat (Shawabkeh, 2015).

2.6. Sources of waste heat

Sources of waste heat are vast and different. Some of the wastes that can be recovered for practical uses consist of power plant exhaust, industrial exhaust, cooling tower, car exhausts, thermoelectric generation, and turbines. Depending on the application, the heat itself may be the coveted product or may be subjected to every other method to offer within the same process, or it can be transferred to an external process for use (Baradey, et al., 2015). Some study suggests that recovering the energy wasted from industrial facilities could fulfill up to 20% of total domestic electricity demand and simultaneously effect of up to a 20% reduction in GHG emissions (Jelena, et al., 2018). Some of the largest known waste heat resources are:

- Combustion exhausts
- Process of gases
- Cooling tower
- Convective, conductive, and radiative losses from pieces of equipment and heated products.

2.7. How does waste heat recovery systems work?

The process of heat recovery works by continually converting waste heat that is extracted from the source of waste heat and transforming it into a hot fluid (air). The system that is integrated into the exhaust (chimneys) are responsible for the process of a heat exchange unit in the heat recovery system. The exhausted flue gasses can have a temperature up to 1200°C and using for recovery that can cool down the wasted heat to 80°C. generally, a heat exchanger unit works through transferring the heat onto a medium like air, water, oil, ceramics, plastic, or metal which can be used for different purposes in the industry sectors or others (Anonymous.8, 2020).

2.8. Factors affecting waste heat recovery achievability

Characterizing waste heat source and the stream to which the heat can be transferred will help us to evaluate the achievability of waste heat recovery. Some of the important waste stream considerations that must be determined include (BCS Incorporated, 2008);

- Heat quantity
- Heat temperature (quality)
- Compositions
- Least allowed temperature and
- Operating schedules.

Analysis of the quality and quantity of the stream and also provide awareness into possible materials or design limitations will be seen under those listed above considerations (Amiri and Rahim.M, 2015).

2.9. Developing of a waste heat recovery system

2.9.1. Process understanding

During the process of development in waste heat recovery system, understanding the process is crucial. To be achieved someone have to study the process flow sheets, diagrams of the layout in the process, electrical and instrumentation cable ducting in the industry, and piping isometrics. In detail, analyzing these documents will help as in pinpointing;

- The waste heat sources and their uses;
- In the plant, arising upset conditions due to recovery of waste heat,
- Space accessibility;
- Any other constraint and limitations, such as dew point occurring in the equipment in the industrials.

Once we pinpoint the sources using the documents we will carry on to the next step, which is a selection of a proper recovery system and equipment and utilization.

2.9.2. Economic evaluation of waste heat recovery system

Economic analysis (financial analysis) based on investment, payback period, depreciation, rate of return, and so on are necessary to assess the selected waste heat recovery system (Jelena, et al., 2018). Additionally, consultation of experienced or manufacturers has to be obtained for additional benefits (balanced choice).

2.10. Basic components of shell and tube heat exchangers

The purpose of this entry is to give some information and guidelines on shell and tube heat exchanger (STHE) for apply on recovery of waste heat. We have various applications in the coal power plants and industrial sectors where waste heat recovery can be used cost-effectively. Among these applications, the bests are situations in which the waste heat stream is of high quality, produced continuously all through the year, with a heat load that fulfills the waste heat availability. We have also numerous pieces of equipment that can be used for waste heat recovery in large industries and power plants. These include STHE, coil heat exchangers, plate and frame heat exchangers, and so on. For our purpose of evaluation of waste heat, we will use shell and tube heat exchanger and explain it briefly.

Shell and tube heat exchangers are the most multipurpose type of heat exchangers. We can use them in bulky (large) process industries and power plants. This type of heat exchangers provide relatively large ratios of heat transfer area to volume and weight and that they can be easily cleaned. Shell and tube heat exchangers offer great flexibility to meet nearly any service requirement (Thakore and Bhatt, 2007). These heat exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids (air) streams (Kuppan, 2013).

Shell and tube heat exchangers are put up of round tubes mounted in a cylindrical shell with the tubes parallel to the shell. One fluid (air) stream flows inside the tubes, while the other fluids (air) stream flows across and along the axis of the exchanger. The main components (parts) of this exchanger are shell, tubes (tube bundle), front-end head, rear-end head, baffles, and tube sheets (Ramesh and Dušan, 2003).

Figure 2.15. Tube bundles of Shell and tube heat exchanger (Anonymous.9, 2019)

2.10.1. Types of shells

Several fronts, rear head, and shell types are standardized by TEMA (Tubular Exchanger Manufacturers Association) (Thakore and Bhatt, 2007).

The E shell - is the most common because of its low cost and ease (Mukherjee, 1998).

J and X shell – for low-pressure drop, this type of shell is used in the designing application. Some of the applications are condenser in vacuum (Mukherjee, 1998).

G and H shell – this type of shell are used for specific applications such as single-phase flows but are very often used as a horizontal thermosiphon reboiler (Mukherjee, 1998).

The K shell -is a kettle reboiler with the tube bundle in the bottom of the shell and it covers about 60% of the shell diameter. We use this shell when a portion of a stream needs to be vaporized (Thakore and Bhatt, 2007).

2.10.2. Tube Bundle

Tube bundles are found inside shell and tube heat exchangers and placed within a cylindrical shell, where fluids at different temperatures pass through and over tubes. These fluids can be either gases or liquid (Thakore and Bhatt, 2007).

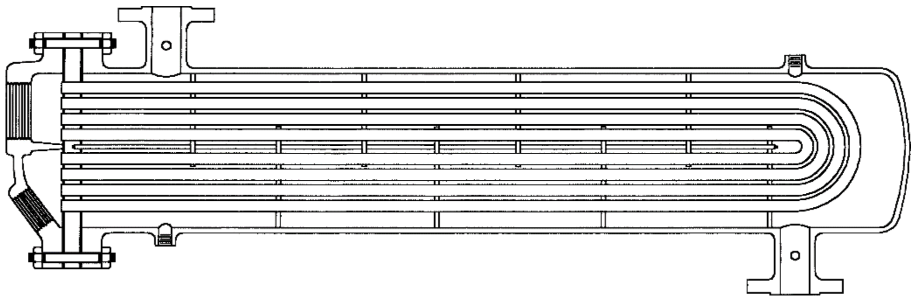

Figure 2.16. Bare U-tube, baffled single pass shell (Thakore and Bhatt, 2007)

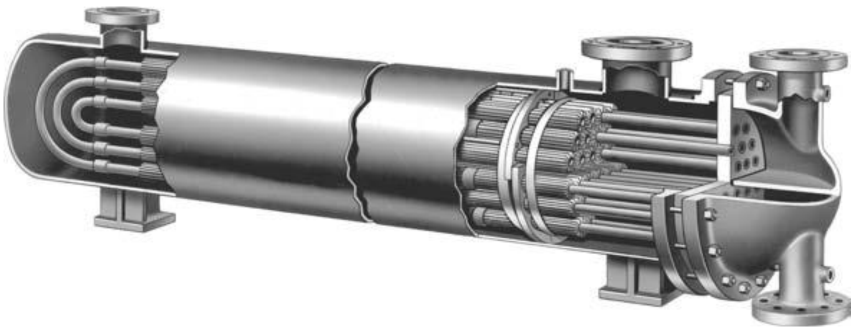

Figure 2.17. Finned U-tube (Thakore and Bhatt, 2007)

2.10.3. Tubes and Tube Passes

Using a large number of tube passes increases tube side fluids (air) velocity and the heat transfer coefficient (within the available pressure drop) and it will minimize fouling (Thulukkunam, 2000). Usually tube metals are made from low carbon steel, steel (low alloy), stainless steel, cupronickel, copper, aluminum (alloy form), or titanium and they are good heat conductors. For special applications we may select other materials. The Birmingham Wire Gage (BWG) standardizes the wall thickness of tubes of the heat exchanger. Using long tubes (for any given total surface) will benefit us for fewer tubes, fewer holes drilled and decreasing diameter of the shell, causing in lower costs (Thakore and Bhatt, 2007).

2.10.4. Layout of tubes

The layout of tube is considered (measured) via the involved angles between the tubes. Tube pitch (P_T) is usually selected so that the pitch ratio ($^{P_T}/_{d_o}$) is between 1.25 and 1.5 (Kuppan, 2013). The tube sheet becomes structurally weak when the tubes are too close to each other. The tube locations and layout have been standardized (Thakore and Bhatt, 2007). The tube number (tube counts) can be placed inside of the shell can be depends on:

- Tube layout used
- Outside diameter of the tube
- The diameter of the shell.
- Pitch size
- Number of passes

2.10.5. Baffle types and their geometry

Baffles has 2 different functions and they are listed here as follows:

1. Supporting the tubes to be structurally rigid, sagging and preventing the tubes vibration.
2. Diverting the flow across the bundle to achieve a higher heat transfer coefficient.

Baffles can be classified as longitudinal and transverse types (Mukherjee, 1998). The transverse baffles can be classified as plate baffles and rod baffles.

The single and double segmental baffles are the most frequently used ones. They divert the flow efficiently transversely on the tubes. The optimum baffle spacing amount is 0.4 up to 0.6 of the diameter of the shell (04 or $0.6D_s$) and commonly recommended baffle cut 25% to 35% (Thulukkunam, 2000).

For low-pressure drop applications and which are about 0.5 and 0.3 of the segmental value, we will use triple segmental baffles and no-tubes-in-window (Thakore and Bhatt, 2007).

Disc and ring (doughnut) baffles are composed of alternating inner discs and outer rings, which directs the flow radial transversely to the tube fields.

An orifice baffle is in which shell-side fluids (air) flows through the clearance between baffle-hole diameter and tube outside diameter.

Rod or grid baffles are made by a grid of strip or rod supports. The flow is essentially longitudinal and brings about a very low-pressure drop (Thakore and Bhatt, 2007). Because of the close baffles spacing, the tube vibration danger will almost be rejected. This type of construction can be used effectively for vertical reboilers and condensers (Thakore and Bhatt, 2007).

Advantages of the Shell and tube heat exchangers

- It is less expensive when compared to compact and plate type heat
- It can be used in a system with higher temperatures and pressures.
- Its orientation can vary depending or the demand (i.e. either vertical or horizontal)
- The pressure drop across the shell and tube heat exchanger is less
- Detecting leakages in shell and tube heat exchanger is easy with the use of a pressure test
- The shell and tubes can be made with different materials and this substantial flexibility with regards to the choice of materials allow the exchanger to accommodate corrosion and other concerns.
- Cleaning and repair of the shell and tube exchanger are comparatively straight forward since it can be dismantled.

Disadvantages of shell and tube heat exchanger

- It is large and expensive per unit of heat transfer surface area.
- Relatively it has low heat transfer efficiency.
- It occupies a large space in comparison to other heat exchangers.
- Heat transfer efficiency is less compared to other heat exchangers.
- The capacity of the shell and tube heat exchanger cannot be increased.

2.11. Fouling of heat exchanger surface

The formation and buildup of unwanted constituents on the surfaces of processing equipment are called fouling (Thakore and Bhatt, 2007). It can seriously deteriorate the capability of the surface to transfer heat under the temperature difference conditions that it was designed for. Fouling of heat transfer surfaces is one of the major problems in heat transfer equipment. It is an exceptional multifaceted phenomenon. We can characterize fouling as a combined, unsteady state, momentum, mass and heat transfer problem with chemical, solubility, and corrosion (Thulukkunam, 2000). Fouling affects the operation of the apparatus in two ways (Khayal, 2018):

- Fouling layers will have low thermal conductivity and it will increase the resistance of heat transfer. This will reduce the effectiveness of heat exchangers.
- When deposition takes place, the cross-sectional area is reduced and causes an increase in pressure drop through the heat transfer equipment.

In the design of heat exchangers, fouling is considered the most unknown factor (Thulukkunam, 2000). The wide range of the operating conditions and process streams present in the industry tends to make most fouling situations unique. This makes a general analysis of the problem difficult. Generally, in many industrial processes, the ability to transfer heat with efficiency remains a central feature. To improve the understanding of heat transfer mechanisms and the development of appropriate correlations and techniques which will be applied to the design of heat exchanger have been paid plenteous attention. At the same time for the problem of surface fouling in heat exchangers comparatively, a little consideration has been given. To eliminate or reduce the problem in guaranteed situations we should have a better understanding of the problem and the mechanisms that lead to the buildup of deposits on the surface. It will occur as a consequence of the fluids or air being handled and their constituents in mixture with the operational conditions like velocity and temperature (Thulukkunam, 2000). Heat exchanger foulants are any solid or semi-solid material will be. Some materials that are usually stumbling upon in industrial operations as foulants includes (Khayal, 2018):

- Inorganic materials:

    - Airborne dust and grit
    - Waterborne mud and silts
    - Calcium and magnesium salts
    - Iron oxide.
- Organic materials:
    - Oils, waxes, and greases
    - Heavy organic deposits, e.g. polymers, tars
    - Carbon.

A factor in the economics of a particular process is a conservation of energy. Simultaneously concerning the remainder of the process equipment, the proportion of capital cost that is required to have in heat exchangers is comparatively low. As a result of this heat, exchanger fouling has been neglected and most of the fouling issues are unique to a particular process and heat exchanger design (Thulukkunam, 2000). The problem of heat exchanger fouling thus represents a challenge for all, not only in terms of heat transfer technology but also in the wider aspect of economics and environmental suitability (Khayal, 2018).

3. MATERIALS AND METHODS

Power plants using coal energy exerts waste heat energy through flue gas. As a seconder energy resource, heat recovery is the best method to decrease energy production cost, decrease CO_2 and other contaminant elements and increase facility efficiency. First before starting economic and environmental evaluation has begun applicable STHE should be designed depending on the given values. After all, relations were derived on the shell and tube heat exchanger, we will drive the next step on the saving of coal and CO_2.

3.1. Design Procedure of STHE using Kern method

During design, before the next schedule of plant cleaning, the selected shell and tube heat exchanger (STHE) should fulfill the requirement process with acceptable pressure drop. First, we have to identify the problem as completely as possible. A primary estimation will be done as described in the next stages. Then the rating of the primary design will follow. Finally, the thermal performance and pressure drop for both (inlet and outlet) streams can be calculated according to the steps given below (Shawabkeh, 2015).

3.1.1. Preliminary estimation of unit Size

To obtain the size of a heat exchanger we use Equation 3.1:

$$A_o = \frac{Q}{U_o \Delta T_m} = \frac{Q}{U_o F \Delta T_{lm,cf}} \tag{3.1}$$

Where: A_o- heat transfer surface area of heat transfer on the outside tube

Q - The heat duty of the exchanger, kW.

First, we will estimate using fouling factors, the individual heat transfer coefficient. This estimation is preferable for the overall heat transfer coefficient. Based on the tubes of outside diameter, the overall heat transfer coefficient (U_o) will be estimated from the estimated values of individual heat transfer coefficients, the overall surface efficiency and wall, and fouling resistance using Equation 3.2.

$$\frac{1}{U_o} = \frac{A_o}{A_i}\left(\frac{1}{\eta_i h_i} + \frac{R_{fi}}{\eta_i}\right) + A_o R_w + \frac{R_{fo}}{\eta_o} + \frac{1}{\eta_o h_o} \tag{3.2}$$

We will have to determine the distribution of the thermal resistances under both (clean and fouled) conditions.

For single (1) tube pass, purely counter-current heat exchanger F = 1.00

For the initial design with any even numbers of tube side passes F = 0.9(estimated)

We can estimate heat load from the heat balance as:

$$Q = \left(\dot{m}c_p\right)_c (Tc_2 - Tc_1) = \left(\dot{m}c_p\right)_h (Th_1 - Th_2) \tag{3.3}$$

If one stream change phase,

$$Q = \dot{m}h_{fg} \tag{3.4}$$

Where $(\dot{m})$ is the mass of the stream changing phase per unit time and (h_{fg}) is the latent heat of the phase change. We must calculate the LMTD $(\Delta T_{lm,cf})$ for counter-current flow from the four given inlet and outlet temperatures. If three (3) out of four (4) temperatures are known, we can find the fourth (4) from the heat balance:

$$\Delta T_{lm,cf} = \frac{\left(T_{h_1} - T_{c_2}\right) - \left(T_{h_2} - T_{c_1}\right)}{\ln \frac{T_{h_1} - T_{c_2}}{T_{h_2} - T_{c_1}}} \tag{3.5}$$

Here, the challenge is converting the calculated area from *Equation 3.1* into reasonable dimensions for the first trial. Our biggest objective now is to get the correct number of tubes diameter (d_o) and the diameter of the shell (D_s) to accommodate the tube numbers (N_t) with the given length of the tubes (L).

$$A_o = \pi d_o N_T L \tag{3.6}$$

The shell diameter (D_s) and it contains the right number of tubes (N_t) of diameter (d_o). The total number of tubes (N_t) can be estimated in the rational (fair) approximation is the function of the shell diameter by taking the shell circle and dividing it by the expected area of the tube layout refers to a single tube (A_1);

$$N_t = (CTP)\frac{\pi D_s^2}{4A_1} \tag{3.7}$$

Tube count calculation constant (CTP), which accounts for the insufficient coverage of the shell diameter by the tubes due to crucial clearances between the shell and outer tube circle and tube faults due to tubes pass lanes for multi-tube pass design. Depending on a fixed tube sheet, the following table values are suggested:

Number of passes	CTP value
One tube pass	0.93
Two tubes pass	0.90
Three tubes pass	0.85

$$A_1 = (CL)P_T^2 \tag{3.8}$$

CL: Layout of the tubes constant.

CL =1.0= sin 90 for 90° and 45°

CL =0.87= cos 30 for 30° and 60°

We can rewrite *Equation 3.7* as:

$$N_t = 0.785\left(\frac{CTP}{CL}\right)\frac{D_s^2}{(PR)^2 d_o^2} \tag{3.9}$$

And where PR is the tube pitch ratio ${}^{P_T}/_{d_o}$

Expressing the diameter of shell in terms of basic constructional diameter can be achieved by substituting number of tubes (N_t) from *Equation 3.6* in to *Equation 3.9*:

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_o (PR)^2 d_o}{L}\right]^{1/2} \tag{3.10}$$

3.1.2. Rating of the preliminary design

For the rating process, as the input into the heat transfer and the pressure drop correlations, all the preliminary geometrical calculations must be carried out. In this rating process, for every stream specified, heat transfer coefficient and pressure drop calculations are the two basic assessment. When the heat exchanger length is fixed, the rating program calculates the outlet temperatures of both (outlet and inlet) streams. The rating program result is the length of the heat exchanger needed to satisfy the fixed heat duty of the exchanger when the heat duty (heat load) is fixed. In both cases, the streams (both) in the heat exchanger pressure drops are calculated. While on process, if the analysis of the rating output is not acceptable or below allowable output, a new geometrical modification must be made, and here are some suggestions (Kern, 1965).

- To increase the tube side heat transfer coefficient, increase the tube side velocity, and that increases the number of tube passes,
- To increase the shell side heat transfer coefficient, we have to reduce baffle spacing between the baffles or reduce the cutting area of baffles,
- Increasing the shell diameter and length of the heat exchanger can increase the area and adding multiple shells in series can increase the area too,
- To decrease the tube side pressure drop, when it's higher than the acceptable value, we reduce the number of tube passes or increase the tube diameter which can decrease the tube length and increase the number of tubes and shell diameter,
- When shell side pressure drop is greater than the allowable value, we can increase baffle cut, tube pitch, and baffle spacing or changing the types of baffles.

3.1.3. Heat transfer of shell side tube and pressure drop

The prediction of overall transfer coefficient needs assessment of the tubes and heat transfer coefficient of the shell side. The detail assessment (calculation) described below is called the *Kern* method.

3.1.3.1. Shell side heat transfer coefficient

The shell side heat transfer coefficient is the heat transfer coefficient outside the tube bundles. Shell side heat transfer coefficient correlations;

$$\frac{h_o D_e}{k} = 0.36\left(\frac{D_e G_s}{\mu}\right)^{0.55}\left(\frac{c_p \mu}{k}\right)^{1/3}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3.11}$$

$$for\ 2\times 10^3 < Re_s = \frac{G_s D_e}{\mu} < 1\times 10^6$$

h_o = heat transfer coefficient of shell side

D_e = equivalent diameter on the shell side

G_s = mass velocity of shell side.

We can evaluate these properties at the average fluids (air) temperature in the shell. The equivalent diameter of the shell is;

$$D_e = \frac{4 \times free\ flow\ area}{wetted\ parimeter} \tag{3.12}$$

For every pitch layout, we can apply Equation 3.12.

For the square pitch:

- Circumference of the circle is the perimeter.

- Pitch size square (P_T^2) minus the shaded section (area of the circle) will be the area.

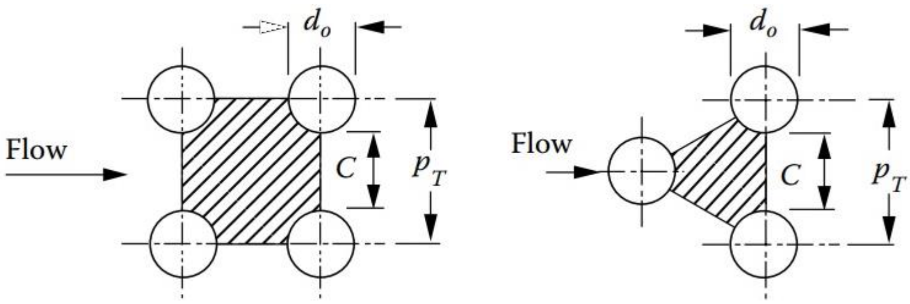

Figure 3.1. Square and triangular pitch-tube layouts respectively.

We can write as follows for the square pitch:

$$D_e = \frac{4(P_T^2 - \pi\, d_o^2/4)}{\pi d_o} \tag{3.13}$$

And for the triangular pitch, we can write as:

$$D_e = \frac{4\left(\frac{P_T^2\sqrt{3}}{4} - \frac{\pi d_o^2}{8}\right)}{\pi\, d_o/2} \tag{3.14}$$

d_o - *The tube outside diameter.*

Fictitious values of shell side mas velocity (G_s) will be defined based on the bundle crossflow area at the hypothetical tube row possessing the highest flow area corresponding to the center of the shell and this is because of no free flow are on the shell side by which mass velocity of shell side (G_s) can be evaluated. This velocity can be affected by those variables;

- shell diameter (D_s)
- the pitch size (P_T)
- the clearance (C) between adjacent tubes and
- The baffle spacing (B)

The flow area width at the tubes positioned at the middle of the shell is $\left({D_s}/{P_T}\right) C$ and the flow area length is taken as the spacing in baffle (B). And so, the cross-flow area (A_s) of the bundle at the midpoint of the shell will be;

$$A_s = \frac{D_s C B}{P_T} \tag{3.15}$$

Shell tubes inside diameter is(D_s).
Then we have mass velocity of shell-side correlation as;

$$G_s = \frac{\dot{m}}{A_s} \tag{3.16}$$

3.1.3.2. Pressure drop on shell side

It depends on the number of tubes the fluids (air) passes through the tube bundle between the baffles as well as the length of each crossing. A correlation has been achieved by using the product of distance across the bundle, taken as the inside diameter of the shell (D_s) and the number of times the bundle will be crossed. For calculating the pressure drop and heat transfer we use the same equivalent diameter.

The shell side pressure drop is calculated by the following expression.

$$\Delta P_s = \frac{f G_s^2 (N_b + 1) \cdot D_s}{2 \rho D_e \phi_s} \tag{3.17}$$

$\phi_s = \left({\mu_b}/{\mu_w}\right)^{0.14}$, $N_b = {L}/{B} - 1$ baffle number, and $N_b + 1$ is the number of times the shell fluids (air) passes the tube bundle. For the shell friction factor (f) is calculated from:

$$f = exp(0.576 - 0.19 \ln Re_s) \tag{3.18}$$

Where

$$400 < Re_s = \frac{G_s D_e}{\mu} \leq 1 \times 10^6$$

Note: this correlation based on the data obtained on actual heat exchanger has been tested and become acceptable to be used.

Entrance and exit losses have been taken into account on the friction coefficient.

3.1.3.3. Pressure drop of tube side

To calculate the pressure drop of the tube side, tube pass numbers (N_p) and the length (L) of heat exchanger under assessment should be known.

$$\Delta P_t = 4f \frac{LN_p}{d_i} \rho \frac{u_m^2}{2} \tag{3.19}$$

Or

$$\Delta P_t = 4f \frac{LN_p}{d_i} \frac{G_t^2}{2\rho} \tag{3.20}$$

The pass, the change of direction brings an additional pressure drop (ΔP_r), due to sudden expansion and contractions that the tube fluids (air) experience during a return, which is accounted for letting four-velocity heads per pass;

$$\Delta P_r = 4N_p \frac{\rho u_m^2}{2} \tag{3.21}$$

The tube side total pressure drop becomes;

$$\Delta P_{total} = \left(4f \frac{LN_p}{d_i} + 4N_p\right) \frac{\rho u_m^2}{2} \tag{3.22}$$

4. RESULTS AND DISCUSSION

Utilizing waste heat has great potential for the production of clean by reducing energy consumption, improving the efficiency of energy, and rising the engineering functionality of coal power plants and heat energy-producing industries. In developing countries, utilizing of wasted heat is highly neglected in different industries and power plants. It focused on the common waste-heat sources and readily implementable technologies considering both technical and economic aspects.

4.1. Design analysis

Sample example on how to design a shell and tube heat exchanger, calculating the amount of air feed and amount of coal burned in the process with a detailed way using the Kern method.

Design a heat exchanger to heat air by the use of exhaust flue gas at T_1 = 250 °C and 10 bar and that can flow in the shell-side with a flow rate mass of 73,000 kg/hr. the heat should be transmitted to 27,000 kg/hr of air coming from the ambient (atmosphere) at 25°C (Cp=1005 J/kg·K). The preferable ones are the single pass and single shell if possible. The resistance of fouling is 0.0001 m^2 K/W is recommended and below 30% of surface over design is essential. Because of the limitation in space, the tube length should be not more than 5m. Carbon steel is used for tube material (k=60$W/m \cdot K$). The ambient air flows through ¾ inch (straight tubes with OD of 19 mm and ID of 16 mm). The tubes arrangement will have a pitch ratio of 1.25 on square pitch. 0.6 of the shell diameter will be the baffle approximation and 27% cut of baffles. 10 bar is the permitted maximum shell side pressure drop. The air that goes to combustion chamber should not be less than 85°C.

A. Perform the complete analysis preliminary design.
B. Consumption of coal in the shell and tube that you perform
C. The actual amount of air

4.1.1. Design analysis (Solution)

Note that the value of Q is the total amount of waste heat that is ideally available for recovery, but not the total amount of waste heat that will be recovered. Not all of this waste heat will be recovered or even can be recovered with the recent technologies available so far.

4.1.1.1. Calculation of heat duty and exit temperature of hot stream

$$Q = \left(\dot{m}c_p\right)_c (Tc_2 - Tc_1)$$

$$Q = \left(\left(\frac{27{,}000}{3600}\right)(1005\, J/kg \cdot K)\right)_c (85°\mathrm{C} - 25°\mathrm{C})$$

$$Q = 452.250\ kW$$

To get the outlet temperature of the hot air we use this formula;

$$Q = \left(\dot{m}c_p\right)_h (Th_1 - Th_2)$$

And Th_2 becomes as follows;

$$Th_2 = Th_1 - \frac{Q}{\left(\dot{m}c_p\right)_h}$$

$$Th_2 = 250 - \frac{452{,}250\ W}{\left(\left(\frac{73{,}000}{3600}\right)(1005\, J/kg \cdot K)\right)_h}$$

$$Th_2 = 227.8°\mathrm{C}$$

4.1.1.2. Heat transfer coefficient assumption

Taking from the *Table 2* heat transfer coefficient;

- *Shell side heat transfer coefficient = 250 W/ m² K=*h_o
- *Tube side heat transfer coefficient = 80 W/ m² K=*h_i

4.1.1.2.1. Heat transfer coefficient with fouling calculation

$$\frac{1}{U_f} = \frac{1}{h_o} + \frac{r_o}{r_i}\frac{1}{h_i} + R_{fi} + r_o\frac{\ln(r_o/r_i)}{K}$$

$$U_f = \left[\frac{1}{250} + \frac{19}{16}\frac{1}{80} + 0.0001 + \frac{0.019}{2}\frac{\ln(19/16)}{60}\right]^{-1}$$

$$U_f = 52.7\ W/m^2 \cdot K$$

4.1.1.2.2. Heat transfer coefficient without fouling calculation

$$\frac{1}{U_c} = \frac{1}{h_o} + \frac{r_o}{r_i}\frac{1}{h_i} + r_o\frac{\ln(r_o/r_i)}{K}$$

$$U_c = \left[\frac{1}{250} + \frac{19}{16}\frac{1}{80} + \frac{0.019}{2}\frac{\ln(19/16)}{60}\right]^{-1}$$

$$U_c = 52.9\ W/m^2 \cdot K$$

After we calculated U_f and U_c respectively, we will calculate the ΔT_{lm} (log mean temperature difference) from the four outlet and inlet temperatures.

4.1.1.3. Log mean temperature difference calculation

$$\Delta T_{lm,cf} = \frac{(\Delta T_1 - \Delta T_2)}{\ln\frac{\Delta T_1}{\Delta T_2}}$$

$\Delta T_1 = T_{h_1} - T_{h_2}$ = 250°C − 227.8 °C = 22.2°C

$\Delta T_2 = T_{c_2} - T_{c_1}$ = 85°C − 25 °C = 60°C

$$\Delta T_{lm,cf} = \frac{(\Delta T_1 - \Delta T_2)}{\ln\frac{\Delta T_1}{\Delta T_2}} = \frac{(27.8°\text{C} - 60°\text{C})}{\ln\frac{27.8°\text{C}}{60°\text{C}}} = 41.85°\text{C}\ by\ rounding\ of\ 42°\text{C}$$

We assume correction factor (F) for log mean temperature F=0.9, then the true mean temperature of effectiveness becomes

$$\Delta T_m = F\Delta T_{lm,cf}$$

$$\Delta T_m = 0.9(42°\text{C}) = 38°\text{C}$$

4.1.1.4. Heat transfer area calculation

Find the required area of heat transfer A_f (heat transfer area with fouling) A_c (heat transfer area without fouling) using *Equation 3.1*:

$$A_f = \frac{Q}{U_f \Delta T_m} = \frac{452.25\ kW}{(52.7\ W/m^2 \cdot K)(38°\text{C})} = \frac{452.25 \times 10^3}{2002.6} = 225.83\ m^2$$

$$A_c = \frac{Q}{U_c \Delta T_m} = \frac{452.25\ kW}{(52.9\ W/m^2 \cdot K)(38°\text{C})} = \frac{452.25 \times 10^3}{2010.2} = 224.97\ m^2$$

A_f/A_c The surface over the design value will be:

$$A_f/A_c = \frac{225.83\ m^2}{224.97\ m^2} = 1.03(03\%)$$

Therefore it is acceptable.

Fouling reduces the heat transfer system, so a longer pipe must be used to get the same air temperatures before feeding to the combustion chamber.

4.1.1.5. Shell diameter calculation, layout and tube size decision

Afterward, calculating the shell diameter (D_s) from *Equation 3.10*.

$$D_s = 0.637\sqrt{\frac{CL}{CTP}}\left[\frac{A_f(PR)^2 d_o}{L}\right]^{1/2}$$

$$D_s = 0.637\ \sqrt{\frac{1}{0.93}}\left[\frac{225.83(1.25)^2 0.019}{5}\right]^{1/2}$$

$$D_s = 0.765 \approx 0.8\ m$$

- d_o=0.019
- PR=1.25
- CTP=0.93 (for 1 tube pass)
- CL=1.0 (for 45° and 90°)
- L=5m (assumption length)
- A_f=225.83

The next procedure is calculating the number of tubes (N_t) using *Equation 3.9*.

$$N_t = 0.785\left(\frac{0.93}{1}\right)\frac{(0.8)^2}{(1.25)^2(0.019)^2} = 829\ tubes$$

For baffle spacing, we can use 0.4 to 0.6 of the shell diameter and for our case, we use 0.6.

$B = 0.6(D_s)$

$B = 0.6(0.8m)$

$B = 0.48m$ (And rounding off) to $0.5m$

- D_s = 0.8 m
- OD =19 mm and
- ID = 16 mm
- B = 0.5 m, baffle cut 30%)
- P_t/D_o= 1.25(with square pitch)

On the next stage, we perform the rating analysis, using these values D_s and N_t respectively, we select some values from *Table 9*. The selection depends on the closest value of the number on the table (exact value or exceeding N_t values). N_t Values we calculated are 829 and the TEMA (tubular exchanger manufacturers association) standard becomes 845 tubes for 1-P shell and tube heat exchanger. Using the kern method we will rerate the values according to the table's number.

- Shell internal diameter (D_s) = 31in (80 cm)
- Tube numbers $N_t = 845\ tubes$
- Diameter of the tube OD =19 mm and ID = 16 mm
- Material of the tube k=60 $W/m \cdot K$ (carbon steel)
- Baffle spacing B = 0.5 m, baffle cut 30%)
- Pitch size $P_T = 0.0237$
- Number of tube passes $N_p = 1 - P$

4.1.1.6. Properties of the shell side

Heat duty is fixed with assumed temperature (outlet) 85°C. Properties of the shell side fluids will be as follows;

$$T_b = {(250 + 227.8)}/{2} = 239°\text{C}$$

From *Table 7.*

For 239°C

- ρ = 0.689 kg/m^3
- μ_1 = 2.72 × $10^{-5} kg/m \cdot s$
- $C_p = 1033\ {J}/{kg} \cdot K$
- $k_f = 0.0407\ {W}/{m} \cdot K$
- $P_r = 0.68$

For 38°C

- ρ = 1.135 kg/m^3
- μ_2 = 1.907 × $10^{-5} kg/m \cdot s$
- $C_p = 1006\ {J}/{kg} \cdot K$
- $K_f = 0.0272\ {W}/{m} \cdot K$
- $P_r = 0.706$

  - The maximum length of tube $L_{max=}$ 5m

4.1.1.7. Shell side heat transfer, equivalent diameter and area calculation

Using Equation 3.11. (Shell side heat transfer Equation)

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu}\right)^{0.55} \left(\frac{c_p \mu}{k}\right)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

$$for\ 2 \times 10^3 < Re_s = \frac{G_s D_e}{\mu} < 1 \times 10^6$$

Using Equation 3.13, for square pitch tube layout, the equivalent diameter of the shell side (D_e) will be;

$$D_e = \frac{4(P_T^2 - \pi\, d_o^2/4)}{\pi d_o}$$

$$D_e = \frac{4\left[(0.0237)^2 - \pi\left({0.019^2}/_{4}\right)\right]}{\pi(0.019)}$$

$$D_e = 0.18 \approx 0.19m$$

Bundle cross-flow area (A_s) as the center of the shell will be evaluated by *Equation 3.15*.

$$A_s = \frac{D_s \cdot C \cdot B}{P_T} = \frac{(0.8m\ )(0.0047m)(0.5m)}{0.0237m}$$

$$A_s = 0.08\ m^2$$

4.1.1.8. Shell side mass velocity calculation

Then Shell side mass velocity (G_s) will be evaluated by *Equation 3.16*;

$$G_s = \frac{\dot{m}}{A_s} = \frac{73{,}000\ \text{kg/hr}}{0.08\ m^2} = 912{,}500\ kg/hr \cdot m^2$$

$$G_s = 912{,}500\ kg/hr \cdot m^2 \times \frac{1hr}{3600s}$$

$$G_s = 253.5\ kg/s \cdot m^2$$

Shell side Reynolds number will be calculated by *Equation 3.18*;

$$Re_s = \frac{G_s D_e}{\mu_1} = \frac{(253.5\ kg/s \cdot m^2\ )(0.019\ m)}{2.72 \times 10^{-5} kg/m \cdot s}$$

$$Re_s = 177{,}077.2 \approx 177{,}077$$

After we get all these values we will calculate the wall temperature (T_w)

$$T_w = \frac{1}{2}\left[\left(\frac{T_{c1} + T_{c2}}{2}\right) + \left(\frac{T_{h1} + T_{h2}}{2}\right)\right]$$

$$T_w = \frac{1}{2}\left[\left(\frac{25℃ + 85℃}{2}\right) + \left(\frac{250℃ + 227.8℃}{2}\right)\right]$$

$$T_w = 146.95℃ \approx 147℃$$

$T_{c1} = 25℃$ $T_{h1} = 250℃$ $T_{c2} = 85℃$ $T_{h2} = 227.8℃$

4.1.1.9. Actual shell side heat transfer calculation

The approximate wall temperature viscosity of 147°C will be;

$\mu_w = 2.36 \times 10^{-5}\ kg/m \cdot s$

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{D_e G_s}{\mu_1}\right)^{0.55} \left(\frac{c_p \mu_1}{k}\right)^{1/3} \left(\frac{\mu_1}{\mu_w}\right)^{0.14}$$

$$\frac{h_o D_e}{k} = 0.36 \left(\frac{(0.019)(253.5\,)}{2.78 \times 10^{-5}}\right)^{0.55} \left(\frac{(1033\,)(2.78 \times 10^{-5})}{0.0407}\right)^{1/3} \left(\frac{2.78 \times 10^{-5}}{2.36 \times 10^{-5}}\right)^{0.14}$$

$$\frac{h_o D_e}{k} = 249.5$$

Actual shell side heat transfer (h_o) is found by

$$h_o = \frac{249.5\,k}{D_e} = \frac{(249.5)\left(0.0407\ {}^{W}/_{m} \cdot K\right)}{0.019m} = 535\,W/m^2 \cdot K$$

4.1.1.10. Tube side heat transfer area and average velocity calculation

The next is the tube side heat transfer area (A_{tp}), we calculate by

$$A_{tp} = \frac{\pi d_i^2}{4} \cdot \frac{N_t}{2} = \frac{\pi(0.016^2)}{4} \cdot \frac{845}{2} = 0.085m^2$$

Average velocity inside tubes at a temperature of 85°C and $\rho = 0.7898\,kg/m^3$.

$$u_m = \frac{\dot{m}_t}{\rho_t A_{tp}}$$

$$u_m = \frac{27000kg/hr}{(0.7898\,kg/m^3)0.486m^2} \cdot \frac{1hr}{3600s} = 19.54\,m/s$$

4.1.1.11. Actual tube side heat transfer coefficient calculation

Reynolds number (Re) will be:

$$Re = \frac{\rho u_m d_i}{\mu_2} = \frac{(0.7898\,kg/m^3)(19.54\,m/s)(0.016m)}{1.907 \times 10^{-5} kg/m \cdot s} = 12{,}947.8$$

The airflow is turbulent because $Re > 10^4$ and the next is calculating the friction factor (f).

$$f = (1.58 \ln Re - 3.28)^{-2}$$

$$f = (1.58 \ln(12{,}947.8) - 3.28)^{-2}$$

$$f = 0.00733$$

Using the correlation between Gnielinski's

$$Nu_b = \frac{\left({}^{f}/_{2}\right)(Re - 1000)(Pr)}{1 + 12.7\left({}^{f}/_{2}\right)^{1/2}\left(Pr^{2/3} - 1\right)}$$

$$Nu_b = \frac{\left({}^{0.00733}/_{2}\right)(12{,}947.8 - 1000)(0.706)}{1 + 12.7\left({}^{0.00733}/_{2}\right)^{1/2}\left(0.706^{2/3} - 1\right)}$$

$$Nu_b = 36.77$$

Tube side heat transfer coefficient (h_i) (actual) will be calculated by:

$$h_i = \frac{Nu_b k}{d_i} = \frac{36.77(0.0272\ {}^{W}/_{m} \cdot K)}{0.016m} = 62.5\ W/m^2 \cdot K$$

4.1.1.12. Actual overall heat transfer coefficient calculation

After we find h_i, we will calculate the actual overall heat transfer coefficient

$$U_f = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o R_{fi}}{d_i} + \frac{d_o \ln\left({}^{d_o}/_{d_i}\right)}{2k} + R_{fi} + {}^{1}/_{h_o}}$$

$$U_f = \frac{1}{\frac{0.019}{(0.016)(62.5)} + \frac{(0.019)(0.0001)}{0.016} + \frac{(0.0019)\ln\left(\frac{0.019}{0.016}\right)}{2(60)} + 0.0001 + \frac{1}{535}}$$

$$U_f = 47.41\ W/m^2 \cdot K$$

After we find (U_f) we will continue to the overall heat transfer for clean surface (U_c) based on the outside tube area using this formula;

$$U_c = \frac{1}{\frac{d_o}{d_i h_i} + \frac{d_o \ln\left({}^{d_o}/_{d_i}\right)}{2k} + {}^{1}/_{h_o}}$$

$$U_c = \frac{1}{\frac{0.019}{(0.016)(62.5)} + \frac{(0.0019)\ln\left({}^{0.019}/_{0.016}\right)}{2(60)} + \frac{1}{535}}$$

$$U_c = 47.91\ W/m^2 \cdot K$$

4.1.1.13. Shell side pressure drop calculation

The next step is to determine the shell side pressure drop using *Equation 3.17*.

$$\Delta P_s = \frac{f G_s^2 (N_b + 1) \cdot D_s}{2\rho D_e \phi_s}$$

Before we calculate the shell side pressure drop (ΔP_s) we have to find friction factor (f) for the shell with *Equation 3.18* and the number of baffles in the shell (N_b) viscosity correction factor for shell side (ϕ_s) respectively.

$$f = exp(0.576 - 0.19 \ln Re_s) \; where \; 400 < Re \leq 10^6$$

$$f = exp(0.576 - 0.19 \ln(12{,}947.8))$$

$$f = 0.294331$$

Number of baffles in the shell (N_b) calculated by;

$$N_b = {}^{L}\!/_{B} - 1 = {}^{5}\!/_{0.5} - 1 = 9$$

Viscosity correction factor for shell side (ϕ_s) calculated by;

$$\phi_s = \left(\frac{\mu_1}{\mu_w}\right)^{0.14}$$

$$\phi_s = \left(\frac{2.72 \times 10^{-5} \; kg/m \cdot s}{2.36 \times 10^{-5} \; kg/m \cdot s}\right)^{0.14} = 1.02$$

As we get those results we replace the values and calculate shell side pressure drop (ΔP_s) as follows;

$$\Delta P_s = \frac{f G_s^2 (N_b + 1) \cdot D_s}{2\rho D_e \phi_s}$$

$$\Delta P_s = \frac{(0.294331)(253.5^2)(9 + 1) \cdot (0.80)}{2(0.7898)(0.19)(1.02)}$$

$$\Delta P_s = 494289 \; Pa$$

Since 4.94 bar $<$ 10 bar, the pressure drop of shell-side is acceptable. So we can carry on to the next stages.

$$\Delta P_s = 4.94 \; bar$$

4.1.1.14. Shell and tube heat exchanger length calculation

For the tube length;

$$Q = \left(\dot{m} c_p\right)_c (Tc_2 - Tc_1)$$

$$Q = \left(\left(\frac{27{,}000}{3600}\right)\left(1005\,\frac{\mathrm{J}}{\mathrm{kg}}\cdot\mathrm{K}\right)\right)_c (85°\mathrm{C} - 25°\mathrm{C})$$

$$Q = 452{,}250\ W \approx\ 452.25\ kW$$

$$\Delta T_{lm,cf} = \frac{(\Delta T_1 - \Delta T_2)}{\ln\frac{\Delta T_1}{\Delta T_2}} = \frac{(22.2 - 60)}{\ln\frac{22.2}{60}} = 38\ \mathrm{K}$$

A correction factor (F) using the value from the chart we calculate the true mean temperature difference (ΔT_m), According to the Fig F=0.99

$$\Delta T_m = F\Delta T_{lm,cf}$$

$$\Delta T_m = (0.99)(38) = 37.62 \approx 38K$$

Heat transfer area based on the outside with fouling (A_{of}) will be calculated by

$$A_{of} = \frac{Q}{U_f \Delta T_m}$$

$$A_{of} = \frac{452.25\ kW}{(47.41\ W/m^2\cdot K)(38\ K)} = 251\ m^2$$

To get the length (L) of the heat exchanger use heat transfer area (A_o) *Equation 3.6.*

$$A_{of} = \pi d_o N_t L$$

$$L = \frac{A_{of}}{\pi d_o N_t}$$

$$L = \frac{251\ m^2}{\pi(0.019\ m)(845)}$$

$L = 4.97\ m$ And by rounding off it will be 5 m

Therefore the maximum length is 5 m and according to our design of the heat exchanger, it is 5 m. This will give the result of acceptability.

4.1.1.15. Total tube side pressure drop calculation

And finally, we will calculate the total tube side pressure drop (ΔP_{total}) using *Equation 3.22*

$$\Delta P_{total} = \left(4f\,\frac{LN_p}{d_i} + 4N_p\right)\frac{\rho u_m^2}{2}$$

$$\Delta P_{total} = \left(4(0.00733)\frac{(5 \times 1)}{0.016} + 4(1) \right) \frac{(1.135\ )(19.54)^2}{2}$$

$$\Delta P_{total} = 2852.1\ pa \approx 4.14\ Psi$$

4.2. Coal elemental value calculations

4.2.1. Amount of saved coal

$$(\dot{m})_{coal} = \frac{Q}{\eta \cdot C_v}$$

The basic faction of a coal-powered power plant is to convert the coal energy to electrical energy. Therefore, the first thing we have to know is how much coal has energy, the contents of this energy in coal is given in terms of kilojoules per kilogram or kilocalorie per kilogram of coal as the gross calorific value (GCV). Depending on the quality and types of coal, the value will vary from 2508 to 5971 kcal/kg.

Converting $452.25\ kW$ to MW

1000KW =1MW

$452.25\ kW$ = Q

$$Q = \frac{452.25\ kW \times 1mW}{1000kW} = 0.45225\ mW$$

$$(\dot{m})_{coal} = \frac{0.45225\ mW}{0.4 \times 2900kcal/kg} \times \frac{1kcal}{1.163 \times 10^{-3}kw/hr} \times \frac{1000\ kw}{1\ mw}$$

$$(\dot{m})_{coal} = 335.23\ kg\ coal/hr \times \frac{1hr}{3600\ s}$$

$$(\dot{m})_{coal} = 0.093\ kg\ coal/s$$

4.2.2. Elemental analysis of coal

The coal we use in every power plant has some elements with different values

CONSTITUENTS	W%
C	51.12
H	3.89
O	14.65
N	0.61
S	1.87
ASH	14.36
MOISTURE	13.5

4.2.3. Required amount of O_2 in carbon (C)

If 100 kg of coal includes 51.12 kg of carbon (C)

0.093 Kg coal includes = W_1

Atomic masses of coal elements;

- N_2=28
- O_2=32
- C=12

$$W_1 = \frac{0.093kg \times 51.12}{100\frac{kg}{s}c}$$

$$W_1 = 0.049$$

12 kg C needs 32 kg of O_2

0.049 Kg needs = W_2

$$W_2 = \frac{\frac{0.049kg}{s}C \times 32\ kg}{12\ kg}$$

$$W_2 = \frac{0.13kg}{s}O_2$$

4.2.4. Required amount of O_2 in hydrogen (H)

If 100 kg of coal includes 3.89 kg of hydrogen (H)

0.13 Kg/s coal includes = W_3

$$W_3 = \frac{\frac{0.13\ kg}{s}C \times 3.89\ kg}{100\ kg}$$

$$W_3 = \frac{0.00505\ kg}{s}H$$

$H_2 \quad + \quad {}^{1}/_{2}\,O_2 \rightarrow H_2O$

$2kg \qquad 16kg$

If 2 kg of H needs 16kg O_2

0.00505 Kg/s H needs = W_4

$$W_4 = \frac{\frac{0.00505\ kg}{s}H \times 16\ kg}{2\ kg}$$

$$W_4 = \frac{0.04\ kg}{s}O_2$$

4.2.5. Required amount of O_2 in nitrogen (N)

Calculating for nitrogen is not possible, because it depends on the temperature of the combustion system.

4.2.6. Required amount of O_2 in sulfur (S)

For sulfur, if 100 kg of coal includes 1.87 kg

0.049 kg includes $= W_5$

$$W_5 = \frac{\frac{0.049kg}{s} \times 1.87\ kg}{100\ kg} = \frac{0.000916\ kg}{s}\ O_2$$

$S \quad + \quad O_2 \rightarrow \quad SO_2$

$32kg \quad 32kg$

32 kg S needed 32 kg of O_2

0.04 kg/s needed $= W_6$

$$W_6 = \frac{\frac{0.04\ kg}{s} \times 32\ kg}{32\ kg}$$

$$W_6 = \frac{0.04\ kg}{s} O_2$$

4.2.7. Total oxygen (O_2) amount

Total (pure oxygen) $O_2 = W_1 + W_4 + W_6 - W_5$

Mass of elements in the air

- $O_2 = 20.95\%$
- $N = 78.01\%$
- $Ar = 0.93\%$

$$O_2 = 0.049\ kg + 0.04\ kg + 0.04\ kg - 0.000916\ kg$$

$$O_2 = 0.1281\ kg/s$$

4.2.8. Molecular weight of air

$AIR = 79\ V\%\ N_2 + 21\ V\%\ O_2$

28 kg/kmol 32kg/kmol

$$M_{air} = molecular\ weigth\ of\ air$$

$$M_{air} = 0.79 \times 28 + 0.21 \times 32$$

$$M_{air} = 28.84\, kg/kmol$$

4.2.9. O2 percentage and mass in the air

Percentage of O_2 in the air (W%) will be

$$W\%\ O_2 = \frac{0.21 \times 32}{28.84} \times 100$$

$$W\%\ O_2 = 23.3$$

If 100 kg of air includes 23.3 kg O_2

W_7 = 0.1281 kg/s O_2

$$W_7 = \frac{\frac{0.1281\ kg}{O_2} \times 100\ kg}{23.3\ kg}$$

$$W_7 = \frac{0.55\ kg}{s} air$$

4.2.10. Required amount of air

The amount of air (m^3) needed in 1 second (1s)

First, we have to know the air density (using ideal gas law)

Ideal gas law

$$\rho V = nRT \qquad R = 0.08206\ \frac{L \cdot atm}{mol \cdot K}$$

$$1\ atm \times 1m^3 = \frac{\rho_{air} 1m^3}{28.84\, kg/kmol} \times \frac{0.08206 m^3 \cdot atm}{kmol \cdot K} \times 298\ K$$

$$1\ atm \times 1m^3 = 0.848$$

$$\rho_{air} = 1.18 kg/s$$

If 1.18 kg air = $1m^3$

0.321kg air = W_8

$$W_8 = \frac{0.55 \text{ kg air } \times 1m^3}{1.18 \text{ kg air}}$$

$$W_8 = 0.466\, {}^{m^3}/_{s}\, of\ air$$

W_8 - is the theoretical air feed in to the combustion. The actual may exceed this results.

In practice, every mole of O_2 will not give a reaction in a given combustion system due to some factors such as retention time, temperature, mixing, and so on. Therefore we feed more air to the combustion system than the theoretical one.

4.2.11. Actual air feed amount

To get the amount of actual air feed amount (Q_{act})

$$Q_{act} = Q_{th} + Q_{th}\left(\frac{x_2 - x_1}{x_2}\right)$$

- $Q_{th} = W_8$
- x_1 *and* x_2 *the amount used in percent*

Let's assume that we used 50% of excess air as actual value combustion air,

$$Q_{act} = 0.466\,{m^3}/_{s}\,of\ air + 0.466\,{m^3}/_{s}\,of\ air\left(\frac{100 - 50}{100}\right)$$

$$Q_{act} = 0.7\,{m^3}/_{s}$$

4.2.12. Actual mass of air feed

Actual mass ($\dot{m}_{act}$) of air

$$\dot{m}_{act} = Q_{act} \times \rho_{air}$$

$$\dot{m}_{act} = 0.7\,{m^3}/_{s} \times 1.18 kg/m^3$$

$$\dot{m}_{act} = 0.826\frac{kg}{s}air$$

4.3. Comparative calculation with different values

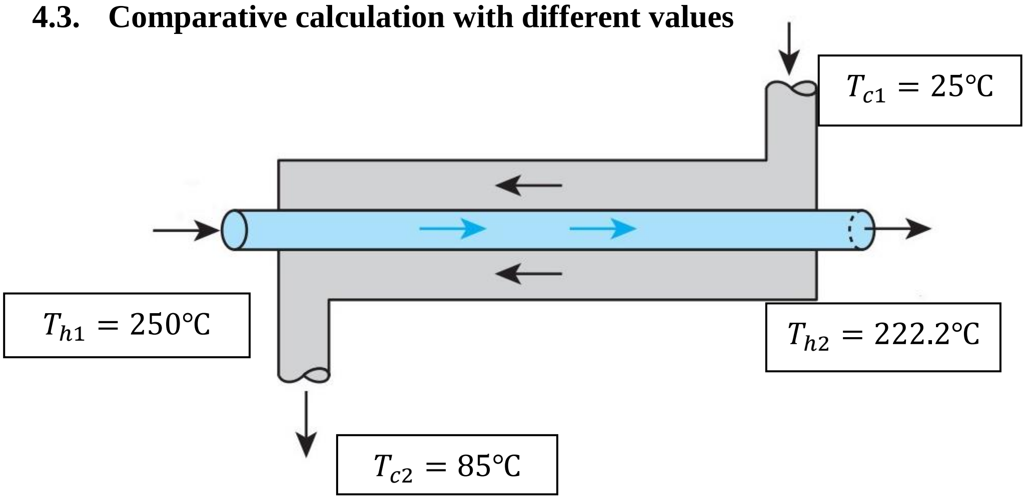

Figure 4.1. Schematic drawing of shell and tube heat exchanger.

- $\dot{m}_{air} = 0.826\ kg/s$
- $C_p at\ 85°C = 1010\, J/Kg \cdot K$ $\quad$ $C_p at\ 95°C = 1011\, J/Kg \cdot K$
- $C_p at\ 105°C = 1012\, J/Kg \cdot K$ $\quad$ $C_p at\ 115\ °C = 1013\, J/Kg \cdot K$

4.3.1. Heat gained in heat exchanger with different temperature value

$T_{c2} = 85°C + 273.15 = 358.15K$ (Energy saved up when we heat up combustion air from 25 to 85).

$Q_1 = m_{air} \cdot (C_p) \cdot \Delta T_1$

$Q_1 = 0.826\ kg/s \cdot (1010\ J/Kg \cdot K) \cdot (85°C - 25°C)$

$Q_1 = 0.826\ kg/s \cdot (1010\ J/Kg \cdot K) \cdot (60K)$

$Q_1 = 50{,}105\ W \approx 50.105\ kW$

Heating up combustion air from 25°C to 85°C will save $50.105\ kW\ of$ heat energy

If we don’t use waste heat, this energy will be directly used from coal.

$T_{c2} = 95°C + 273.15 = 368.15K$ (Energy saved up when we heat up combustion air from 25°C to 95°C).

$Q_2 = m_{air} \cdot (C_p) \cdot \Delta T_2$

$Q_2 = 0.826\ kg/s \cdot (1011\ J/Kg \cdot K) \cdot (95°C - 25°C)$

$Q_2 = 0.826\ kg/s \cdot (1011\ J/Kg \cdot K) \cdot (70K)$

$Q_2 = 58{,}456\ W \approx 58.456\ kW$

Heating up combustion air from 25°C to 95°C will save $58.456\ kW\ of$ heat energy.

$T_{c2} = 105°C + 273.15 = 378.15K$ (Energy saved up when we heat up combustion air from 25°C to 105°C)

$Q_3 = m_{air} \cdot (C_p) \cdot \Delta T_3$

$Q_3 = 0.826\ kg/s \cdot (1012\ J/Kg \cdot K) \cdot (105°C - 25°C)$

$Q_3 = 0.826\ kg/s \cdot (1012\ J/Kg \cdot K) \cdot (80K)$

$Q_3 = 66{,}873\ W \approx 66.873\ kW$

Heating up combustion air from 25°C to 105°C will save $66.873\ kW\ of$ heat energy

$T_{c2} = 115°C + 273.15 = 388.15K$ (Energy saved up when we heat up combustion air from 25°C to 115°C).

$Q_4 = m_{air} \cdot (C_p) \cdot \Delta T_4$

$Q_4 = 0.826\ kg/s \cdot (1013\ J/Kg \cdot K) \cdot (115°C - 25°C)$

$Q_4 = 0.826 \;\; kg/s \cdot (1013 \;\; J/Kg \cdot K) \cdot (90K)$

$Q_4 = 75{,}306\, W \approx 75.306\, kW$

Heating up combustion air from 25°C to 115°C will save 75.306 $kW\ of$ heat energy.

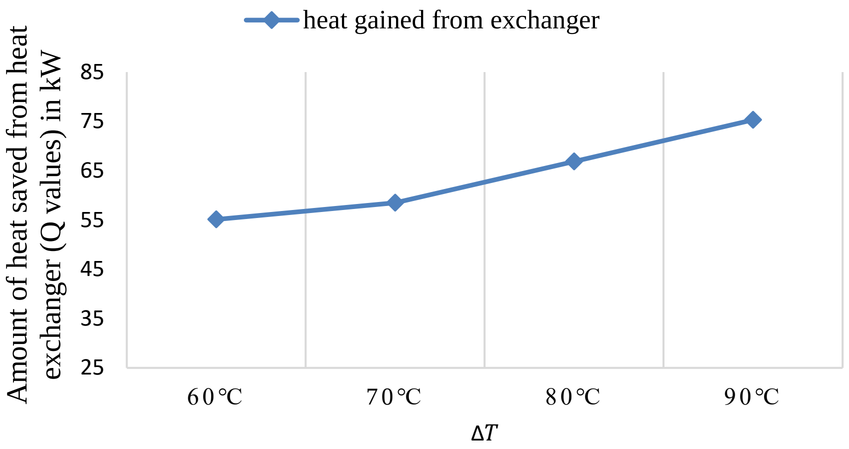

Figure 4.2. The gained heat in a heat exchanger at different temperatures

An increase in Q value depends on the exit temperature of fresh air (T_{c2}). Additionally the exit temperature T_{h2} will decrease. This will lead us to more heat gaining from waste.

4.3.2. Saving of coal ($\dot{m}_{SC}$) consumption

$$\dot{m}_{SC} = \frac{Q}{0.4 \times 2900 \;\; kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

At $\Delta T =$ 60°C (the amount of coal saved up in 1 second when we heat up combustion air from 25°C to 85°C).

$$\dot{m}_{SC1} = \frac{Q_1}{0.4 \times 2900 \;\; kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC1} = \frac{50.105\ kW}{0.4 \times 2900 \;\; kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$\dot{m}_{SC1} = 10.32 \times 10^{-3}\ kg/s$

At $\Delta T =$ 70°C (the amount of coal saved up in 1 second when we heat up combustion air from 25°C to 95°C).

$$\dot{m}_{SC2} = \frac{Q_2}{0.4 \times 2900 \;\; kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC2} = \frac{58.456\ kW}{0.4 \times 2900\ \ kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC2} = 12.04 \times 10^{-3}\ kg/s$$

At $\Delta T =\ 80$°C (the amount of coal saved up in 1 second when we heat up combustion air from 25°C to 105°C).

$$\dot{m}_{SC3} = \frac{Q_3}{0.4 \times 2900\ \ kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC3} = \frac{66.873\ kW}{0.4 \times 2900\ \ kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC3} = 13.78 \times 10^{-3}\ kg/s$$

At $\Delta T =\ 90$°C (the amount of coal saved up in 1 second when we heat up combustion air from 25°C to 115°C).

$$\dot{m}_{SC4} = \frac{Q_4}{0.4 \times 2900\ \ kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC4} = \frac{75.306\ kW}{0.4 \times 2900\ \ kcal/kg} \times \frac{1\ kcal}{1.163 \times 10^{-3} kWh} \times \frac{1\ hr}{3600\ s}$$

$$\dot{m}_{SC4} = 15.51 \times 10^{-3}\ kg/s$$

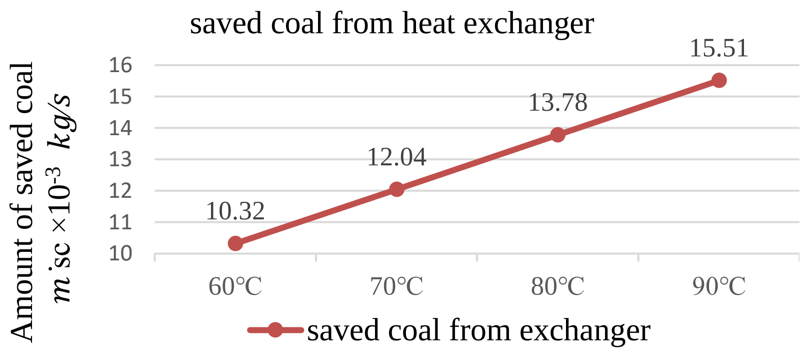

Figure 4.3. The amount of saved coal by heating up the combustion air

As we rise T_{c2} of Saved coal for $\dot{m}_{SC}$ the amount of saved coal will increase. The increase in saved coal will lead us to lower our coal consumption.

4.3.3. The percentage of saved coal

For $\dot{m}_{SC1}$ at $\Delta T =\ 60$°C (the percentage of saved coal in 1 second when we heat up combustion air from 25°C to 85°C).

$$\%\dot{m}_{SC1} = \frac{\dot{m}_{SC1}}{\dot{m}_{air}} \times 100\%$$

$$\%\dot{m}_{SC1} = \frac{10.32 \times 10^{-3}\ kg/s}{0.826\ \ kg/m^3} \times 100\%$$

$$\%\dot{m}_{SC1} = 1.25\%$$

For $\dot{m}_{SC2}$ at $\Delta T = 70$°C (the percentage of saved coal in 1 second when we heat up combustion air from 25°C to 95°C).

$$\%\dot{m}_{SC2} = \frac{\dot{m}_{SC2}}{\dot{m}_{air}} \times 100\%$$

$$\%\dot{m}_{SC2} = \frac{12.04 \times 10^{-3}\ kg/s}{0.826\ \ kg/s} \times 100\%$$

$$\%\dot{m}_{SC2} = 1.46\%$$

For $\dot{m}_{SC3}$ at $\Delta T = 80$°C (the percentage of saved coal in 1 second when we heat up combustion air from 25°C to 105°C).

$$\%\dot{m}_{SC3} = \frac{\dot{m}_{SC3}}{\dot{m}_{air}} \times 100\%$$

$$\%\dot{m}_{SC3} = \frac{13.78 \times 10^{-3}\ kg/s}{0.826\ \ kg/s} \times 100\%$$

$$\%\dot{m}_{SC3} = 1.67\%$$

For $\dot{m}_{SC4}$ at $\Delta T = 90$°C (the percentage of saved coal in 1 second when we heat up combustion air from 25°C to 115°C).

$$\%\dot{m}_{SC4} = \frac{\dot{m}_{SC4}}{\dot{m}_{air}} \times 100\%$$

$$\%\dot{m}_{SC4} = \frac{15.51 \times 10^{-3}\ kg/s}{0.826\ \ kg/s} \times 100\%$$

$$\%\dot{m}_{SC4} = 1.87\%$$

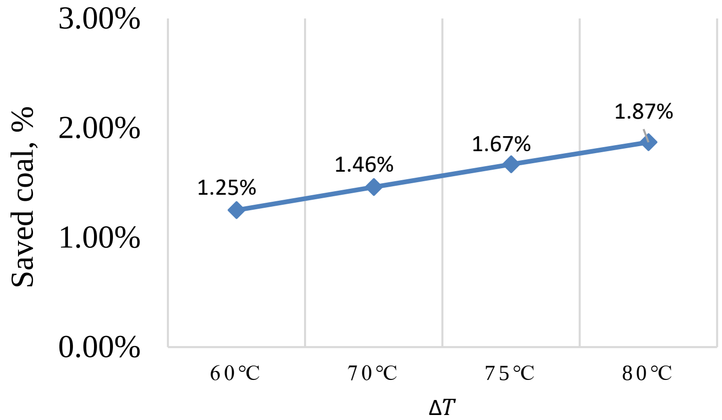

Figure 4.4. Effect of combustion temperature on coal saving

4.3.4. Reduction percentage in O2

As we raise the temperature we will reduce the carbon emissions that was leaked to the environment. (The reduction percentage in CO_2 when we heat up combustion air from 25°C to 85°C).

$$C + O_2 \rightarrow CO_2 \text{ at } T_{c2} = 85°\mathrm{C}$$

$C + O_2 \rightarrow CO_2$

$12kg \qquad 44kg$

Combustion of 12 kg C produces 44 kg CO_2

$10.32 \times 10^{-3}\, kg\ C/s$ Creates X_1 kg CO_2/s

$$X_1 = \frac{10.32 \times 10^{-3}\, kg/s \times 44\, kg}{12\, kg}$$

$$X_1 = 0.0378\, kg/s\ \ of\ C$$

$$\%reduction\ in\ CO_2 = \frac{X_1}{\dot{m}_{air} \cdot W\%} \times 100\%$$

$$\%reduction\ in\ CO_2 = \frac{0.0378\, kg/s}{0.826\ \ kg/s\, (0.5112)} \times 100\% = 8.96\%$$

$C + O_2 \rightarrow CO_2$ At $T_{c2} = 95°\mathrm{C}$ (the reduction percentage in CO_2 when we heat up combustion air from 25°C to 95°C).

$C + O_2 \rightarrow CO_2$

$12kg \qquad 44kg$

Combustıon of 12 kg C produces 44 kg CO_2

$6.834 \times 10^{-3}\, kg\ C/s$ Creates X_2 kg CO_2/s

$$X_2 = \frac{12.04 \times 10^{-3}\, kg/s \times 44\, kg}{12\ kg}$$

$$X_2 = 0.0441\, kg/s\ \ of\ C$$

$$\%reduction\ in\ CO_2 = \frac{X_2}{\dot{m}_{air} \cdot W\%} \times 100\%$$

$$\%reduction\ in\ CO_2 = \frac{0.0441\, kg/s}{0.826\ \ kg/m^3 \cdot 0.5112} \times 100\% = 10.43\%$$

$C + O_2 \rightarrow CO_2$ At $T_{c2} =\ $ 105°C (the reduction percentage in CO_2 when we heat up combustion air from 25°C to 105°C).

$C + O_2 \rightarrow CO_2$

$12kg \qquad 44kg$

Combustıon of 12 kg C produces 44 kg CO_2

$7.818 \times 10^{-3}\, kg\ C/s$ Creates X_3 kg CO_2/s

$$X_3 = \frac{13.78 \times 10^{-3}\, kg/s \times 44\, kg}{12\ kg}$$

$$X_3 = 0.0505\, kg/s\ \ of\ C$$

$$\%reduction\ in\ CO_2 = \frac{X_3}{\dot{m}_{air} \cdot W\%} \times 100\%$$

$$\%reduction\ in\ CO_2 = \frac{0.0505\, kg/s}{0.826\ \ kg/s \cdot 0.5112} \times 100\%$$

$$= 11.96\ \%$$

$C + O_2 \rightarrow CO_2$ At $T_{c2} =\ $ 115°C (the reduction percentage in CO_2 when we heat up combustion air from 25°C to 115°C).

$C + O_2 \rightarrow CO_2$

$12kg \qquad 44kg$

Combustıon of 12 kg C produces 44 kg CO_2

$8.804 \times 10^{-3}\, kg\ C/s$ Creates X_4 kg CO_2/s

$$X_4 = \frac{15.51 \times 10^{-3}\, kg/s \times 44\, kg}{12\, kg}$$

$$X_4 = 0.0569\, kg/s \;\; of \;\; C$$

$$\% reduction\ in\ CO_2 = \frac{X_4}{\dot{m}_{air} \cdot W\%} \times 100\%$$

$$\% reduction\ in\ CO_2 = \frac{0.0569\, kg/s}{0.826\;\, kg/s \cdot 0.5112} \times 100\% = 13.47\ \%$$

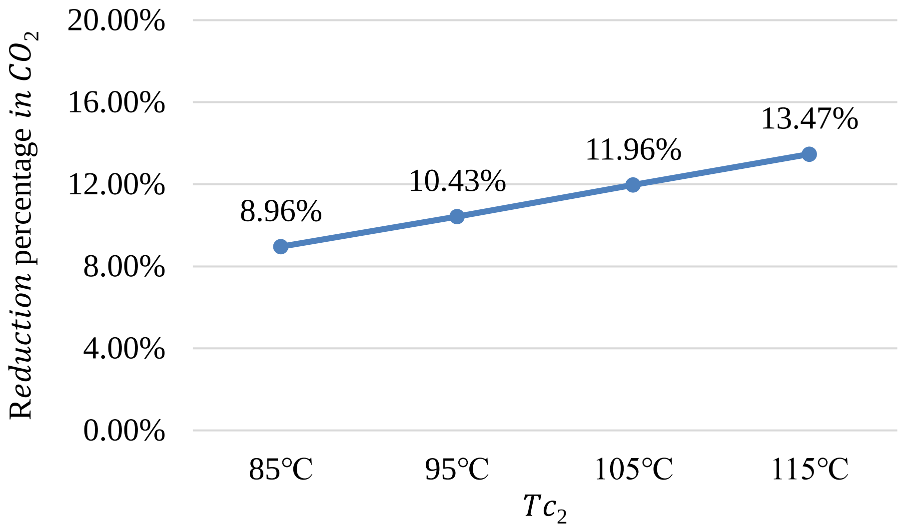

Figure 4.5. Reduction percentage of CO_2 in the power plant

4.3.5. Heat transfer surface area with different values

The heat transfer surface area of heat exchanger to heat up combustion air will be calculated by *Equation 3.1*:

$$A_o = \frac{Q}{U_c \Delta T_m}$$

For $\Delta T_1 = 60℃ = 60℃ + 273.15K = 333.15K$ (heat up combustion air from 25℃ to 85℃).

$$A_{o1} = \frac{Q_1}{U_c \Delta T_1}$$

$$A_{o1} = \frac{50{,}105\ W}{(53\;\, W/m^2 \cdot K)(333.15K)}$$

$$A_{o1} = 2.84\ m^2$$

For $\Delta T_2 = 70°C = 70°C + 273.15K = 343.15K$ (heat up combustion air from 25°C to 95°C).

$$A_{o2} = \frac{Q_2}{U_c \Delta T_2}$$

$$A_{o2} = \frac{58{,}456\ W}{(53\ \ W/m^2 \cdot K)(343.15K)}$$

$$A_{o2} = 3.21\ m^2$$

For $\Delta T_3 = 80°C = 80°C + 273.15K = 353.15K$(heat up combustion air from 25°C to 105°C).

$$A_{o3} = \frac{Q_3}{U_c \Delta T_3}$$

$$A_{o3} = \frac{66{,}873\ W}{(53\ \ W/m^2 \cdot K)(353.15K)}$$

$$A_{o3} = 3.57\ m^2$$

For $\Delta T_4 = 90°C = 90°C + 273.15K = 363.15K$ (heat up combustion air from 25°C to 115°C).

$$A_{o4} = \frac{Q_4}{U_c \Delta T_4}$$

$$A_{o4} = \frac{75{,}306\ W}{(53\ \ W/m^2 \cdot K)(363.15K)} = 3.91\ m^2$$

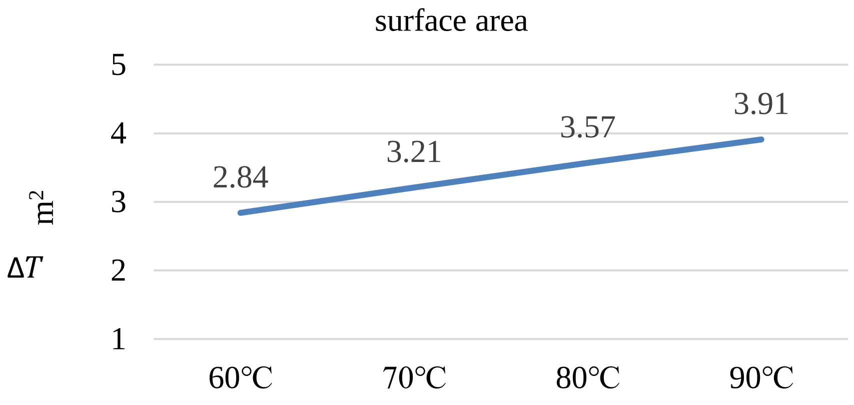

Figure 4.6. Effects of surface area on coal power plant

Table 4.1.Result of design analysis

Area of heat transfer (A_{of})	251 m^2
Heat transfer (Q)	452.25 kW
Tube length (L)	5 m
Tube numbers (N_t)	845
Shell Diameter (D_s)	0.8 m
Baffle spacing (B)	0.5 m
Number of baffles (N_b)	10
Bundle crossflow area(A_s)	0.08
Tube arrangement =	1- P
Tube material (Carbon steel)	k=60$W/m \cdot K$
Shell side mass velocity (G_s)	253.5 kg/s·m
Average velocity inside tubes (u_m) at 85℃, ρ =0.9878 kg/m^3	19.54 m/s
Friction factor	0.00733
Actual shell side heat transfer (h_o)	535 $W/m^2 \cdot K$
Actual tube side heat transfer (h_i)	62.5 $W/m^2 \cdot K$
Actual overall heat transfer coefficient with fouling (u_f)	47.41$W/m^2 \cdot K$
Actual overall heat transfer coefficient clean (u_c)	47.91$W/m^2 \cdot K$
Shell side pressure drop (ΔP_s)	4.94 bar
Total pressure drop (ΔP_{total})	4.14 Psi
Saved coal ($\dot{m}$)	0.093 kg/s
Actual air feed amount (Q_{act}) with 50 % usage of air	0.7 m^3/s
The actual mass of air needed ($\dot{m}_{act}$)	0.826 kg/m^3

Table 4.2. Results of the evaluation

Using shell and tube heat exchanger on the waste heat from the power plant can give us this result				
Temperature, ºC	85℃	95℃	105℃	115℃
Amount of gained heat (KW)	50.105	58.456	66.873	75.306
Amount of saved coal ($\times 10^{-3}$ kg/s)	10.32	12.04	13.78	15.51
Percentage of saved coal (%)	1.25	1.46	1.67	1.87
Saved amount of CO_2 per second (kg)	0.0378	0.0441	0.0505	0.0569
Saved amount of CO_2 per year (ton)	1192.1	1391	1593	1,794.4
% reduction in CO_2	8.96	10.43	11.96	13.47

Our evaluation using shell and tube heat exchanger in power plants will show how much waste heat will save coal consumption, reduce CO_2, and other GHG emissions.

5. CONCLUSIONS

This paper shows that significant fuel cost savings, reduction of CO_2 and GHG emissions can be achieved by adding a waste heat recovery system to coal power plants and other industries. Depending on the selected waste heat recovery system, quality of the waste heat, quantity, and operational profile the reduction of fuel between 1-11% is possible. Larger coal power plant waste heat will leads to larger fuel savings. Recovery of waste heat system gives large CO_2, NO_x, SO_x, and other particulate reductions to the benefit of environmental protection in addition to large fuel savings. All these benefits are almost impossible to imagine a coal power plant without a heat exchange system, as it plays a vital role in energy transfer from waste heat.

A heat exchanger is a device that offers heat from a fluid (air) to transfer to a second fluid (another air source) whereas the two fluids not having to mix along or obtain direct contact or a device that transfers heat between different two process fluids (air) without this two fluids mixing across the process. Heat exchangers have a lot of applications in coal power plants, automotive industries, nuclear reactors, air conditioning systems, and other waste heat producing industries. Today, heat exchangers have various types and which are available and used in coal power plants and industries. However, the most widely used heat exchanger is the shell and tube heat exchanger. Because of their advantages, this shell and tube heat exchanger has importance in coal power plants and other industries.

6. RECOMMENDATIONS

Coal power plants mainly consume coal to generate process heat and this leads to the production of CO_2, O_2, NO_x, $SO_{x,}$ greenhouse gases (GHG) emissions, and other particulate matters. Additionally, they consume more fuel costs to generate process heat. Installing waste heat to coal power plants in an economical way will minimize the costs and carbon footprint.

I recommend that power plants pay attention to global profits rather than individual profit:

- Focus on overall efficiency at the plant boundaries and reduce some energy that lives the plant that will still be wasted.
- Identify waste heat recovery and reuse opportunities (energy balance of energy sources and uses) in the power plants.
- Using a counter flow system to get the highest efficiency (to get the maximum result)
- While designing and selecting a waste heat exchanger system, real load instead of maximum or theoretical capacity should be used.
- While manufacturing using standard components and parts that are already on the market to get the highest efficient device.

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APPENDIXES

Table 1. Heat transfer coefficients of shell and tube heat exchanger

Substance used for heat transfer	Fluid condition	W $(m^2 \cdot k)$
Water	Liquid	5000-7000
Organics (light)	Liquid	1500-200
Organics (medium)	Liquid	750-1500
Organics (heavy)	Liquid	
	Heating	250-750
	Cooling	150-400
Very heavy organics	Liquid	
	Heating	100-300
	Cooling	60-150
Gas	1-2 bar abs	80-125
Gas	10 bar abs	250-400
Gas	100 bar abs	500-800

Table 2. Conductivity of materials

Conductivity of materials	
Metal	$k = W/m \cdot K$
Aluminum	236
Brass	150
Chrome	95
Carbon steel	60
Copper	401
Cast Iron	52
Nickel	91
Stainless steel	15
Steel	80
Titanium	21

Table 3. Typical overall coefficient of shell and tube heat exchanger

The typical overall coefficient of shell and tube heat exchanger		
Hot fluid	Cold fluid	$U(W/m^2 \cdot °C)$
Heat exchangers		
Water	Water	800-1500
Organic solvents	Organic solvents	100-300
Light oils	Light oils	100-400
Heavy oils	Heavy oils	50-300
Gases	Gases	10-50
Coolers		
Organic solvents	water	250-750
Light oils	Water	350-900
Heavy oils	Water	60-300
Gases	Water	20-300
Organic solvents	Brine	150-500
Water	Brine	600-1200
Gases	Brine	15-250
Heaters		
Steam	Water	1500-4000
Steam	Organic solvents	500-1000
Steam	Gases	30-300
Flue gases	steam	30-100
Flue	Hydrocarbon vapors	30-100

Table 3. (continued)		
Condensers		
Aqueous vapors	Water	1000-1500
Organic vapors	Water	700-1000
Organics	Water	500-700
Vaporizers		
Steam	Aqueous solutions	1000-1500
Steam	Light organics	900-1200
Steam	Heavy organics	600-900

Table 4. Air properties (Density, gas constant, specific heat, viscosity at 20 ^{O}C)

Gas name	ρ [kg/m^3]	R [J/kg*K]	C_p [J/kg*K]	C_v [J/kg*K]	k=Cp/Cv [J/kg*K]
Air	1.293	287.0	1010.0	720.0	1.4

Table 5. Air properties at 1 bar (Density, viscosity, and specific heat)

T [^{O}C]	C_p [kJ/kgK]	ρ [kg/m^3]	μ*10^6 [Pa*s]
-50.0	1.007	1.563	14.65
0.0	1.006	1.275	17.2
50.0	1.008	1.078	19.61
100.0	1.012	0.932	21.82
150.0	1.018	0.8226	23.92
200.0	1.026	0.7356	25.85
400.0	1.069	0.517	32.76

Table 6. Flue gases properties (Density, specific heat, viscosity)

Temperature [^{O}C]	C_p [kJ/kgK]	ρ [kg/m^3]	μ*10^6 [Pa*s]	ν*10^6 [m^2/s]
0.0	1.042	1.295	15.8	12.2
100.0	1.068	0.95	20.4	21.54
200.0	1.097	0.748	24.5	32.8
300.0	1.122	0.617	28.2	45.81
400.0	1.151	0.525	31.7	60.38
500.0	1.185	0.457	34.8	76.3
600.0	1.214	0.405	37.9	93.61
800.0	1.264	0.33	43.4	131.8
1000.0	1.306	0.275	48.4	174.3
1200.0	1.34	0.24	53.0	221.0

Table 7. Dimensional data for commercial tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.^{2})	Sq. Ft. External Surface Per Ft. length	Sq. Ft. Internal Surface Per Ft. Length	Weight Per Ft. Length, Steel (Ib.)	ID Tubing (in,)	OD/ID
¼	22	0.028	0.0295	0.0655	0.0508	0.066	0.194	1.289
¼	24	0.022	0.0333	0.0655	0.0539	0.054	0.206	1.214
¼	26	0.018	0.0360	0.0655	0.0560	0.045	0.214	1.168
3/8	18	0.049	0.0603	0.0982	0.0725	0.171	0.277	1.354
3/8	20	0.035	0.0731	0.0982	0.0798	0.127	0.305	1.233
3/8	22	0.028	0.0799	0.0982	0.0835	0.104	0.319	1.176
3/8	24	0.022	0.0860	0.0982	0.0867	0.083	0.331	1.133
½	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	1.351
½	18	0.049	0.1269	0.1309	0.1052	0.236	0.402	1.244
½	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	1.163

Table 7. (continued)

½	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	1.126
5/8	12	0.109	0.1301	0.1636	0.1066	0.602	0.407	1.536
5/8	14	0.083	0.1655	0.1636	0.1202	0.479	0.459	1.362
5/8	15	0.072	0.1817	0.1636	0.1259	0.425	0.481	1.299
5/8	16	0.065	0.1924	0.1636	0.1296	0.388	0.49s	1.263
5/8	17	0.058	0.2035	0.1636	0.1333	0.350	0.509	1.228
5/8	18	0.049	0.2181	0.1636	0.1380	0.303	0.527	1.186
5/8	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	1.136
¾	10	0.134	0.1825	0.1963	0.1262	0.884	0.482	1.556
¾	11	0.120	0.2043	0.1963	0.1335	0.809	0.510	1.471
¾	12	0.109	0.2223	0.1963	0.1393	0.748	0.532	1.410
¾	13	0.095	0.2463	0.1963	0.1466	0.666	0.560	1.339
¾	14	0.083	0.2679	0.1963	0.1529	0.592	0.584	1.284
¾	15	0.072	0.2884	0.1963	0.1587	0.520	0.606	1.238
¾	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	1.210
¾	17	0.058	0.3157	0.1963	0.1660	0.428	0.634	1.183
¾	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	1.150
¾	20	0.035	0.3632	0.1963	0.1780	0.269	0.680	1.103
7/8	10	0.134	0.2892	0.2291	0.1589	1.061	0.607	1.441
7/8	11	0.120	0.3166	0.2291	0.1662	0.969	0.635	1.378
7/8	12	0.109	0.3390	0.2291	0.1720	0.891	0.657	1.332
7/8	14	0.083	0.3948	0.2291	0.1856	0.704	0.709	1.234
7/8	16	0.065	0.4359	0.2291	0.1950	0.561	0.745	1.174
7/8	18	0.049	0.4742	0.2291	0.2034	0.432	0.077	1.126
7/8	20	0.035	0.5090	0.2291	0.2107	0.313	0.805	1.087
1	8	0.165	0.3526	0.2618	0.1754	1.462	0.670	1.493
1	10	0.134	0.4208	0.2618	0.1916	1.237	0.670	1.493
1	11	0.120	0.4536	0.2618	0.1990	1.129	0.732	1.366
1	12	0.109	0.4803	0.2618	0.20447	1.037	0.782	1.279
1	13	0.095	0.5153	0.2618	0.2121	0.918	0.810	1.235
1	14	0.083	0.5463	0.2618	0.2183	0.813	0.834	1.199
1	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	1.167
1	16	0.065	0.5945	0.2618	0.2278	0.649	0.870	1.119
1	18	0.049	0.6390	0.2618	0.2361	0.496	0.902	1.109
1	20	0.035	0.6793	0.2618	0.2435	0.360	0.930	1.075
1-1/4	7	0.180	0.6221	0.3272	0.2330	2.057	0.890	1.404
1-1/4	8	0.165	0.6648	0.3272	0.2409	1.921	0.920	1.359
1-1/4	10	0.134	0.7574	0.3272	0.2571	1.598	0.982	1.273
1-1/4	11	0.120	0.8012	0.3272	0.2644	1.448	1.010	1.238
1-1/4	12	0.109	0.8365	0.3272	0.2702	1.329	1.032	1.21
1-1/4	12	0.095	0.8825	0.3272	0.2773	1.173	1.060	1.179
1-1/4	14	0.083	0.9229	0.3272	0.2838	1.033	1.084	1.153
1-1/4	16	0.065	0.9852	0.3272	0.2932	0.823	1.120	1.116
1-1/4	18	0.049	1.042	0.3272	0.3016	0.629	1.152	1.085
1-1/4	20	0.035	1.094	0.3272	0.3089	0.456	1.180	1.059
1-1/2	10	0.134	1.192	0.3927	0.3225	1.955	1.232	1.218
1-1/2	12	0.109	1.291	0.3927	0.3356	1.618	1.282	1.170
1-1/2	14	0.083	1.398	0.3927	0.3492	1.258	1.334	1.124
1-1/2	16	0.065	1.474	0.3927	0.3587	0.996	1.370	1.095
2	11	0.120	2.433	0.5236	0.4608	2.410	1.760	1.136
2	13	0.095	2.573	0.5236	0.4739	1.934	1.810	1.105
2-1/2	9	0.148	3.815	0.6540	0.5770	3.719	2.204	1.134

Table 8. Tube shell layouts (Tube counts)

Shell ID (in.)	1-P	2-P	4-P	6-P	8-P
¾-in. OD tubes on 1-in. triangular pitch					
8	*37*	*30*	*24*	*24*	
10	61	52	40	36	
12	92	82	76	74	70
13 ¼	109	106	86	82	74
15 ¼	151	138	122	118	110
17 ¼	203	196	178	172	166
19 ¼	262	250	226	216	210
21 ¼	316	302	278	272	260
23 ¼	384	376	352	342	328
25	470	452	422	394	382
27	559	534	488	474	464
29	630	604	556	538	508
31	745	728	678	666	640
33	856	830	774	760	732
35	970	938	882	864	848
1-in. OD tubes on 1 ¼-in. triangular pitch					
8	*21*	*16*	*16*	*14*	-
10	32	32	26	24	-
12	55	52	48	46	44
13 ¼	68	66	58	54	50
15 ¼	91	86	80	74	72
17 ¼	131	118	106	104	94
19 ¼	163	152	140	136	128
21 ¼	199	188	170	164	160
23 ¼	241	232	212	212	202
25	294	282	256	252	242
27	349	334	302	296	286
29	397	376	338	334	316
31	472	454	430	424	400
33	538	522	486	470	454
35	608	592	562	546	532
¾-in.OD tubes on 1-in. square pitch					
8	*32*	*26*	*20*	*20*	-
10	52	52	40	36	-
12	81	76	68	68	60
13 ¼	97	90	82	76	70
15 ¼	132	124	116	108	108
17 ¼	177	166	158	150	142
19 ¼	224	220	204	192	188
¾-in. OD tube on 1-in.square pitch					
21 ¼	*227*	*270*	*264*	*240*	*234*
23 ¼	341	324	308	292	292
25	413	394	370	346	346
27	481	460	432	408	408
29	553	526	480	456	456
31	657	640	600	560	560
33	746	718	688	648	648
35	845	824	780	748	748
37	934	914	886	838	838
39	1049	1024	982	948	948

1-in. OD tube on 1 ¼ in. square pitch

8	*21*	*16*	*14*	-	-
10	32	32	26	24	-
12	48	45	40	38	36
13 ¼	61	56	52	48	44
15 ¼	81	76	68	68	64
17 ¼	112	112	96	90	82
19 ¼	138	132	128	122	116
21 ¼	177	166	158	152	148
23 ¼	213	208	192	184	184
25	260	252	238	226	222
27	300	288	278	268	260
29	341	326	300	294	286
31	406	398	380	368	358
33	465	460	432	420	414
35	522	518	488	484	472
37	596	574	562	544	532

¾ -in. OD tube on 15/16-in. triangular pitch

8	*36*	*32*	*26*	*24*	*18*
10	62	56	47	42	36
12	109	98	86	82	78
13 ¼	127	114	96	90	86
15 ¼	170	160	140	136	128
17 ¼	239	224	194	188	178
19 ¼	301	282	252	244	234
21 ¼	361	342	314	306	290
23 ¼	442	420	386	378	364
25	532	506	468	446	434
27	637	602	550	536	524
29	721	692	640	620	594
31	847	822	766	722	720
33	974	938	878	852	826
35	1102	1068	1004	988	958

1 ¼-in. OD tubes on 1 9/16-in. square pitch

10	*16*	*12*	*10*	-	-
12	30	24	22	16	16
13 ¼	32	30	30	22	22
15 ¼	44	40	37	35	31
17 ¼	56	53	51	48	44
19 ¼	78	73	71	64	56
21 ¼	96	90	86	82	78
23 ¼	127	112	106	102	96
25	140	135	127	123	115
27	166	160	151	146	140
29	193	188	178	174	166
31	226	220	209	202	193
33	258	252	244	238	226
35	293	287	275	268	258
37	334	322	311	304	293
39	370	362	348	342	336

1 ½- in. OD tube on 1 7/8- in. square pitch

12	*16*	*16*	*12*	*12*	-
13 ¼	22	22	16	16	-
15 ¼	29	29	24	24	22
17 ¼	29	39	34	32	29
19 ¼	50	48	45	43	39
21 ¼	62	60	57	54	50
23 ¼	78	74	70	66	62
25	94	90	86	84	78
27	112	108	102	98	94
29	131	127	120	116	112
31	151	146	141	138	131
33	176	170	164	160	151
35	202	196	188	182	176

1 ½-in. tubes on 1 7/8- in. triangular pitch

12	*18*	*14*	*14*	*12*	*12*
13 ¼	27	22	18	16	14
15 ¼	26	34	32	30	27
17 ¼	48	44	42	38	36
19 ¼	61	58	55	51	48
21 ¼	76	78	70	66	61
23 ¼	95	91	86	80	76
25	115	110	105	98	95
27	136	131	125	118	115
29	160	154	147	141	136
31	184	177	172	165	160
33	215	206	200	190	184
35	246	238	230	220	215

1 ¼ - in. OD tube on 9/16-in. triangular pitch

10	-	-	-	-	-
10	20	18	14	-	-
12 ¼	32	30	26	22	20
13 ¼	38	36	32	28	26
15 ¼	54	51	45	42	38
17 ¼	69	66	62	58	54
19 ¼	95	91	86	78	69
21 ¼	117	112	105	101	95
23 ¼	140	136	130	123	117
25	170	164	155	150	140
27	202	196	185	179	170
29	235	228	217	212	202
31	275	270	255	245	235
33	315	305	297	288	275
35	357	348	335	327	315
37	407	390	380	374	357
39	449	436	425	419	407

Table 9. Heat Exchanger and Condenser Tube Data

A	B	C	D	E	F		G	
					Outside (ft.^{2}/ft)	*Inside (ft.^{2}/ft)*	*Metal Area (in.^{2})*	*Flow Area (in.^{2})*
		40	0.113	0.824	0.275	0.216	0.333	0.533
¾	1.05	80	0.154	0.742	0.275	0.194	0.434	0.432
		40	0.133	1.049	0.344	0.275	0.494	0.864
1	1.315	80	0.179	0.957	0.344	0.250	0.639	0.719
		40	0.140	1.38	0.434	0.361	0.668	1.496
1-1/4	1.660	80	0.191	1.278	0.434	0.334	0.881	1.283
		40	0.145	1.61	0.497	0.421	0.799	2.036
1-1/2	1.900	80	0.200	1.50	0.497	0.393	1.068	1.767
		40	0.154	2.067	0.622	0.541	1.074	3.356
2	2.375	80	0.218	1.939	0.622	0.508	1.477	2.953
		40	0.203	2.469	0.753	0.646	1.704	4.79
2-1/2	2.875	80	0.276	2.323	0.753	0.608	2.254	4.24
		40	0.216	3.068	0.916	0.803	2.228	7.30
3	3.5	80	0.300	2.900	0.916	0.759	3.106	6.60
		40	0.226	3.548	1.047	0.929	2.680	9.89
3-1/2	4.0	80	0.318	3.364	1.047	0.881	3.678	8.89
		40	0.237	4.026	1.178	1.054	3.17	12.73
4	4.5	80	0.337	3.826	1.178	1.002	4.41	11.50
		10 S	0.134	5.295	1.456	1.386	2.29	22.02
5	5.563	40	0.280	5.047	1.456	1.321	4.30	20.01
		80	0.337	4.813	1.456	1.260	6.11	18.19
		10 S	0.134	6.357	1.734	1.664	2.73	31.7
6	6.625	40	0.280	6.065	1.734	1.588	5.58	28.9
		80	0.432	5.761	1.734	1.508	8.40	26.1
		10 S	0.148	8.329	2.258	2.180	3.94	54.5
8	8.625	30	0.277	8.071	2.258	2.113	7.26	51.2
		80	0.500	7.625	2.258	1.996	12.76	45.7
		10 S	0.165	10.420	2.81	2.73	5.49	85.3
10	10.75	30	0.279	10.192	2.81	2.67	9.18	81.6
		Extra heavy	0.500	9.750	2.81	2.55	16.10	74.7
		10 S	0.180	12.390	3.34	3.24	7.11	120.6
	12.75	30	0.330	12.09	3.34	3.17	12.88	114.8
		Extra heavy	0.500	11.75	3.34	3.08	19.24	108.4
		10	0.250	13.5	3.67	3.53	10.80	143.1
14	14.0	Standard	0.375	13.25	3.67	3.47	16.05	137.9
		Extra heavy	0.500	13.00	3.67	3.40	21.21	132.7
		10	0.250	15.50	4.19	4.06	12.37	188.7
16	16.0	Standard	0.375	15.25	4.19	3.99	18.41	182.7
		Extra heavy	0.500	15.00	4.19	3.93	24.35	176.7
		10 S	0.188	17.624	4.71	4.61	10.52	243.9
18	18.0	Standard	0.375	17.25	4.71	4.51	20.76	233.7
		Extra heavy	0.500	17.00	4.71	4.45	27.49	227.0

(A-Nominal pipe size (in.), B-Outside diameter (in.), C- Schedule number or weight, D- Wall thickness (in.), E-Inside diameter (in.), F-Surface area and G- Cross-sectional area)

CURRICULUM VITAE

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