**EGE UNIVERSITY**
**GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES**

**(MASTER OF SCIENCE THESIS)**

# ENHANCEMENT OF THERMAL EFFICIENCY OF A NATURAL GAS FURNACE IN A PETROCHEMICAL PLANT

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**Selay SERT** tarafından **Yüksek Lisans** tezi olarak sunulan "**Enhancement of thermal efficiency of a natural gas furnace in a petrochemical plant**" başlıklı bu çalışma E.Ü. Lisansüstü Eğitim ve Öğretim Yönetmeliği ile E.Ü. Fen Bilimleri Enstitüsü Eğitim ve Öğretim Yönergesi'nin ilgili hükümleri uyarınca tarafımızdan değerlendirilerek savunmaya değer bulunmuş ve 22.08.2016 tarihinde yapılan tez savunma sınavında aday oybirliği/~~oyçokluğu~~ ile başarılı bulunmuştur.

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MEZAN

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22/08/2016

Selay Sert

# ÖZET

## BİR PETROKİMYA FABRİKASINDAKİ DOĞALGAZ FIRINININ ISIL VERİMLİLİĞİNİN İYİLEŞTİRİLMESİ

SERT, Selay

Yüksek Lisans Tezi Kimya Mühendisliği Anabilim Dalı
Tez Danışmanı: Prof. Dr. Firuz BALKAN
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Çalışma iki kısımdan oluşmaktadır. İlk kısmında İzmir'de bir petrokimya fabrikasında mevcut bulunan aromatik hidrokarbon fırının ısıl veriminin arttırılması hedeflenmiştir. Sistemde yer alan mevcut hava ön-ısıtıcısı (APH) ile baca gazı çıkış sıcaklığı 259°C'ye düşürülebilmektedir. Verimin arttırılması için bu sıcaklığın daha da düşürülmesi gereklidir. Bu amaçla iki yöntem ele alınmıştır: 1) Mevcut APH'de alan artışı, 2) APH sonrasına konulacak atık ısı kazanı (WHB). Yapılan hesaplamalar sonucunda ilk yöntemin ekonomik açıdan uygun bulunmadığı görülmüş, kayda değer yakıt tasarrufunun ancak çok yüksek alan artışlarıyla sağlanabileceği sonucuna varılmıştır. Öte yandan ikinci yöntemde; tesisin ihtiyacına yönelik miktarda düşük-orta basınçlı buhar üretiminin APH'ye kıyasla toplamda daha küçük ısı transfer alanı gerektirdiği ve sistemin toplam veriminin bu yöntemle daha düşük maliyetlerle, daha yüksek değerlere çıkabileceği tespit edilmiştir.

Çalışmanın 2. kısmında ise gerçek işletme verileri kullanılarak sistemin klasik ve ileri ekserji analizi yapılmıştır. Fırının klasik ekserji verimi % 41.4 olarak hesaplanmış; ileri ekserji veriminin ise yapılacak uygun modifikasyonlarla ancak % 52.4'e çıkabileceği görülmüştür. Hava önısıtıcısının ekserji verimi % 37.0 olarak bulunmuş, ileri ekserji analizinde kaçınılabilir kayıpların belirlenmesiyle birimin ekserji veriminin % 83.5'e çıkabileceği sonucuna varılmıştır. Aynı şekilde alternatif atık ısı kazanı için klasik ekserji verimi % 45.7 iken ulaşabileceği maksimum ekserji verimi % 62.8 olarak hesaplanmıştır.

**Anahtar sözcükler:** fırın, hava ön-ısıtıcısı, atık ısı kazanı, enerji, ekserji, ileri ekserji analizi.

# ABSTRACT

## ENHANCEMENT OF THERMAL EFFICIENCY OF A NATURAL GAS FURNACE IN A PETROCHEMICAL PLANT

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This study consists two parts. In the first part, it was aimed to improve the thermal efficiency of an aromatic hydrocarbon furnace placed in a petrochemical plant located in Aliağa, İzmir. Exit temperature of the flue (stack) gas is being descreased to 259 °C by an air preheater (APH) existing in the plant. To increase the system effciency this temperature should be further decreased. For this purpose, two processes were investigated. 1) Increasing heat transfer area of existing APH, 2) placing a waste heat boiler after APH and it was seen that first method was not economically appropriate, since the remarkable fuel saving can only be achieved in very high area increase. In the second process, producing medium-low pressure steam, requires less heat transfer area in comparison with the first process and thermal efficiency of the combined system attains higher values with lower costs.

In the second part of the study, the system was investigated from the second law of thermodynamics point of view. Conventional exergy efficiency of the furnace was calculated as 41.4%. By appling advanced exergy analysis, exergy efficiency of the unit can be increased up to 52.4% with suitable modifications. Conventional exergy efficiency of the air preheater was found as 37.0% and it was concluded that by improving working conditions and determining avoidable irreversibilities, the exergy efficiency of APH can be increased up to 83.5%. Likewise, for WHB, it was seen that conventional exergy efficieny was 45.7% and advanced exergy efficiency was 62.8%.

**Keywords:** Furnace, air preheater, waste heat boiler, energy, exergy, advanced exergy analysis.

# ACKNOWLEDGEMENT

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# NOMENCLATURE

| Symbol | Explanation |
|---|---|
| $A$ | area ($m^2$) |
| $C$ | heat capacity flow rate, $\dot{m}C_P$, (kJ/K) |
| $\dot{E}x$ | exergy flow rate (MW) |
| $F$ | LMTD correction factor |
| $h$ | enthalphy (kJ) |
| $\dot{m}$ | mass flow rate (kg/h) |
| $P$ | pressure (bar) |
| $R$ | gas constant (kJ/kmol K) |
| $s$ | entropy (kJ/kmol K) |
| $T$ | temperature (K) |
| $W$ | work (kW) |
| $x$ | molar fraction (-) |
| $\varepsilon$ | efficiency (-) |
| $Q$ | heat transfer rate (kJ) |

## Superscripts

| | |
|---|---|
| $AV$ | avoidable |
| $CH$ | chemical |
| $KN$ | kinetic |
| $PH$ | physical |
| $PT$ | potential |
| $UN$ | unavoidable |

## NOMENCLATURE (continued)

Subscripts

| | |
|---|---|
| *APH* | air preheater |
| *c* | cold |
| *CG* | combustion gas |
| *CV* | control volume |
| *CVC* | chemical vapor compression |
| *D* | destruction |
| *e* | exit |
| *F* | fuel |
| *h* | hot |
| *HC* | hydrocarbons |
| *i* | inlet |
| *L* | loss |
| *lm* | logaritmic mean |
| *NG* | naturalgas |
| *o* | reference state |
| *P* | product |
| *sat* | saturation |
| *st* | steam |
| *surr* | surroundings |

# 1. INTRODUCTION

Fired heaters are commonly used as process heaters in various industries including petroleum rafineries and petrochemical industries. It supplies required heat for process feed by burning an appropriate fuel in the combustion chamber. They are one the most energy consuming units in the plants, hence, thermal efficiency of this devices is very important for plant economy. Thermal efficiency of a fired heater can be defined as the ratio of the amount of heat supplied to process feed and the amount of heat generated by fuel combustion. However, most of the furnaces in industrial systems are not operating with high thermal efficiencies. High temperature stack gases are the main reason of the low efficient operations. A considerable amount of thermal energy is lost via furnace stack and recovering this lost heat should be the main consideration for those being interested in improving the thermal efficiency of heating systems.

Therefore, waste heat recovery units like air preheaters or waste heat boilers are used unavoidably by many factory to increase the thermal efficiency of the systems. In the first part of this thesis, energy analysis of a furnace-air preheater coupled system in a petrochemical plant was performed. Another alternative method, placing a waste heat boiler into this system is considered and thermal design of this recovery unit was accomplished.

In the second part of the study, in addition to enery-based analysis, exergy-based analyses (conventional and advanced exergy analyses) were applied to the existing furnace-air preheater system.

The first law of thermodynamics is conventionally used with the purpose of energy utilization in the industrial systems. It determines the energy losses and effectiveness of the resources. However, it is inadequate when the quality aspect of energy use is taken into account. The exergy analysis, however is based on the second law. It is a more powerful thermodynamic method for assessing and improving the efficiency of processes, devices and systems, as well as for enhancing environmental and economic performances. A conventional exergy analysis identifies the system components with the highest exergy destruction and the processes that cause them. Efficiencies within a system’s component can then be improved by reducing the exergy being destroyed within the component (Shekarchian et.al, 2013). However, conventional exergy analysis is always used to evaluate the performance of an individual unit at certain operating conditions without considering the actual achievable best performance of the unit.

Such limitations in conventional exergy analysis may be considerably decreased by applying advanced or detailed exergy analysis. In the advanced energy analysis real potential for improving a system component is achieved by splitting the exergy destruction into unavoidable and avoidable parts. For this purpose, improvement efforts are then made by focusing mainly on these avoidable parts.

# 2. THEORETICAL BACKGROUND

## PART 1

## 2.1. Industrial Furnaces

A furnace or a fired heater can be defined as a direct-fired heat exchanger used in various industries, petroleum refineries and petrochemical plants to supply high temperature heating to the process fluid. Different unit operations like distillation, alkylation, catalytic reforming, thermal cracking, hydrocracking etc. often require furnaces during the process.

Furnaces generally consist of two parts called radiant section or firebox and a convection section. Required heat for the process fluids is supplied by burning a suitable fuel in the furnace. The combustion reaction of fuel occurs in combustion chamber (burner) which is placed at the bottom section of the fired heaters. When the combustion reaction is completed in burners, very high temperature combustion gases (flue gases) are generated and pass through radiant section of the fired heater. Then, they are sent to convection section where they circulate around tube bundles and give their thermal energy to the cold process feed in the tubes and leave the furnace from the stack. In the Figure 2.1, flow diagram of a typical furnace is shown.

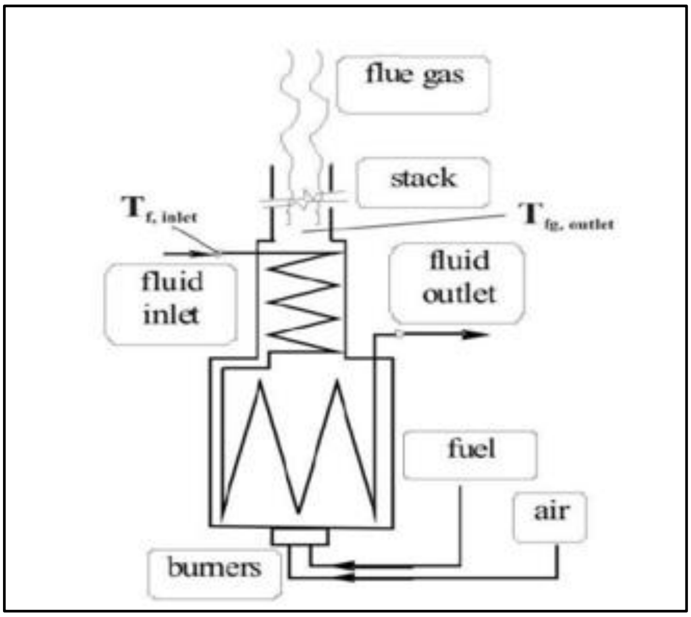

**Figure 2.1.** Flow schematic of a furnace (Jegla, 2011)

### Radiant Section

Radiant section or firebox which recieves almost all heat directly by radiation from the flame, consists of horizontal and vertical radiant tubes located along the walls of radinat section. Since the radiant zone should be lined with fire resisting material, radiant section is the most expensive part of the furnaces.

### Convection Section

Convection section is usually located upon the radiant section of the furnace. Heat transfer mechanism takes place here is convection and generally, finned tubes are prefered to increase heat transfer. The convection section removes heat from the flue gas to preheat the feed in the tubes and significantly reduces the temperature of the flue gas exiting the stack.

### Shield Section

Shield section is located just below the convection section. It contains rows of tubes that shields the convection tubes from the direct radiant heat.

### Stack

Stacks of all fired heaters are located at the top of the units and usually has a cylindrical structure. The breeching which is placed right below the stack, collects the combustion gases and sent them to the atmosphere via stack. Figure 2.2 expresses the sections of a furnace more detaily.

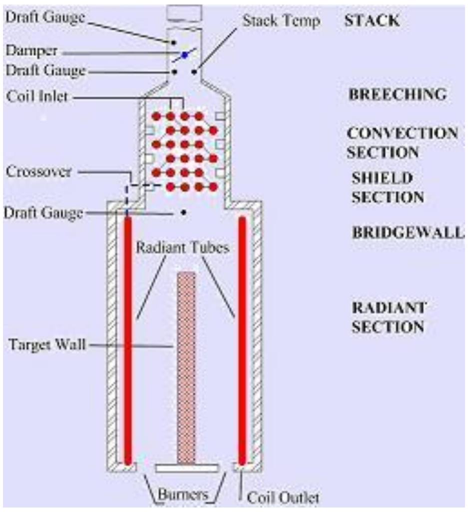

**Figure 2. 2**. Sections of a process heater

To increase the thermal efficiency of a fired heater, following conditions should be satisfied:

- Maximization of the amount of heat delivered to the process feed.
- Minimization of the amount of fuel used.
- Minimization of the stack emissions (waste heat, $CO_2$, $NO_x$)
- Maximization of the safety levels.

## 2.2. Fuels and Combustion

To provide necessary amount of heat to a process feed in a furnace, combustion reaction takes place between an appropriate fuel (natural gas, fuel oil, coal, etc.) and an oxidant (usually air). Efficient combustion process occurs when the largest possible amount of fuel’s heat of combustion is transferred into the process fluid.

**Stochiometric Combustion**

Stochiometric combustion occurs when the fuel reacts with exactly the true amount of oxygen under ideal conditions to achieve complete combustion. In the stochiometric combustion there are no excess fuel or oxygen left. When natural gas (assume contains only methane) combustion reaction is considered, this reaction can be shown as:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \qquad (1)$$

However, when the real combustion reaction takes place, always some excess air is required to insure complete combustion of the fuel. If it’s not used, considerable amount of CO are generated as a result of incomplete combustion which causes reduction in thermal efficiency of the furnace and increase in environmental pollution. Importance of controlling the amount of excess air will be detaily expressed in the next section.

**Fuel**

There are different types of fuels in liquid, gas or solid phase to be used in combustion reactions in boilers, fired heaters or other combustion equipments. In the selection of the type of the fuel; availability, storage, handling and cost of fuel are the important criteria that sould be considered.

Types of fuels can be summarized as follows:

**Solid Fuels (Almost Entirely Bituminous Coals)**: Coal was once a common fuel for industrial furnaces, either hand-fired, stoker-fired, or with powdered coal burners. However, liquid and gaseous fuels gained much more importance in recent years with increasing demand in the accuracy of temperature control and environmental regulation.

**Liquid Fuels (Fuel Oil and Tar)**: To achieve effective combustion of the liquid fuels, atomization is required. Fuel should be separated into little droplets with the aid of steam or air, or in mechanical ways. It is important to maintain the viscosity level of this heavy oils and tar at desired values in the atomizer and it is accomplished by preheating the fuel.

**Gaseous Fuels (Natural Gas, LPG)**: Gaseous fuels are the most common type of fuels nowadays, because of their ease of handling and need of maintenance-free burner systems. They are highly convenient for the areas with high population and industry since the gas is delivered “on tap” via a distribution network.

Natural gas is the most commonly used fuel source of industrial furnaces in Turkey. Other common fuel sources include LPG (liquefied petroleum gas), fuel oil, coal or wood.

### Natural Gas

Natural gas is composed mainly of methane (about 95% by volume). Ethane, propane, butane, pentane, nitrogen, carbon dioxide are the remaining components of natural gas. When properties of methane is investigated and compared with other fuels, usually properties of methane is considered, as it constitutes the largest part of the fuel.

Natural gas is a high calorific value fuel with no sulphur content and it needs no storage facilities. Furthermore, it mixes with combustion air easily and generates no smoke or soot. It’s lighter than air and if any leak is occurred it disperses into air readily.

In our country, a considerable amount of natural gas is consumed in industrial sector (including electricity generation). It is frequently used in petroleum refining, petrochemical plants, metal, chemical, plastic, food processing, glass and paper industries.

## 2.3. Efficiency Improvement of Furnaces

Thermal efficiency of process heating equipments can be defined as the ratio of the amount of heat delivered to the process fluid and the amount of heat produced by the fuel. However, during the heating process a significant amount of heat is lost in the form of flue gases which directly affects the thermal efficiency and depends on the design and operation of the heating equipment.

### 2.3.1. Heat Losses from Fuel-Fired Heating Equipment

Heat lost through the stack in the form of exhaust gases are the lost that cannot not be avoided in any kind of heating processes. When air and fuel are burned in the combustion chamber to generate heat, a portion of this heat is delivered to the process material. When the heat transfer reaches practical limit, rest of the combustion gases are removed from the furnace via stack to make space for the fresh charges. However, these lost gases still hold significant amount of thermal energy in it. Therefore, waste heat can be the major source of thermal inefficiencies in a heating process and reduction of these losses to an acceptable level must be the first step for improving furnace efficiencies.

Other heat losses from the furnace can be expressed as follows:

- Heat lost from the furnace walls.
- Heat transported out of the furnace by the load conveyors, fixtures, trays, etc.
- Radiation losses from openings, hot exposed parts, etc.
- Heat carried by the excess air used in the burners.

Reducing these losses lowers energy consumption and amount of fuel used. Also lowers the emissions of carbon monoxide (CO), nitrogen oxides (NOx) and unburned hydrocarbons (UHCs) and improves the product quality and equipment reliability of the furnaces.

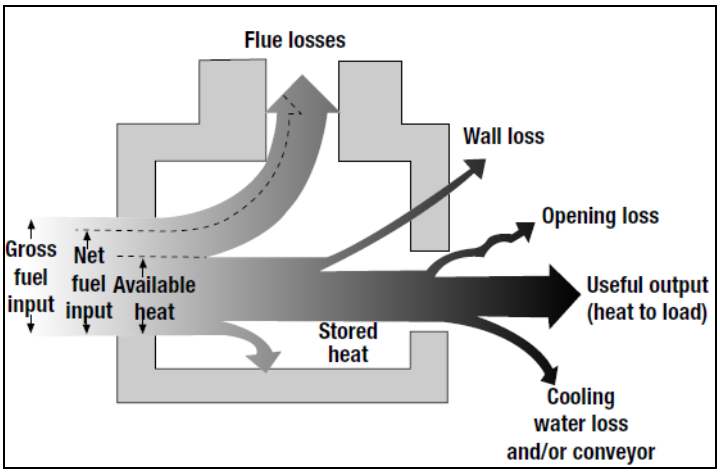

**Figure 2. 3.** Heat losses in industrial heating processes

### 2.3.2. Other Factors Affecting the Furnace Efficiency

#### Fuel-air ratios

To burn the fuel with the correct amount of air is called stoichiometric combustion and it yields high flame temperatures and thermal efficiencies. In the combustion chamber, there is no perfect mixing of fuel and air, supplying the stoichiometric amount of air may not completely burn the fuel (Ugursal, 2013) Hence, to ensure the necessary amount of air reacts with fuel, generally excess air is fed to burners. Controlling level of the excess air is one the most important parameters since, both insufficient and oversupply of air directly affects the furnace efficiency (Cox et.al 1979).

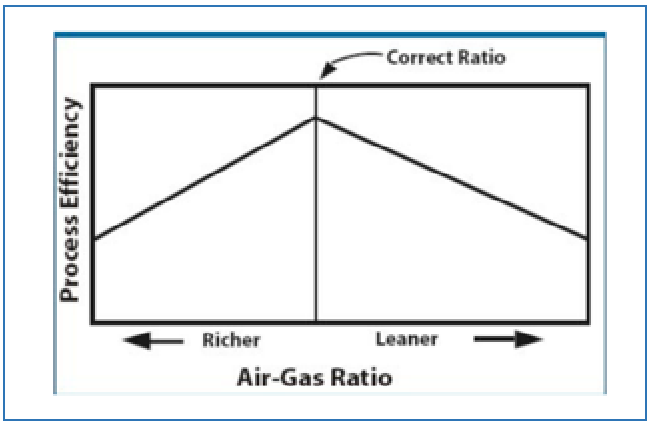

**Figure 2.4.** The variation of efficiency as a function of air-gas ratio

(Cox et.al 1979)

**Air infiltration**

Sometimes excess air infiltrate from the surrounding room in case of negative pressure in the furnace. Because of the draft effect of hot furnace stacks, negative pressures are fairly common and cause low efficiency. A pressure controller should be used in furnace to overcome such a problem.

**Furnace Loading**

Another factor that affects the energy efficiency of heating systems is the scheduling. It means the amount of process material fed to the furnace in a certain period of time and it has a significant role on furnace's energy consumption. Moreover, the amount of exhaust gases also depends on the amount of feed and increases gradually with production volume.

## 2.4. Waste Heat Recovery

Waste heat recovery is a substantial way for improving the thermal efficiencies of various systems. It is about recovering the lost energy from a system and reusing it for heating processes (preheating combustion air, preheating furnace loads, etc.) or generating mechanical or electrical work.

### 2.4.1. Benefits of Waste Heat Recovery

Benefits of 'waste heat recovery' can be divided into two catogories:

**Direct Benefits:**

Since recovering the waste heat in a process reduces the fuel consumption, utility costs hence the process cost, it directly affects the efficiency of the systems.

**Indirect Benefits:**

a) Reduction in pollution: Releasing high temperature stack gas into atmosphere without waste heat recovery, increase the emmission of toxic combustible wastes and increase the environmental pollution levels.

b) Reduction in equipment sizes and utility consumption: Reducing fuel consumption via waste heat recovery directly decreases the amount of combustion gas (flue gas) produced. In this case, size of all flue gas handling equipments (fans, stacks, ducts, burners) will eventually decrease.

On the other hand, this situation also yields a reduction in utility costs used for these devices (such as electricity for fans, pumps etc).

### 2.4.2 Factors Affecting Waste Heat Recovery Feasibility

In the evaluation of waste heat recovery feasibility characterization of the waste heat source and the stream to which the heat will be transferred is required. For this purpose, following parameters of the streams should be well investigated:

- heat quantity,
- heat temperature/quality,
- composition,
- minimum allowed temperature, and operating schedules, availability, and other logistics.

These parameters gives crucial information to the designer about the stream quality and possible materials/design limitations.

**Heat Quantity**: The heat content of a waste heat stream is a measure of how much energy is contained in the stream. Although it’s a significant parameter for the effectiveness of the heat recovery system, it is an insufficient measure alone. Quantity of the waste stream which mainly is the function of temperature should also be taken into consideration for an efficient recovery process.

**Waste Heat Temperature/Quality:** The temperature of the waste heat can be a key factor in the determination of waste heat recovery feasibility. The temperature of the waste determines its quality. The magnitude of the temperature difference between the heat source (waste heat) and sink affects the rate at which heat is transferred per unit surface area of heat exchanger and the maximum theoretical efficiency of converting thermal energy from to another form of energy (i.e., mechanical or electrical). It is has an influence on the selection of materials in heat exchanger designs since corrosion and oxidation reactions increase drastically when the temperature increases.

According to their temperature range, waste heat can be divided into three group.

**1. High Temperature range**- refers to temperature above 650 °C
**2. Medium temperature range**- 230 °C – 650°C

**3. Low temperature range**- below 230° C

High and medium temperature waste heats are generally released from fired heaters while low temperature waste heats are released from condenser, solar flat plate collector, etc.

**Heat Exchanger Area Requirements**: The temperature of waste heat directly affects the rate of heat transfer between a heat source and heat sink as shown in Eq.2.

$$Q = UA\Delta T \qquad (2)$$

where $Q$ is the heat transfer rate; $U$ is the heat transfer coefficient; $A$ is the surface area for heat exchange; and $\Delta T$ is the temperature difference between two streams. For the same heat transfer rate when the overall heat transfer coefficient of the system is constant, as the temperature difference gets larger, required heat transfer area gets smaller.

**Waste Stream Composition:** Although chemical composition and phase of waste heat streams has any direct impact on waste heat recovery, they affect the properties like thermal conductivity and heat capacity. However, they affect the rate of heat transfer of the heat exchanger, hence they can change the effectiveness of the heat transfer.

**Fouling:** Another important consideration is the interaction between chemicals in the waste gas stream and heat exchanger construction materials. Fouling is a common problem in heat exchange, and has a significant effect on the reduction of heat exchanger effectiveness or the system failure. Nevertheless, there are several ways for reducing fouling in the system like filtering contaminated streams, constructing the exchanger with advanced materials, increasing heat exchanger surface areas, and designing the heat exchanger for easy access and cleaning.

**Minimum Allowable Temperature:** According to fuel used, produced combustion gases contain different concentrations of $CO_2$, $H_2O$ (g), $NO_x$, $SO_x$, unoxidized organics, and minerals. If exhaust gases are cooled below the dew point temperature, the water vapor in the gas will condense and deposit corrosive substances on the heat exchanger surface. Hence, heat exchangers are usually designed to maintain the temperature of combustion gases above the condensation point. The minimum temperature for preventing corrosion depends on the composition of the fuel. For example, combustion gases from natural gas might be cooled at most 120°C, while exhaust gases from coal or fuel oils with higher sulfur contents may be limited to 150ºC to 175°C.

## 2.5. Waste Heat Recovery Systems

Waste heat is recovered in a process via heat exchangers. These industrial heat exchangers can be various types.

### 2.5.1. Recuperator

Recuperators are used in recovering flue gas waste heat in medium- to high-temperature applications such as soaking or annealing ovens, melting furnaces, afterburners, gas incinerators, radiant-tube burners, and reheat furnaces. A typical recuperative heat exchanger is shown in Figure 2.5.

In a recuperator, hot combustion gases in tubes are cooled by the incoming combustion air from the atmosphere flowing outside the tubes and carrying extra thermal energy into the furnace burners. So, energy needed to be supplied by burning fuel is reduced. Saving fuel also means a reduction in combustion air. Hence, stack losses are decreased not only by lowering the combustion gas temperatures but also lowering the quantity of exhaust gases.

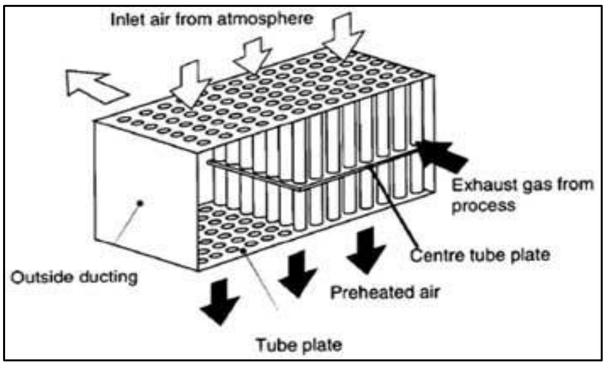

**Figure 2. 5.** A general illustration of a recuperator

### 2.5.2. Regenerator

Regenerators which have large sizes and high capital costs rather than recuperators (Turner and Doty, 2006) are generally used in process with large capacities. They are suitable for high temperature industrial applications with dirty exhaust. They are higly preferable in glass and steel manufacturing.

### 2.5.3. Rotary Regenerator/Heat Wheel

Rotary regenerators are commonly used in low-medium temperature waste heat recovery systems where heat exchange between large amounts of air having small temperature differences is required. Ventilation systems are the main applications of these heat exchangers.

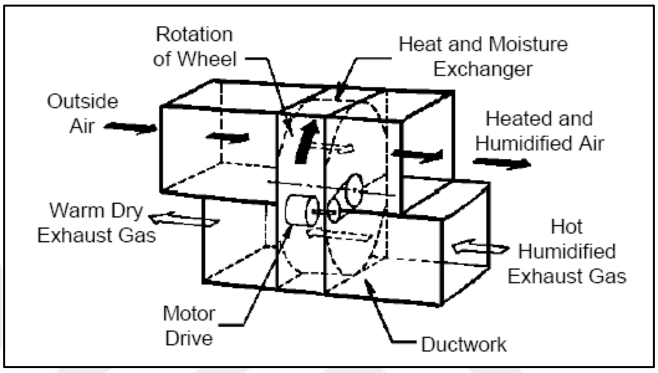

**Figure 2.6.** Rotary regenerator (Dean, 1980)

### 2.5.4. Waste Heat Boilers

In waste heat boilers, the hot waste gases are flowing through a number of parallel tubes containing water. Usually finned tubes are prefered to increase the effective heat transfer area on the gas side. Inside the tubes, the water is vaporized and then collected in a steam drum to be used as heating or processing steam. The pressure of produced steam and the rate of steam production basically depends on the temperature of waste heat. Figure 2.7 shows a typical schematic of a waste heat boiler frequently used in different industries. Waste heat boilers are designed in a capacity of 25 $m^3$ - 30,000 $m^3$ / min. of exhaust gas.

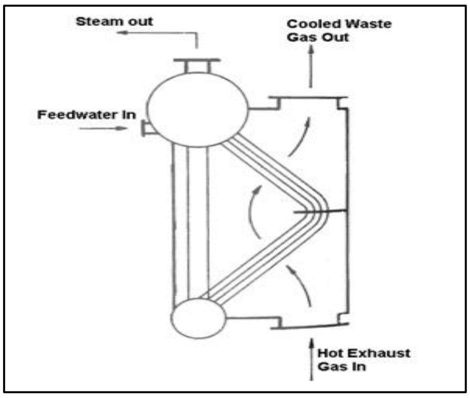

**Figure 2.7.** Waste heat boiler (Goldstick and Thumann, 1986)

# PART 2

## 2.6. EXERGY ANALYSIS OF THERMAL SYSTEMS

### 2.6.1. Exergy

The increase in energy demands, decline in energy resources and the link between energy utilization and environmental impacts have resulted in calls for a sustainable approach to the development and management of the earth's energy resources (Rosen and Dincer, 2001). With finite energy resources and large (and increasing) energy demands, it becomes increasingly important to understand the mechanisms which degrade the quality of energy and energy resources and to develop systematic approaches for improving the systems (Gong and Wall, 1997).

The first law of thermodynamics states that during an interaction energy of a matter can change form from one to another but the total amount of energy remains the same. However, second law of thermodynamics interested in the quality of this energy and propounds that in real processes although the quantity of energy remains constant, the quality of energy decreases. At this point, introducing the concept of exergy helps us to quantify this quality or "work potential" of energy (Dincer and Cengel, 2001).

### 2.6.2. What's Exergy?

Exergy can be defined as the maximum theoretical amount of work which can be produced by a system or a flow of matter or energy as it comes to equilibrium with a reference environment (Dincer and Cengel, 2001). Exergy is a measure of the potential of the system or flow to cause change and unlike energy, exergy is not subject to a conservation law (except for ideal, or reversible, processes). Rather, exergy is consumed or destroyed due to irreversibilities in any real process. The exergy consumption during a process is proportional to the entropy created due to irreversibilities associated with the process.

### 2.6.3. Exergy vs. Energy

In Table 2.1 a general comparison between the energy and the exergy is given.

**Table 2.1.** Comparison of energy and exergy

| ENERGY | EXERGY |
|---|---|
| ➢ is dependent on the parameters of matter or energy flow only and independent of the environmental parameters. | ➢ is dependent both on the parameters of matter or energy flow and on the environmental parameters. |
| ➢ has the values different than zero | ➢ is equal to zero in dead state (in equilibrium with the environment) |
| ➢ is motion or ability to produce motion. | ➢ is work or ability to produce work. |
| ➢ is always conserved in a process, so can neither be destroyed or produced. | ➢ is always conserved in a reversible process but is always consumed in an irreversible process. |
| ➢ is a measure of quantity only. | ➢ is a measure of quantity and quality due to entropy. |

### 2.6.4. Exergy Analysis

Exergy analysis is a powerful tool to identify and quantify energy degradation in a process. It determines the types, locations and true magnitude of energy losses within a system and is widely used in many optimization studies of energy systems. Most cases of thermodynamic inefficiencies cannot be detected by an energy analysis (Rosen, 2002). Some particular processes such as throttling, heat transfers, expansion and friction have no energy losses but they have exergy losses since a degradation in the quality of energy always occurs.

Exergy analysis can reveal whether or not and by how much it is possible to design more efficient energy systems by reducing the inefficiencies caused by entropy generation in existing systems. It provides more meaningful information for efficiency improvement of a system (Rosen and Dincer, 2001). It is also worth to mention that exergy analysis can lead to a significant reduction in the use of natural resources and the environmental pollution by reducing the rate of discharge of waste products.

Figure 2.8 illustrates exergy flow of a system or process. Exergy analysis of thermal systems are accomplished by exergy balance. Detailed information about exergy balance is given in “Material and Method” part of the study.

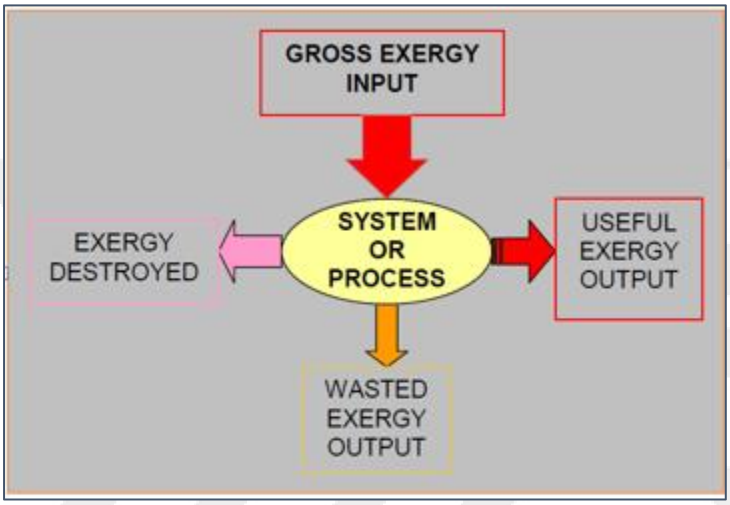

**Figure 2.8**. Exergy flow within a system

### 2.6.5. Advantages and benefits of using exergy analysis

- Exergy analysis is a substantial measure of deviation from ideality in a system.
- It can bring different aspects on design-based studies with a combination of economical analysis. For example, exergoeconomics and thermoeconomics are widely used methods to develop economic evaluations.
- Rather than energy analysis, more meaningful results are obtained via exergy analysis about the efficient use of energy resources, amount of entropy generation within the system components, possible cause of irreversibilities and ways of elimination of these irreversibilities.

## 2.7. ADVANCED EXERGY ANALYSIS

All energy conversion systems should be analyzed by energetic, economic, and environmental aspects for a good operation. Exergy-based analyses are very susceptible ways for assessing the performance of energy conversion systems. Exergy-based analyses help designer determine the source and the magnitude of thermodynamic irreversibilities and energy sources should be used more effectively. Nevertheless, this method becomes insufficient when true potential improvement of the system is taken into consideration. Moreover, they give no information about the intreaction between the system’s components (Acıkkalp et.al, 2014)

This lack of information can be eliminated by introducing advanced exergy analysis. In the advanced exergy analysis, splitting the exergy destruction into unavoidable and avoidable parts in the k-th component provides a more realistic measure of the potential for improving the thermodynamic efficiency of a component (Gungor et.al, 2013). The exergy destruction rate that cannot be reduced due to technological limitations such as availability and cost of materials and manufacturing methods is the unavoidable part of the exergy destruction. The remaining part represents the avoidable part of the exergy destruction (Morosuk and Tsatsaronis, 2009). In order to calculate the amount of unavoidable exergy desturction, unavoidable conditions of the system should be well defined by the designer.

### 2.7.1. Real and unavoidable operation conditions

Conventional exergy analysis is performed under the real operation conditions with real operating data. With the aid of this analysis, actual thermodynamic efficiencies for all components in energy system are considered. However, in conventional exergy analysis application, one cannot get insight into true improvement potential of the entire energy system or individual components. Moreover, interactions of other components are not recognized.

In order to identify the potential for possible further improvement of each component, it is necessary to define the so-called unavoidable operation conditions which are going to be used in the determination of the unavoidable exergy destruction parts. The unavoidable operation conditions are those which are not probably going to be satisfied in the near future if one takes into account the current development tendency for a particular system component. Defining of the unavoidable operation conditions is based on an assessment by decision makers and is more or less influenced by knowing the process, theoretical knowledge, following the current progress in science (Vuckovıv et.al, 2012). In general, the unavoidable operation conditions are the better than the real working ones, but they are not equivalent to the ideal (theoretical) operation conditions.

# 3. LITERATURE SURVEY

In the past years, various studies have been made about improving furnace efficiency in different industries including petrochemical industry and petroleum rafineries. High stack gas temperature that causes low thermal efficiency is the most common problem of industrial furnaces. Introducing waste heat recovery devices such as air preheater or waste heat boiler overcome this common problem, decrease the stack gas temperatures and reduce the fuel consumption and the level of $CO_2$, $NO_x$ emissions. Tahmasebi et.al (2012) performed a study about the effect of increasing fresh air temperature supplied to burners by utilizing air preheater (APH) on the fuel consumption in a refinery in Iran. It was concluded that 5-6% fuel reduction can be made with a surface area (APH) of 1440 $m^2$ when the fresh air temperature is increased up to 200 °C. Udriste and Musat (2012) present a mathematical model for sizing APHs for the heat treatment of furnaces used the steel construction materials and find that when the air temperature leaving the APH is 500°C and the stack gas temperature leaving the APH is 508°C, required heat transfer area is 150.7 $m^2$.

Different types of APHs have been investigated during the past studies. Yodrak et.al (2010) conducted a study about a furnace in a hot brass forging process. The purpose of the study was to design, construct and test the waste heat recovery by heat pipe air-preheater from the furnace. They investigated the effect of hot gas temperature, internal diameter and tube arrangement of APH on heat transfer rate. It was indicated that when the hot gas temperature increased, the heat transfer rate also increased. If the internal diameter increased, the heat transfer rate increased and when the tube arrangement changed from inline to staggered arrangement, the heat transfer rate increased. A similar study was carried out by the same authors (2011) about waste heat recovery in the hot forging process by the thermosyphon air-preheater. Same parameters were investigated and it was seen that when the flow arrangement changed from parallel flow to a counter flow arrangement, the heat transfer rate increased.

In this study, however a cross flow recuperative type APH having a heat transfer surface area of 614 $m^2$ is going to be investigated. Using a simulation program, effect of increasing combustion air temperature on fuel economy and effect of decreasing stack gas temperature on environmental impacts will be studied.

Schekarchian and Zarifi (2013) performed a study in order to analyze economic benefits of incorporating both heat recovery and air preheating techniques into the existing different fired heater units in a petroleum refinery. In addition to energy analysis, second law efficiency and the rate of irreversibilities in the system were also analyzed via an exergy analysis. It was seen from the study that both analyses were satisfying that heat recovery enhanced both first and second law efficiencies from 63.4% to 71.7% and from 49.4% to 54.8%, respectively. In addition, heat recovery and air preheating methods leads to a substantial fuel reduction (7.4%) while simultaneously decreasing the heat loss and irreversibilies of the unit.

Regulagadda et.al (2010) carried out a study about second law analysis of a thermal power plant along with a parametric study that investigates the effects of different parameters like temperature and pressure on the system performance. They concluded that the maximum exergy destruction occurred in boiler unit and efforts at improving plant’s performance should be directed to improving boiler efficiency.

Bejan et.al and Tsatsaronis (1996) developed ‘fuel-product’ concept which became very cruical in recent studies, in defining exergetic efficiency, one of the most significant criterion for evaluating thermal perfomance of a component in a system.

Furthermore, Tsatsaronis et.al (2002) introduced advanced exergy analysis by splitting exergy destructions into avoidable and unavoidable parts and with the aid of a cogeneration plant example, he proved the advantages of dividing exergy destruction into avoidable and unavoidable parts by exergoeconomic analysis.

Morosuk and Tsatsaronis (2011) explanied how to calculate the amount of unavoidable exergy destruction in an advanced exergy analysis and they applied their approach to a simple gas-turbine system that reveals the potential improvement and interaction among the system’s components.

Vuckovic et.al (2012) also accomplished a study on conventional and advanced exergy analysis and exergoeconomic performance evaluation of an industrial plant. They found that more than 97% of exergy destruction was caused by steam boiler and 92.3% of exergy destruction in steam boiler can be avoided.

In this study, both conventional and advanced exergy analysis were performed on an existing furnace-APH coupled system by using real operating data of the plant. By the help of conventional exergy analysis, the components with the highest exergy destruction and irreversibility were determined. Moreover, true improvement potential of each unit in the system were determined by splitting exergy destructions into avoidable and unavoidable parts in advanced exergy anaylsis. Apart from the other studies, a new expression for modified exergetic efficieny was defined and used in calculations.

# 4. MATERIAL AND METHOD

In this study, a fired heater-air preheater system coupled system were investigated on the basis of the first law and the second law of thermodynamics;

- ✓ To modify the existing air preheater in the plant by increasing the heat transfer area and determine the thermal efficiency increase and fuel consumption decrease in APH.
- ✓ To install a waste heat boiler next to air preheater and produce low-medium steam for the plant.

The study consists of two parts. The first part is about increasing thermal energy efficiency of aromatic furnace by recovering the waste heat from furnace. To achieve this, two different pathways were investigated and the results were compared. The methods are as follows:

The calculations were performed in Matlab 2012b with the aid of mass and energy balances and heat transfer rate equations by using real operating data.

The second part is about the determination of irrevesibilities within the system to identify the possible reasons of thermal inefficiencies more clearly by using exergy analysis and revealing the real potential improvements by “advanced exergy analysis”.

## 4.1. Process Description

The furnace in the plant has a capacity of 42 Gcal/h and it is used to vaporize process fluid composed of hydrocarbon mixtures. 998,700 kg/h of aromatic mixture is fed to the heater at 273°C. Natural gas is used as fuel for combustion and combustion reaction takes place in 20 burners with 20% excess air. The flue gas exits the furnace stack at 408 °C is fed to the air preheater unit. Since flue gas leaving the furnace is at very high temperature, significant amount of energy is lost with it. Recovering this high-energy flue gas via an APH increase the temperature of fresh combustion air entering the furnace and eventually decreases the amount of fuel used for combustion. So, this can directly increase the thermal efficiency of the furnace. Available APH in the plant is a cross flow recuperative heat exchanger with a heat transfer area of 614 m$^2$. In the APH, the temperature of flue gas coming from the furnace is being decreased to 259°C while the temperature of ambient air increased up to 195°C. The flow diagram of the system is shown in Figure 4.1.

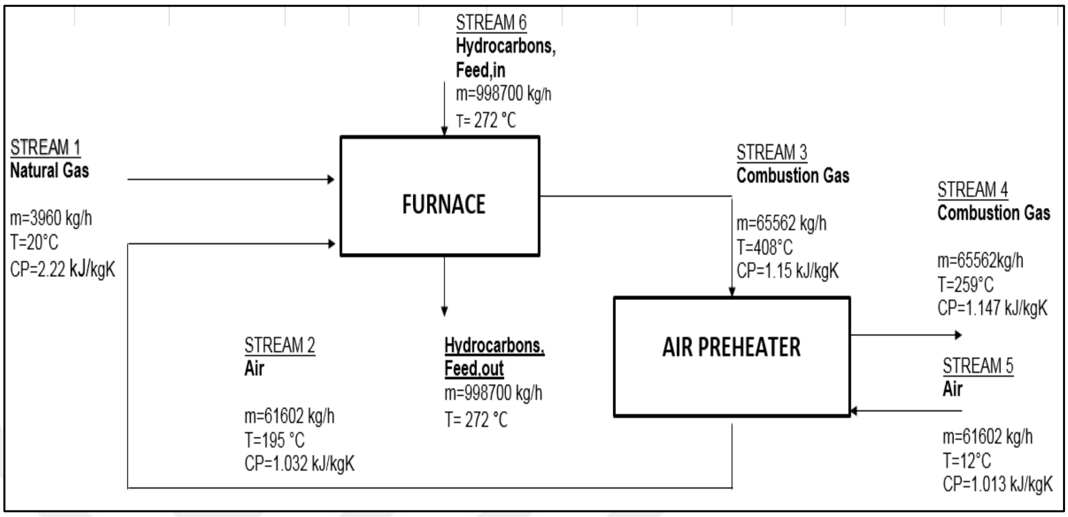

**Figure 4.1** Flow scheme of furnace-air preheater coupled system

## 4.2. Combustion in Burners

Fuel used in combustion is natural gas. The composition of natural gas was taken as 95%methane and 5% ethane by volume. Combustion reaction of the fuel with 20% excess air is as follows:

$$CH_4 + 2.4(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O_{(g)} + 0.4O_2 + 9.02N_2 \qquad (3)$$

$$\Delta H^{\circ}_{rxn,1} = -802.3\frac{kJ}{mol}$$

$$C_2H_6 + 4.2(O_2 + 3.76N_2) \rightarrow 2CO_2 + 3H_2O_{(g)} + 1.4O_2 + 15.79N_2 \qquad (4)$$

$$\Delta H^{\circ}_{rxn,2} = -1427.8\frac{kJ}{mol}$$

Calculation of heat of combustion and operating data of furnace were shown in Appendix.

## 4.3 Mass and Energy Balances in Furnace

Mass balance in control volume can be described as:

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt} \qquad (5)$$

At steady state, mass balance around the furnace gives us:

$$\dot{m}_{NG} + \dot{m}_{Air} = \dot{m}_{CG} \qquad (6)$$

To calculate the heat load of furnace an energy balance around the furnace should be written.

Control volume energy balance equation at steady state:

$$Q_{CV} + \sum \dot{m}_i h_i = W + \sum \dot{m}_e h_e \qquad (7)$$

In the abscence of any shaft work or heat, rearranging the equation gives:

$$\dot{m}_{NG} Cp_{NG}\left(T_{NG,in} - T_{ref}\right) + \dot{m}_{Air} Cp_{Air}\left(T_{Air,in} - T_{ref}\right) + \dot{m}_{NG}\Delta H_{comb} - \dot{m}_{CG} Cp_{CG}\left(T_{CG,out} - T_{ref}\right) - \dot{m}_{feed}\Delta H_{feed} = 0 \qquad (8)$$

## 4.4. Improving Waste Heat Recovery

### 4.4.1 Alternative 1: Increasing Heat Transfer Area of Available Air Preheater

#### 4.4.1.1. Mass and Energy Balances on Air Preheater

Existing air preheater in the plant is a cross flow recuperative air preheater. Combustion gas coming from the furnace is the hot fluid in this heat exchanger and the cold fluid is ambient air.

General mass balance around the air preheater:

$$\dot{m}_{hot,in} + \dot{m}_{cold,in} = \dot{m}_{hot,out} + \dot{m}_{cold,out} \qquad (9)$$

Energy balance:

Energy gained by air: $\dot{Q}_{air} = \dot{m}_{air}(h_{air,out} - h_{air,in})$ (10)

Energy lost by combustion gas: $\dot{Q}_{CG} = \dot{m}_{CG}(h_{CG,in} - h_{CG,out})$ (11)

Overall convective rate of heat exchange:

$$\dot{Q}_{air} = \dot{Q}_{CG} = UA_{APH}\Delta T_{lm} \qquad (12)$$

At steady state when $Q_{CV} = W = 0$ ;

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e \qquad (13)$$

$$\sum \dot{m}_i h_i = \sum \dot{m}_e h_e = UA_{APH} \Delta T_{lm} \qquad (14)$$

Heat load of APH is calculated in Appendix.

In the rate equation of the exchanger $\Delta T_{lm}$ represents the logaritmic mean temperature of fluids and can be determined in a cross flow exchanger as:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \qquad (15)$$

where $\Delta T_1 = T_{h,i} - T_{c,o}$ and $\Delta T_2 = T_{h,o} - T_{c,i}$.

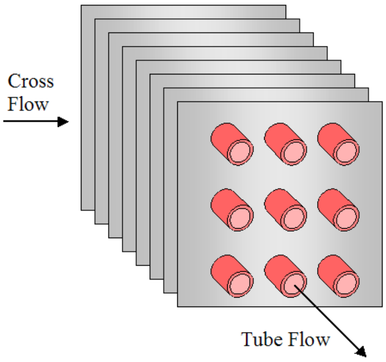

**Figure 4.2** Cross flow heat exchanger (Cengel Y.A., 2012)

For cross flow; logaritmic mean temperature in counter flow must be multiplied with a correction factor of F.

$$\Delta T_{mean} = F \Delta T_{lm_{counter-flow}} \qquad (16)$$

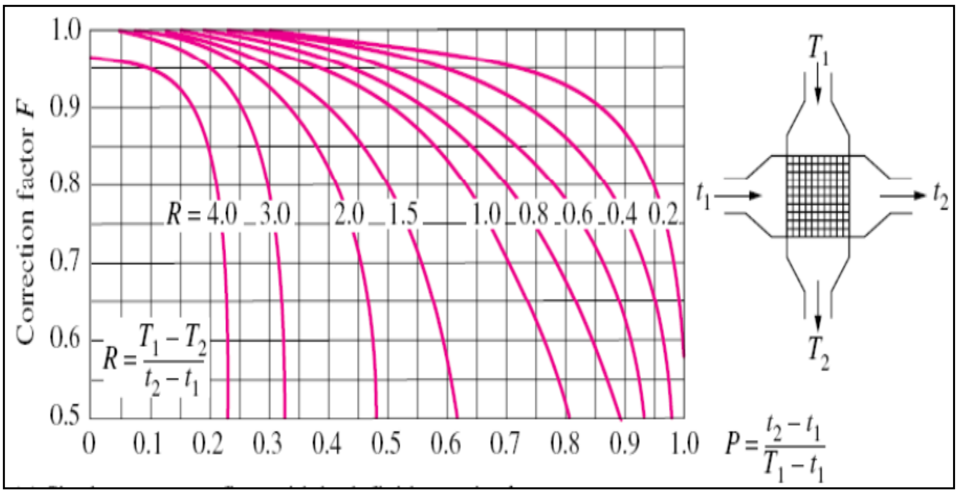

**Figure 4.3** Correction factor for cross flow heat exchangers (Incropera et.al, 2007)

The overall heat transfer coefficient, U for the heat exchanger was calculated indirectly by using Eq.17.

$$U = \frac{Q_{APH}}{A_{APH}\,\Delta T_{mean}} \qquad (17)$$

In this study, a simulation program on MATLAB was written to calculate the increase in furnace efficiency and decrease in fuel consumption for various percentages of heat transfer area increase.

#### 4.4.1.2. Methods used in Heat Exchanger Analysis

In the analysis of heat transfer in heat exchangers, both LTMD (logaritmic mean temperature difference) and ε-NTU methods are frequently used. LMTD method is suitable in the determination of overall heat transfer coefficient (or area) when both inlet and outlet temperaures of streams are known. ε-NTU method, on the other hand, is more convenient in the calculation of outlet stream temperatures when only the inlet temperatures and overall heat transfer coefficient of the heat exchanger are known, besides area.

ε-NTU method was used in this simulation. This method is based on a dimensionless parameter called effecitiveness, ε and it can be defined as:

$$\varepsilon = \frac{Actual\ heat\ load\ of\ heat\ exchanger}{Maximum\ possible\ heat\ load\ of\ heat\ exchanger} \qquad (18)$$

Actual heat transfer rate of a heat exchanger is:

$$Q = C_c\,\Delta T_C = C_h\,\Delta T_h \qquad (19)$$

Where $C_h$ and $C_c$ are heat capacity flow rates of hot and cold fluid and can be written as:

$$C_c = \dot{m}_c Cp_c \quad \text{and} \quad C_h = \dot{m}_h Cp_h \qquad (20)$$

To calculate the maximum possible heat transfer rate, the term of 'maximum temperature diffirence' should be defined first. That is:

$$\Delta T_{max} = T_{h,in} - T_{c,in} \qquad (21)$$

Heat transfer in an exchanger reaches its maximum value in two different ways: either the hot fluid temperature is decreased to cold fluid temperature, or the cold fluid temperature is increased up to hot fluid temperature. These two conditions satisfy each other only if the heat capacity flow rates of both fluids are identically equal. However, in most of the real systems, heat capacity flow rates are not equal. At this point, the fluid which has smaller heat capacity value is used in determination of the maximum heat transfer rate.

$$Q_{max} = C_{\min}(T_{h,in} - T_{c,in}) \qquad (22)$$

If $C_c < C_h \rightarrow C_{\min} = C_c \rightarrow C_{\max} = C_h$

otherwise, $C_{\min} = C_h \rightarrow C_{\max} = C_c$

$$C_R = \frac{C_{\min}}{C_{\max}} \qquad (23)$$

If the effectiveness of a heat exchanger is known, the actual operating heat load of the exchanger can be calculated by using Eq.18 and with the aid of heat load, outlet temperatures of the streams can be directly calculated from energy balance.

The efficiency ε depends on the heat exchanger geometry, flow pattern (parallel flow, counter-flow, cross-flow, etc.) and the number of transfer units.

For any heat exchanger:

$$\varepsilon = f(NTU, C_R) \qquad (24)$$

NTU (number of transfer units) designates the non-dimensional heat transfer size of the heat exchanger:

$$NTU = \frac{UA}{C_{min}} \qquad (25)$$

Relationships between the effectiveness and NTU have been established for a large variety of heat exchanger configurations. Figure 4.4 expresses the relation between $\varepsilon$ and $NTU$ in a cross flow heat exchanger:

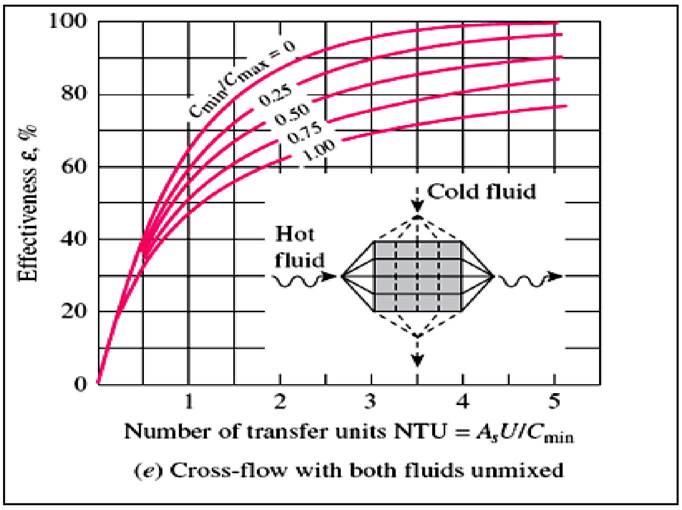

**Figure 4.4** Effectiveness vs NTU graph for cross flow heat exchanger

(Incropera et.al, 2007)

Following correlation also gives the relation between $\varepsilon$ vs $NTU$ in a cross flow heat exchanger:

$$NTU = -\ln\left[1 + \frac{\ln(1-\varepsilon C_R)}{C_R}\right] \qquad (26)$$

### 4.4.1.3. Algorithm of Simulation

In the calculations, following assumptions were conducted:

- ✓ Flow rates and heat capacities of hot and cold streams are constant.
- ✓ Inlet temperatures of combustion gas ( hot fluid) and air (cold fluid) are constant.
- ✓ Overall heat transfer coefficient is independent of the fluid temperature and position
- ✓ There are no phase changes in the fluids
- ✓ There is no energy loss to the environment

<u>Calculation steps:</u>

- ➢ Computation of $C_c$ and $C_h$ for the given flow rates and fluid properties.
- ➢ Identification of $C_{min}$ and determination of.heat capacity ratio, $C_R$.
- ➢ Inlet temperatures are fixed and outlet temperatures will be changed by designer. Since the inlet and outlet temperatures are known, calculate ε.
- ➢ Calculate the overall heat transfer coefficient, U

- When e and C and the flow arrangement are known, determine NTU from the ε-NTU equations.
- When NTU is known, calculate the total heat transfer surface area

### 4.4.2. Alternative 2: Waste Heat Boiler

In the second alternative, design of a waste heat boiler was considered. The combustion gas leaves the available APH in the plant still has a high temperature and carries high thermal energy. By using this thermal energy, low-medium pressure steam can be produced via a waste heat boiler.

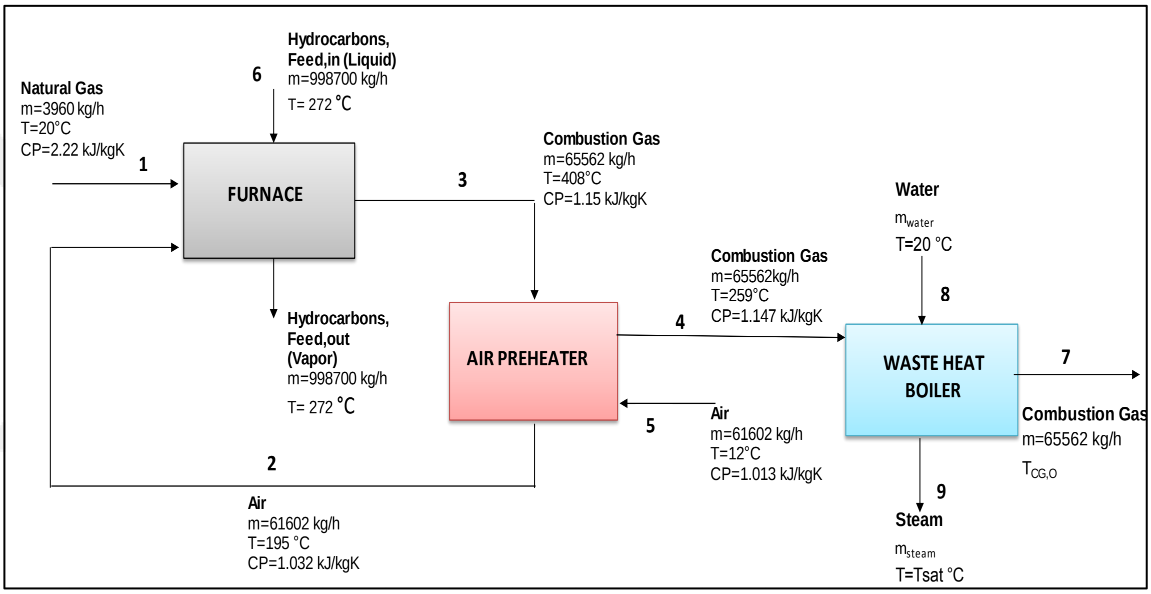

**Figure 4.5** Flow scheme of the system with waste heat boiler

#### 4.4.2.1. Mass and Energy Balance in Waste Heat Boiler

Steady state mass balance in the boiler can be written as:

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt} = 0 \qquad (27)$$

$$(\dot{m}_{CG} + \dot{m}_{W}) - (\dot{m}_{CG} + \dot{m}_{st}) = 0 \qquad (28)$$

Steady state energy balance of the waste heat boiler:

$$\dot{Q} + \dot{W}_s = \Delta\dot{H} + \Delta E_k + \Delta E_p = \frac{dE}{dt} = 0 \qquad (29)$$

In the abscence of any shaft work, heat loss, kinetic and potential energy changes:

$$\Delta\dot{H} = 0$$

$$\dot{m}_{CG}Cp_{CG}\left(T_{CG,in} - T_{ref}\right) + \dot{m}_{w}Cp_{w}\left(T_{w,in} - T_{ref}\right) - \dot{m}_{CG}Cp_{CG}\left(T_{CG,out} - T_{ref}\right) - \dot{m}_{st}Cp_{st}\left(T_{st} - T_{ref}\right) - \dot{m}_{st}h_{fg} = 0 \qquad (30)$$

Since the produced steam is at saturated temperature for given pressure:

$$T_{st} = T_{sat}$$

Saturated temperature of a component for a given vapor pressure, $P$ is calculated by using Antoine equation:

$$T_{sat} = \frac{B}{A - lnP} - C \qquad (31)$$

where $A, B$ and $C$ are Antoine constants and given in the Appendix.

It was assumed that steam behaves like an ideal gas, hence, latent heat of vaporization of water given in Eq.32 is the function of temperature only:

$$h_{fg} = 2501.3 - 2.36T - 0.00046T^2 - 0.0000126 \quad [kJ/kg] \qquad (32)$$

### 4.4.2.2. Design Calculation of Waste Heat Boiler

To calculate the required area in a waste heat boiler, following equation is used:

$$\dot{Q}_{WHB} = UA\Delta T_{lm} = \dot{m}_{CG}Cp_{CG}\left(T_{CG,in} - T_{CG,out}\right)$$

$$= \dot{m}_{st}Cp_{st}(T_{sat} - T_{w}) + \dot{m}_{st}h_{fg} \qquad (33)$$

where $U$ is the overall heat transfer coefficient for the boiler system. $\Delta T_{lm}$ is the logaritmic mean temperature of fluids and calculated as in Eq.34:

$$\Delta T_{lm} = \frac{(T_{CG,in} - T_{sat}) - (T_{CG,out} - T_{sat})}{ln\frac{(T_{CG,in} - T_{sat})}{(T_{CG,out} - T_{sat})}} \qquad (34)$$

As in the APH design, ε-NTU method was used to calculate the required area of the boiler. For various pressures, required boiler area and amount of produced steam ($\dot{m}_{st}$) were calculated on MATLAB and compared for feasibility.

### 4.4.2.3. Algorithm of Simulation

In the calculations, following assumptions were stated:

- ✓ Cold fluid (water) was flowing inside the tubes while hot fluid (combustion gas) was flowing outside the tubes.
- ✓ Tube configuration was triangular pitch and the tubes had dimensions of 21 mm inlet ($d_i$) and 25 mm outlet ($d_o$) diameter with 2 m tube length.
- ✓ Recirculation ratio in the boiler is 4. ($CR$=4)
- ✓ Inlet conditions of water and combustion gas were constant.
- ✓ Flow rates and physical properties of hot (combustion gas) and cold (water and steam) streams are constant.
- ✓ There is no energy loss to the environment.
- ✓ Desired pressure of the produced steam were changed from 2 bar to 4 bar.

Calculation steps:

- ➢ Calculate the saturation temperature ($T_{sat}$) by using Antoine equation at given pressure.
- ➢ Calculate the latent heat of vaporization ($h_{fg}$) with respect to saturation temperature.
- ➢ Calculate the maximum conversion ratio ($x_{max}$) from liquid phase to gas phase.
- ➢ Calculate the total amount of heat transfer rate as the sum of sensible and latent heat transfer rates.
- ➢ Assume the total number of tubes required and then apply trial and error method.
- ➢ Once the number of tubes were assumed, calculate the outside heat transfer coefficient ($h_o$) by using appropriate correlations.
- ➢ To calculate inside heat transfer coefficient ($h_i$) seperate the calculations into two parts as sensible heating zone and latent heating zone.
- ➢ Then calculate the $h_i$ in sensible heating zone by using suitable correlations according to flow regime.
- ➢ Since both $h_i$ and $h_o$ were determined, calculate the overall heat transfer coefficient and required heat transfer area for the sensible heating zone.

- Calculate $h_i$ in the latent heating zone as the combination of nucleate boiling ($h_{nb}$) and convective boiling ($h_{cb}$) according to Chen's Method (Chen, 1963) for two-phase boiling.
- Then calculate the overall heat transfer coefficient and required heat transfer area for the latent heating zone.
- Calculate total heat transfer area as the sum of sensible and latent heat transfer area and then calculate the total number of tubes.
- If the difference in assumed and calculated number of tubes are close to zero, stop the calculation.
- Determine the amount of steam produced at that pressure.

## 4.5. Conventional Exergy Analysis of Furnace, Air Preheater and Waste Heat Boiler

In the conventional exergy analysis, exergy value of a stream consists of four parts:

$$\dot{E}x = \dot{E}x^{PH} + \dot{E}x^{PT} + \dot{E}x^{KN} + \dot{E}x^{CH} \qquad (35)$$

where $\dot{E}x^{PH}, \dot{E}x^{PT}, \dot{E}x^{KN}, \dot{E}x^{CH}$are the physical, potential, kinetic and chemical exergy rates, respectively. When the change of velocity and elevation of the system is zero or small, the terms of $\dot{E}x^{PT}$and $\dot{E}x^{KN}$ can be ignored as in this study and the total exergy rate of each stream can be represented by the sum of the $\dot{E}x^{PH}$and $\dot{E}x^{CH}$only, where;

$$\dot{E}x^{PH} = \dot{m}[(h - h_o) - T_o\,(s - s_o)] \qquad (36)$$

$$\dot{E}x^{CH} = \dot{m}[\sum x_k e^{CH} + RT_O \sum x_k \ln(x_k)] \qquad (37)$$

In Eq(37), $e^{CH}$ indicates the standart molar chemical exergy of species and $x_k$ indicates the molar composition of a component in a stream.

Since the exergy rates of streams are determined with respect to reference state conditions in any study, calculations of the available system was performed by designating dead state conditions as $T = 298.15$ K and $P = 1.013$ bar.

## 4.6. Exergy Balance

The control volume exergy balance at the steady state for the *k*-th component of the energy system can be expressed as follows:

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{D,k} + \dot{E}_{L,k} \qquad (38)$$

where $\dot{E}_{F,k}$, $\dot{E}_{P,k}$ are the fuel and product exergy rates, $\dot{E}_{D,k}$ is the rate of exergy destructed in the component due to irreversibilites and provides a thermodynamic measurement of the system inefficiencies. $\dot{E}_{L,k}$ indicates the loss exergy within the system due to heat loss and is shown as:

$$\dot{E}_{L,k} = \dot{Q}_{loss}\left(1 - \frac{T_{surr}}{T}\right) \qquad (39)$$

If all the boundaries of a system is set to reference conditions, as in this study, then the term of $\dot{E}_{L,k}$ can be eliminated.

The exergetic efficiency is a common measure for assesing the thermodynamic perfomance of a system component in an industrial plant and is defined as:

$$\varepsilon = \frac{\dot{E}_{P,k}}{\dot{E}_{F,k}} = 1 - \frac{\dot{E}_{D,k}}{\dot{E}_{F,k}} = \frac{1}{1+\left(\frac{\dot{E}_{D,k}}{\dot{E}_{P,k}}\right)} \qquad (40)$$

As seen above, defining the “fuel” and the “product” streams in a unit is a crucial point in the calculation of exergetic efficiency.

Using Bejan’s form (1996), the exergetic efficiency of the furnace with reference to Figure 4.1. can be written as:

$$\varepsilon_{Furnace}\,(\%) = \frac{Ex_3 + \Delta Ex_{feed}}{Ex_1 + Ex_2} \times 100 \qquad (41)$$

In this equation, $\Delta Ex_{feed}$ is the exergy change of feed stream (hydrocarbon mixture) and calculated by using Eq.42:

$$\Delta Ex_{feed} = (h_{vapor} - h_{liquid}) - T_0(S_{vapor} - S_{liquid}) \qquad (42)$$

The conventional exergetic efficiency of APH unit, however is defined as:

$$\varepsilon_{APH}\,(\%) = \frac{Ex_2 - Ex_5}{Ex_4 - Ex_3} \times 100 \qquad (43)$$

For waste heat boiler, exergetic efficiency of the unit is expressed as:

$$\varepsilon_{WHB} = \frac{Ex_9 - Ex_8}{Ex_4 - Ex_7} \times 100 \qquad (44)$$

## 4.7. Advanced Exergy Analysis of the System

In advanced exergy analysis, the destruction of the exergy is split into two parts:

$$\dot{E}_{D,k} = \dot{E}_{D,k}^{AV} + \dot{E}_{D,k}^{UN} \qquad (45)$$

$\dot{E}_{D,k}^{UN}$ is the unavoidable part of the exergy destruction in k-th component which cannot be eliminated even if ideal conditions (highest efficiency with minimum losses)  are provided for the component with the best technology available. Whereas, $\dot{E}_{D,k}^{AV}$ respresents the avoidable exergy destruction which is found by substracting the unavoidable part of destruction from the total actual exergy destruction.

To split the total exergy destruction into avoidable and unavoidable parts, unavoidable conditions should be well identified. Exergy rates of fuel and product of the unit change when unavoidable (best) conditions are taken into account. The unavoidable conditions refer to best and unapproachable working conditions associated with the technical and economics limits of today’s technology. According to process specifications, the unavoidable conditions considered in this study was summarized in Table 5.18 and inTable 5.36 in section of Results.

Once unavoidable conditions have been defined, unvoidable exergy destruction of *k-th* component in a system, with an actual product exergy, $\dot{Ex}_{P,k}$ can be calculated as:

$$\dot{Ex}_{D,k}^{UN} = \dot{Ex}_{P,k}\left(\frac{\dot{Ex}_{D,k}}{\dot{Ex}_{P,best,k}}\right)^{UN} \qquad (46)$$

where the ratio $\left(\frac{\dot{Ex}_{D,k}}{\dot{Ex}_{P,best,k}}\right)^{UN}$ represents the unavoidable exergy destruction per unit of product exergy.

In order to compute unavoidable exergy destruction rates, each unit should be considered as in isolation and separated from the system. In this manner, for the three units, unavoidable and avoidable exergy destructions were calculated using equation above and are given in Tables 5.25, 5.32 and 5.39 in section of Results.

Since exergetic efficiency ($\varepsilon$) is a measure of improvement potential and some exergy destruction can not be avoided, a new value of efficiency is called as modified exergetic efficiency ($\varepsilon^{*}$) and defined as:

$$\varepsilon^{*} = \frac{\dot{Ex}_{P,k}}{\dot{Ex}_{F,k} - \dot{Ex}_{D,k}^{AV}} = \frac{1}{1 + \left(\frac{\dot{Ex}_{D,k}}{\dot{Ex}_{P,k}}\right)^{UN}} \tag{47}$$

# 5. RESULTS

Using the actual operational data given below, enhancement of thermal energy use and exergy analyses were performed and results are given in the following tables Table 5.1-5.39.

**Table 5.1.**Operating data of the system

| **Stream No** | $\dot{m}(kg/h)$ | **T(°C)** | **P** $(bar)$ |
|---|---|---|---|
| **1-Natural Gas** | 3960 | 20 | 1.6 |
| **2-Air** | 61602 | 195 | 1.008 |
| **3-Combustion Gas** | 65562 | 408 | 1.0004 |
| **4-Combustion Gas** | 65562 | 259 | 1.0017 |
| **5-Air** | 61602 | 12 | 1.026 |
| **6-Aromatics** | 998700 | 272 | 1.078 |

## 5.1. Part 1- Improving thermal efficiency of aromatic furnace

### 5.1.1 Furnace Results

**Table 5.2.** Results obtained from furnace energy balance

| | |
|---|---|
| Adiabatic Flame Temperature (K) | 1,249 |
| Furnace Heat Load (Gcal/h) | 42.12 |
| Furnace Thermal Efficiency (Without APH, %) | 77.7 |
| Furnace Thermal Efficiency (With APH, %) | 82.67 |

### 5.1.2. Air Preheater Results

**Table 5.3.** Energy balance results of available APH

| | |
|---|---|
| Heat load of APH (Gcal/h) | 2.6 |
| Heat Transfer Area ($m^2$) | 614 |
| Overall Heat Transer Coefficient, U ($W/m^2K$) | 24.4 |
| Logaritmic Mean Temperature, $\Delta T_{lm}$(K) | 209 |

**Table 5.4.** Matlab simulation results of APH

| $T_{air,o}$ (°C) | $T_{cg,o}$ (°C) | $A_{new}$ ($m^2$) | $A_{increase}$ (%) | Fuel saving (%) | New furnace efficiency(%) |
|---|---|---|---|---|---|
| **195** | 256 | 614 | 0 | 0 | 82.67 |
| **205** | 248 | 686 | 12 | 0.32 | 82.93 |
| **215** | 243 | 771 | 26 | 0.63 | 83.2 |
| **225** | 235 | 870 | 42 | 0.95 | 83.47 |
| **235** | 227 | 990 | 61 | 1.26 | 83.74 |
| **245** | 219 | 1140 | 88 | 1.58 | 84.01 |
| **255** | 211 | 1340 | 118 | 1.89 | 84.28 |

## 5.1.3. Waste Heat Boiler Results

**Table 5.5.** Matlab simulation results of WHB at 2 bar

| $P = 2\ bar$ ($T_{sat} = 120.38$ ℃) | | | | |
|---|---|---|---|---|
| $T_{CG,o}$ (°C) | $\dot{m}_{steam}$ (kg/h) | $U$ ($W/m^2K$) | $A_{boiler}$ ($m^2$) | $N_{tube}$ (-) |
| **249** | 275.03 | 45.51 | 18.94 | 121 |
| **239** | 550.06 | 76.13 | 25.51 | 162 |
| **229** | 825.09 | 101.73 | 32.12 | 204 |
| **219** | 1100.13 | 124.12 | 39.41 | 251 |
| **209** | 1375.25 | 144.23 | 47.75 | 304 |
| **199** | 1650.24 | 162.42 | 57.61 | 367 |
| **189** | 1925.23 | 179.14 | 69.71 | 443 |
| **179** | 2200.29 | 194.50 | 85.16 | 541 |
| **169** | 2475.36 | 208.97 | 105.31 | 670 |
| **159** | 2750.30 | 222.44 | 133.67 | 850 |
| **149** | 3025.34 | 235.01 | 176.20 | 1121 |

**Table 5.6.** Matlab simulation results of WHB at 2.5 bar

| $\boldsymbol{P = 2.5\ bar\ (T_{sat} = 127.58}$ ℃) | | | | |
|---|---|---|---|---|
| $T_{CG,o}$ **(°C)** | $\dot{m}_{steam}$ **(kg/h)** | $U$ **(W/m$^2$K)** | $A_{boiler}$ **(m$^2$)** | $N_{tube}$ **(-)** |
| **249** | 273.75 | 45.35 | 20.00 | 127 |
| **239** | 547.50 | 75.88 | 27.01 | 172 |
| **229** | 821.25 | 101.40 | 34.11 | 217 |
| **219** | 1095.38 | 123.70 | 42.04 | 267 |
| **209** | 1368.82 | 143.77 | 51.20 | 325 |
| **199** | 1642.53 | 161.92 | 62.29 | 396 |
| **189** | 1986.32 | 178.65 | 76.01 | 484 |
| **179** | 2190.00 | 194.08 | 94.06 | 598 |
| **169** | 2463.80 | 208.45 | 118.54 | 754 |
| **159** | 2737.55 | 221.88 | 154.50 | 984 |
| **149** | 3011.30 | 234.46 | 213.55 | 1359 |

**Table 5.7**.Matlab simulation results of WHB at 3 bar

| $\boldsymbol{P = 3\ bar\ (T_{sat} = 133.69}$ ℃) | | | | |
|---|---|---|---|---|
| $T_{CG,o}$ **(°C)** | $\dot{m}_{steam}$ **(kg/h)** | $U$ **(W/m$^2$K)** | $A_{boiler}$ **(m$^2$)** | $N_{tube}$ **(-)** |
| **249** | 272.68 | 45.22 | 20.98 | 134 |
| **239** | 545.36 | 75.67 | 28.41 | 181 |
| **229** | 818.04 | 101.13 | 36.00 | 229 |
| **219** | 1090.70 | 123.40 | 44.50 | 283 |
| **209** | 1363.45 | 143.44 | 54.61 | 347 |
| **199** | 1636.13 | 161.51 | 66.87 | 425 |
| **189** | 1908.82 | 178.13 | 82.54 | 525 |
| **179** | 2181.52 | 193.57 | 103.36 | 658 |
| **169** | 2454.16 | 207.92 | 132.93 | 846 |
| **159** | 2726.85 | 221.38 | 179.04 | 1139 |
| **149** | 2999.50 | 233.96 | 263.14 | 1675 |

**Table 5.8.**Matlab simulation results of WHB at 3.5 bar

| $\boldsymbol{P = 3.5\ bar\ (T_{sat} = 139.03}$ ℃) | | | | |
|---|---|---|---|---|
| $\boldsymbol{T_{CG,o}}$ **(°C)** | $\boldsymbol{\dot{m}_{steam}}$ **(kg/h)** | $\boldsymbol{U}$ **(W/m$^2$K)** | $\boldsymbol{A_{boiler}}$ **(m$^2$)** | $\boldsymbol{N_{tube}}$ **(-)** |
| **249** | 271.75 | 45.11 | 21.92 | 140 |
| **239** | 543.51 | 75.49 | 29.75 | 189 |
| **229** | 815.26 | 100.90 | 37.82 | 241 |
| **219** | 1087.02 | 123.10 | 47.03 | 299 |
| **209** | 1358.85 | 143.12 | 57.98 | 368 |
| **199** | 1630.56 | 161.22 | 71.59 | 455 |
| **189** | 1902.37 | 177.81 | 89.14 | 567 |
| **179** | 2174.04 | 193.16 | 113.34 | 721 |
| **169** | 2445.81 | 207.56 | 149.17 | 949 |
| **159** | 2717.66 | 220.92 | 208.91 | 1330 |
| **149** | 2989.30 | 233.47 | 335.82 | 2137 |

**Table 5.9.** Matlab simulation results of WHB at 4 bar

| $\boldsymbol{P = 4\ bar\ (T_{sat} = 143.77}$ ℃) | | | | |
|---|---|---|---|---|
| $\boldsymbol{T_{CG,o}}$ **(°C)** | $\boldsymbol{\dot{m}_{steam}}$ **(kg/h)** | $\boldsymbol{U}$ **(W/m$^2$K)** | $\boldsymbol{A_{boiler}}$ **(m$^2$)** | $\boldsymbol{N_{tube}}$ **(-)** |
| **249** | 270.93 | 45.01 | 22.82 | 145 |
| **239** | 541.87 | 75.33 | 31.04 | 198 |
| **229** | 812.81 | 100.69 | 39.61 | 252 |
| **219** | 1083.08 | 122.9 | 49.40 | 315 |
| **209** | 1354.78 | 142.86 | 61.22 | 399 |
| **199** | 1625.61 | 160.89 | 76.23 | 485 |
| **189** | 1896.61 | 177.40 | 96.03 | 611 |
| **179** | 2167.54 | 192.83 | 124.11 | 790 |
| **169** | 2438.53 | 207.15 | 167.72 | 1068 |
| **159** | 2709.43 | 220.54 | 247.32 | 1574 |
| **149** | 2980.30 | 233.03 | 463.20 | 2948 |

## 5.2. Part 2-Conventional and Advanced Exergy Analyses

### 5.2.1. Results of Conventional Exergy Analysis of Available System

**Table 5.10.**Entalphy values of process streams

| **Stream number** | **Entalphy of Compounds** | | | | | **Mixture Enthalpy h, $\left[\frac{kJ}{kmol}\right]$** | **Mixture Standart Enthalpy, $h^0$, $\left[\frac{kJ}{kmol}\right]$** |
|---|---|---|---|---|---|---|---|
| | $N_2$(g) | $O_2$(g) | $CO_2$(g) | $H_2O$(g) | $CH_4$(g) | | |
| **1** | 0 | 0 | 0 | 0 | -74872 | -74872 | -74872 |
| **2** | 5040 | 5122 | -386341 | -236020 | 0 | 4940 | -118 |
| **3** | 11671 | 11928 | -375899 | -228021 | 0 | -37491 | -49762 |
| **4** | 6914 | 7036 | -383452 | -233788 | 0 | -42530 | -49762 |
| **5** | -375 | -380 | -393983 | -242270 | 0 | -494 | -118 |

**Table 5.11.**Entropy values of process streams

| **Stream number** | **Entropy of Compounds** | | | | | **Entropy of Mixture, s, $\left[\frac{kJ}{kmolK}\right]$** | **Mixture Standart Entropy, $s^0$, $\left[\frac{kJ}{kmolK}\right]$** |
|---|---|---|---|---|---|---|---|
| | $N_2$(g) | $O_2$(g) | $CO_2$(g) | $H_2O$(g) | $CH_4$(g) | | |
| **1** | 0 | 0 | 0 | 0 | 186.0 | 186.0 | 186.0 |
| **2** | 205.0 | 218.7 | 232.7 | 204.2 | 0 | 207.9 | 194 |
| **3** | 216.6 | 230.7 | 251.0 | 218.3 | 0 | 219.9 | 194 |
| **4** | 208.7 | 222.6 | 238.5 | 208.7 | 0 | 211.5 | 194 |
| **5** | 190.3 | 203.9 | 212.2 | 187.4 | 0 | 193.2 | 194 |

**Table 5.12.** Physical exergy rates of streams ($Ex^{PH}$)

| **Stream No** | **Molar Flow** $\left[\frac{\mathbf{kmol}}{\mathbf{h}}\right]$ | **Entalphy** $\left[\frac{\mathbf{kJ}}{\mathbf{kmol}}\right]$ | **Entropy** $\left[\frac{\mathbf{kJ}}{\mathbf{kmol\ K}}\right]$ | **Ref. Entalphy** $\left[\frac{\mathbf{kJ}}{\mathbf{kmol}}\right]$ | **Ref. Entropy** $\left[\frac{\mathbf{kJ}}{\mathbf{kmol\ K}}\right]$ | $\mathbf{Ex^{PH}}$**,** [**MW**] |
|---|---|---|---|---|---|---|
| **1** | 236 | -74,872 | 186 | -74,872 | 186 | 0.000 |
| **2** | 2,124 | 4,940 | 208 | -118 | 194 | 0.630 |
| **3** | 2,333 | -37,491 | 220 | -49,762 | 194 | 2.880 |
| **4** | 2,333 | -42,530 | 212 | -49,762 | 194 | 1.230 |
| **5** | 2,124 | -494 | 193 | -118 | 194 | 0.003 |

**Table 5.13**.Standart molar chemical exergies of the components ($Ex^{CH}$)

| **Component** | **Standart Molar Chemical Exergy,** [**kJ/kmol**] |
|---|---|
| **$N_2$ (g)** | 639 |
| **$O_2$ (g)** | 3,951 |
| **$CO_2$ (g)** | 14,176 |
| **$H_2O$ (g)** | 8,636 |
| **$CH_4$(g)** | 824,348 |

**Table 5. 14.**Chemical exergy rates of process streams ($\dot{m}\ Ex^{CH}$)

| **Stream Number** | **Chemical Exergy Rates,** [$\boldsymbol{MW}$] |
|---|---|
| **1** | 54.168 |
| **2** | 0.035 |
| **3** | 0.770 |
| **4** | 0.770 |
| **5** | 0.034 |

**Table 5.15.**Exergy change of feed stream (hydrocarbon mixture) ($\Delta$Ex)

| **Stream Number** | **Enthalpy of vaporization,** $[\mathbf{kJ/kmol}]$ | **Liquid Phase Entropy,** $[\mathbf{kJ/kmol\,K}]$ | **Vapor Phase Entropy,** $[\mathbf{kJ/kmolK}]$ | **Exergy Change of Vaporization** $\mathbf{\Delta Ex}$**,**$[\mathbf{MW}]$ |
|---|---|---|---|---|
| **6** | 33,900 | 269.0 | 173.2 | 19.05 |

**Table 5.16.**Total exergy rates of the streams

| **Stream Number** | **Physical Exergy,** $\mathbf{E^{PH}}$**,**$[MW]$ | **Chemical Exergy,** $\mathbf{E^{CH}}$**,**$[MW]$ | **Total Exergy** $[MW]$ |
|---|---|---|---|
| **1** | 0.000 | 54.170 | 54.169 |
| **2** | 0.630 | 0.035 | 0.660 |
| **3** | 2.880 | 0.770 | 3.650 |
| **4** | 1.230 | 0.770 | 2.000 |
| **5** | 0.003 | 0.034 | 0.037 |

**Table 5.17.**Conventional exergetic efficiencies of process units, ε (%)

| **Unit Name** | **Exergy of Fuel** $[MW]$ | **Exergy of Product** $[MW]$ | **Exergy Destruction** $[MW]$ | **Efficiency (%)** |
|---|---|---|---|---|
| **Furnace** | 54.83 | 22.74 | 32.09 | 41.4 |
| **Air Preheater** | 1.65 | 0.62 | 1.02 | 37.0 |

### 5.2.2. Results of Advanced Exergy Analysis of Available System

**Table 5.18.** Real and best conditions of the units

| **Unit Name** | **Real Conditions** | **Best Conditions** |
|---|---|---|
| **FH** | $Q \neq 0$<br>$T_{fuel}$= 20°C<br>$T_{air}$= 12 °C<br>$T_{flue\ gas}$= 408 °C | $Q = 0$<br>$T_{fuel}$= 577°C<br>$T_{air}$= 527 °C<br>$T_{flue\ gas}$= 957 °C |
| **APH** | $\Delta T_{min}$= 213°C | $\Delta T_{min}$= 10°C |

### 5.2.2.1. Results of advanced exergy analysis of furnace

**Table 5.19.** Operating data of furnace with **best** conditions

| Stream No | Flow Rate $(\mathrm{kg/h})$ | Temperature $(\mathrm{K})$ | Pressure $(\mathrm{bar})$ |
|---|---|---|---|
| **1-Natural Gas** | 3,960 | 850 | 1.6000 |
| **2-Air** | 61,602 | 800 | 1.0080 |
| **3-Combustion Gas** | 65,562 | 1,250 | 1.0004 |

**Table 5.20.**Entalphy values of furnace streams at **best** working conditions

| Stream number | Entalphy of Compounds | | | | | Mixture Enthalpy $h, \left[\frac{kJ}{kmol}\right]$ | Mixture Standart Entalphy $h^0, \left[\frac{kJ}{kmol}\right]$ |
|---|---|---|---|---|---|---|---|
| | $N_2(g)$ | $O_2(g)$ | $CO_2(g)$ | $H_2O(g)$ | $CH_4(g)$ | | |
| **1** | 0 | 0 | 0 | 0 | -46,985 | -46,985 | -74,872 |
| **2** | 15,463 | 15,861 | -369,752 | -223,332 | 0 | 15,431 | -118 |
| **3** | 29,547 | 30,683 | -346,506 | -205,313 | 0 | -18,348 | -49,762 |

**Table 5.21.** Entropy values of furnace streams at **best** working conditions

| Stream number | Entropy of Compounds | | | | | Mixture Entropy $s, \left[\frac{kJ}{kmolK}\right]$ | Mixture Standart Entropy $s^0, \left[\frac{kJ}{kmolK}\right]$ |
|---|---|---|---|---|---|---|---|
| | $N_2(g)$ | $O_2(g)$ | $CO_2(g)$ | $H_2O(g)$ | $CH_4(g)$ | | |
| **1** | 0 | 0 | 0 | 0 | 236.5 | 236.5 | 186.0 |
| **2** | 221.8 | 220.4 | 259.3 | 224.6 | 0 | 221.5 | 194 |
| **3** | 235.8 | 250.8 | 282.5 | 242.6 | 0 | 240.4 | 194 |

**1Table 5.22.** Physical exergy rates of furnace streams at **best** conditions

| **Stream No** | **Molar Flow** $[\mathbf{kmol/h}]$ | **Entalphy** $[\mathbf{kJ/kmol}]$ | **Entropy** $[\mathbf{kJ/kmol\,K}]$ | **Ref. Entalphy** $[\mathbf{kJ/kmol}]$ | **Ref. Entropy** $[\mathbf{kJ/kmolK}]$ | $\mathbf{Ex^{PH}}$, $[\mathbf{MW}]$ |
|---|---|---|---|---|---|---|
| **1** | 236.6 | -46,985 | 236.5 | -74,872 | 186 | 0.86 |
| **2** | 2124.2 | 15,431 | 221.5 | -118 | 194 | 4.42 |
| **3** | 2333.0 | -18,348 | 240.4 | -49,762 | 194 | 11.32 |

**Table 5.23.** Chemical exergy rates of furnace streams

| **Stream Number** | **Chemical Exergy Rates,** $[\mathbf{MW}]$ |
|---|---|
| **1** | 54.168 |
| **2** | 0.035 |
| **3** | 0.770 |

**Table 5.24.**Total exergy rates of furnace streams at **best** conditions

| **Stream Number** | **Physical Exergy,** $\mathbf{E^{PH}}$,[MW] | **Chemical Exergy,** $\mathbf{E^{CH}}$,[MW] | **Total Exergy** $[\mathbf{MW}]$ |
|---|---|---|---|
| **1** | 0.86 | 54.17 | 55.03 |
| **2** | 4.42 | 0.03 | 4.46 |
| **3** | 11.32 | 0.77 | 12.09 |

**Table 5.25.** Modified exergetic efficiency of the furnace

| **Unit Name** | $\dot{\mathbf{Ex}}_{\mathbf{D,k}}^{\mathbf{AV}}$ $[\mathbf{MW}]$ | $\dot{\mathbf{Ex}}_{\mathbf{D,k}}^{\mathbf{UN}}$ $[\mathbf{MW}]$ | $\boldsymbol{\varepsilon}^{*}$ **(%)** |
|---|---|---|---|
| **Furnace** | 20.63 | 11.45 | 52.4 |

#### 5.2.2.2. Results of advanced exergy analysis of air preheater

**Table 5. 26.** Operating data of air preheater with **best** conditions

| Stream No | Flow Rate $[\mathrm{kg/h}]$ | Temperature $[\mathrm{K}]$ | Pressure $[\mathrm{bar}]$ |
|---|---|---|---|
| **2-Air** | 61,602 | 671 | 1.0080 |
| **3-Combustion Gas** | 65,562 | 681 | 1.0004 |
| **4-Combustion Gas** | 65,562 | 318 | 1.0017 |
| **5-Air** | 61,602 | 285 | 1.0260 |

**Table 5. 27.** Entalphy values of air preheater streams at **best** working conditions

| Stream number | Entalphy of Compounds | | | | | Mixture Enthalpy $\mathbf{h}, \left[\frac{\mathbf{kJ}}{\mathbf{kmol}}\right]$ | Mixture Standart Entalphy $\mathbf{h^0}, \left[\frac{\mathbf{kJ}}{\mathbf{kmol}}\right]$ |
|---|---|---|---|---|---|---|---|
| | $N_2(g)$ | $O_2(g)$ | $CO_2(g)$ | $H_2O(g)$ | $CH_4(g)$ | | |
| **2** | 11355 | 11602 | -376408 | -228408 | 0 | 11290 | -118 |
| **3** | 11671 | 11928 | -375900 | -228021 | 0 | -37491 | -49762 |
| **4** | 568 | 577 | -392787 | -241212 | 0 | -49175 | -49762 |
| **5** | -375 | -380 | -393983 | -242270 | 0 | -494 | -118 |

**Table 5.28.** Entropy values of air preheater streams at **best** working conditions

| Stream number | Entropy of Compounds | | | | | Mixture Entropy $\mathbf{s}, \left[\frac{\mathbf{kJ}}{\mathbf{kmolK}}\right]$ | Mixture Standart Entropy $\mathbf{s^0}, \left[\frac{\mathbf{kJ}}{\mathbf{kmolK}}\right]$ |
|---|---|---|---|---|---|---|---|
| | $N_2(g)$ | $O_2(g)$ | $CO_2(g)$ | $H_2O(g)$ | $CH_4(g)$ | | |
| **2** | 216.2 | 230.2 | 250.3 | 217.7 | 0 | 219.1 | 194 |
| **3** | 216.6 | 230.7 | 251.0 | 218.3 | 0 | 219.9 | 194 |
| **4** | 193.5 | 207.1 | 216.2 | 190.9 | 0 | 195.5 | 194 |
| **5** | 190.3 | 203.9 | 212.2 | 187.4 | 0 | 193.2 | 194 |

**Table 5.29.** Physical exergy rates of air preheater streams at **best** conditions

| Stream No | Molar Flow $[\mathrm{kmol/h}]$ | Entalphy $[\mathrm{kJ/kmol}]$ | Entropy $[\mathrm{kJ/kmol\,K}]$ | Ref. Entalphy $[\mathrm{kJ/kmol}]$ | Ref. Entropy $[\mathrm{kJ/kmolK}]$ | $\mathrm{Ex^{PH}}$, [MW] |
|---|---|---|---|---|---|---|
| **2** | 2,124 | 11,290 | 219.1 | -118 | 194 | 2.400 |
| **3** | 2,333 | -37,491 | 219.9 | -49,762 | 194 | 2.880 |
| **4** | 2,333 | -49,175 | 195.5 | -49,762 | 194 | 0.011 |
| **5** | 2,124 | -494 | 193.2 | -118 | 194 | 0.002 |

**Table 5.30.** Chemical exergy rates of air preheater streams

| Stream Number | Chemical Exergy Rates, $[MW]$ |
|---|---|
| **2** | 0.035 |
| **3** | 0.770 |
| **4** | 0.770 |
| **5** | 0.035 |

**Table 5.31.** Total exergy rates of air preheater streams at **best** conditions

| Stream Number | Physical Exergy, $\mathrm{E^{PH}}$,$[MW]$ | Chemical Exergy, $\mathrm{E^{CH}}$,$[MW]$ | Total Exergy $[MW]$ |
|---|---|---|---|
| **2** | 2.400 | 0.035 | 2.434 |
| **3** | 2.880 | 0.77 | 3.650 |
| **4** | 0.010 | 0.770 | 0.780 |
| **5** | 0.003 | 0.035 | 0.037 |

**Table 5.32.** Modified exergetic efficiency of the air preheater

| Unit Name | $\dot{\mathrm{E}}\mathrm{x}_{\mathrm{D,k}}^{\mathrm{AV}}$ $[MW]$ | $\dot{\mathrm{E}}\mathrm{x}_{\mathrm{D,k}}^{\mathrm{UN}}$ $[MW]$ | $\varepsilon^{*}$ (%) |
|---|---|---|---|
| **Air Preheater** | 0.9 | 0.12 | 83.5 |

### 5.2.3. Results of Conventional Exergy Analysis of Alternative Waste Heat Boiler

Exergy rate of the streams and the exergetic efficiency of the waste heat boiler unit were calculated by considering different outlet temperatures of the combustion gas (Stream 7) varying from 249 C to 149 C and different pressures of produced steam (Stream 9) varying from 2 bar to 4 bar.

**Table 5.33.** Physical and chemical exergy rates of the streams in waste heat boiler

| **$T_7$(°C)** | **$P_9$(bar)** | $E^{PH}$,$[MW]$ | | | | $E^{CH}$,$[MW]$ | | | |
|---|---|---|---|---|---|---|---|---|---|
| | | **Stream** | | | | **Stream** | | | |
| | | **4** | **7** | **8** | **9** | **4** | **7** | **8** | **9** |
| **249** | 2 | 1.24 | 1.15 | 0.0020 | 0 | 0.719 | 0.719 | 0.001 | 0.037 |
| | 2.5 | 1.24 | 1.15 | 0.0023 | 0 | 0.719 | 0.719 | 0.001 | 0.037 |
| | 3 | 1.24 | 1.15 | 0.0025 | 0 | 0.719 | 0.719 | 0.001 | 0.037 |
| | 3.5 | 1.24 | 1.15 | 0.0027 | 0 | 0.719 | 0.719 | 0.001 | 0.036 |
| | 4 | 1.24 | 1.15 | 0.0028 | 0 | 0.719 | 0.719 | 0.001 | 0.036 |
| **229** | 2 | 1.24 | 0.99 | 0.0055 | 0 | 0.719 | 0.719 | 0.002 | 0.111 |
| | 2.5 | 1.24 | 0.99 | 0.0063 | 0 | 0.719 | 0.719 | 0.002 | 0.110 |
| | 3 | 1.24 | 0.99 | 0.0069 | 0 | 0.719 | 0.719 | 0.002 | 0.109 |
| | 3.5 | 1.24 | 0.99 | 0.0075 | 0 | 0.719 | 0.719 | 0.002 | 0.108 |
| | 4 | 1.24 | 0.99 | 0.0080 | 0 | 0.719 | 0.719 | 0.002 | 0.107 |
| **209** | 2 | 1.24 | 0.82 | 0.0092 | 0 | 0.719 | 0.719 | 0.004 | 0.190 |
| | 2.5 | 1.24 | 0.82 | 0.0104 | 0 | 0.719 | 0.719 | 0.004 | 0.184 |
| | 3 | 1.24 | 0.82 | 0.0115 | 0 | 0.719 | 0.719 | 0.004 | 0.182 |
| | 3.5 | 1.24 | 0.82 | 0.0125 | 0 | 0.719 | 0.719 | 0.004 | 0.180 |
| | 4 | 1.24 | 0.82 | 0.0134 | 0 | 0.719 | 0.719 | 0.004 | 0.179 |
| **189** | 2 | 1.24 | 0.64 | 0.0129 | 0 | 0.719 | 0.719 | 0.005 | 0.260 |
| | 2.5 | 1.24 | 0.64 | 0.0146 | 0 | 0.719 | 0.719 | 0.005 | 0.257 |
| | 3 | 1.24 | 0.64 | 0.0161 | 0 | 0.719 | 0.719 | 0.005 | 0.255 |
| | 3.5 | 1.24 | 0.64 | 0.0175 | 0 | 0.719 | 0.719 | 0.005 | 0.253 |
| | 4 | 1.24 | 0.64 | 0.0187 | 0 | 0.719 | 0.719 | 0.005 | 0.251 |
| **169** | 2 | 1.24 | 0.52 | 0.0167 | 0 | 0.719 | 0.719 | 0.007 | 0.334 |
| | 2.5 | 1.24 | 0.52 | 0.0188 | 0 | 0.719 | 0.719 | 0.007 | 0.330 |
| | 3 | 1.24 | 0.52 | 0.0207 | 0 | 0.719 | 0.719 | 0.007 | 0.327 |
| | 3.5 | 1.24 | 0.52 | 0.0225 | 0 | 0.719 | 0.719 | 0.007 | 0.325 |
| | 4 | 1.24 | 0.52 | 0.0240 | 0 | 0.719 | 0.719 | 0.007 | 0.322 |
| **149** | 2 | 1.24 | 0.39 | 0.0202 | 0 | 0.719 | 0.719 | 0.009 | 0.408 |
| | 2.5 | 1.24 | 0.39 | 0.0229 | 0 | 0.719 | 0.719 | 0.009 | 0.404 |
| | 3 | 1.24 | 0.39 | 0.0253 | 0 | 0.719 | 0.719 | 0.009 | 0.400 |
| | 3.5 | 1.24 | 0.39 | 0.0275 | 0 | 0.719 | 0.719 | 0.009 | 0.397 |
| | 4 | 1.24 | 0.39 | 0.0294 | 0 | 0.719 | 0.719 | 0.009 | 0.394 |

**Table 5.34.** Product, fuel and destruction exergy rates of waste heat boiler

| $T_7$(°C) | $P_9$(bar) | Exergy of Product,$[MW]$ | Exergy of Fuel,$[MW]$ | Exergy of Destruction,$[MW]$ |
|---|---|---|---|---|
| **249** | 2 | 0.0369 | 0.0906 | 0.0537 |
| | 2.5 | 0.0372 | 0.0906 | 0.0534 |
| | 3 | 0.0373 | 0.0906 | 0.0533 |
| | 3.5 | 0.0375 | 0.0906 | 0.0531 |
| | 4 | 0.0377 | 0.0906 | 0.0529 |
| **229** | 2 | 0.1145 | 0.2650 | 0.1505 |
| | 2.5 | 0.1143 | 0.2650 | 0.1507 |
| | 3 | 0.1139 | 0.2650 | 0.1511 |
| | 3.5 | 0.1135 | 0.2650 | 0.1515 |
| | 4 | 0.1130 | 0.2650 | 0.1520 |
| **209** | 2 | 0.1952 | 0.4283 | 0.2331 |
| | 2.5 | 0.1904 | 0.4283 | 0.2379 |
| | 3 | 0.1895 | 0.4283 | 0.2388 |
| | 3.5 | 0.1890 | 0.4283 | 0.2393 |
| | 4 | 0.1884 | 0.4283 | 0.2399 |
| **189** | 2 | 0.2679 | 0.6010 | 0.3331 |
| | 2.5 | 0.2666 | 0.6010 | 0.3344 |
| | 3 | 0.2661 | 0.6010 | 0.3349 |
| | 3.5 | 0.2655 | 0.6010 | 0.3355 |
| | 4 | 0.2647 | 0.6010 | 0.3363 |
| **169** | 2 | 0.3437 | 0.7192 | 0.3755 |
| | 2.5 | 0.3418 | 0.7192 | 0.3774 |
| | 3 | 0.3407 | 0.7192 | 0.3785 |
| | 3.5 | 0.3405 | 0.7192 | 0.3787 |
| | 4 | 0.3390 | 0.7192 | 0.3802 |
| **149** | 2 | 0.4192 | 0.8450 | 0.4258 |
| | 2.5 | 0.4179 | 0.8450 | 0.4271 |
| | 3 | 0.4163 | 0.8450 | 0.4287 |
| | 3.5 | 0.4155 | 0.8450 | 0.4295 |
| | 4 | 0.4144 | 0.8450 | 0.4306 |

**Table 5.35.** Conventional exergetic efficiency of waste heat boiler at given design conditions

| $P_9$ (bar) | ε (%) | | | | | |
|---|---|---|---|---|---|---|
| | $T_7$=249°C | $T_7$=229°C | $T_7$=209°C | $T_7$=189°C | $T_7$=169°C | $T_7$=149°C |
| 2 | 42.1 | 43.3 | 44.6 | 46.1 | 47.8 | 49.7 |
| 2.5 | 41.8 | 43.0 | 44.4 | 45.9 | 47.6 | 49.5 |
| 3 | 41.7 | 42.9 | 44.3 | 45.7 | 47.4 | 49.4 |
| 3.5 | 41.7 | 42.8 | 44.2 | 45.6 | 47.3 | 49.2 |
| 4 | 41.6 | 42.7 | 44.1 | 45.5 | 47.2 | 49.1 |

### 5.2.4. Results of Advanced Exergy Analysis of Waste Heat Boiler

**Table 5.36.** The **best** conditions of the unit

| Unit Name | Real Conditions | Best Conditions |
|---|---|---|
| **WHB** | $\Delta T_{min}$= 117°C | $\Delta T_{min}$= 50°C |

**Table 5.37.** Operating data of waste heat boiler with best conditions

| Stream No | Flow Rate $[\mathbf{kg/h}]$ | Temperature $[\mathbf{K}]$ | Pressure $[\mathbf{bar}]$ |
|---|---|---|---|
| **4-Combustion Gas** | 65,562 | 532 | 1.00 |
| **7-Combustion Gas** | 65,562 | 422 | 1.03 |
| **8-Water** | 16,520 | 368 | 1.02 |
| **9-Steam** | 4,130 | 417 | 4.00 |

**Table 5.38.** Total exergy rates of waste heat boiler streams at **best** conditions

| Stream Number | Physical Exergy, $\mathrm{E^{PH}}$,[MW] | Chemical Exergy, $\mathrm{E^{CH}}$,[MW] | Total Exergy [MW] |
|---|---|---|---|
| **4** | 0.0494 | 0.011 | 1.96 |
| **7** | 0.0407 | 0.550 | 1.12 |
| **8** | 1.2447 | 0.719 | 0.061 |
| **9** | 0.3997 | 0.719 | 0.59 |

**Table 5.39.** Modified exergetic efficiency of waste heat boiler

| Unit Name | $\dot{\mathrm{Ex}}_{\mathrm{D,k}}^{\mathrm{AV}}[MW]$ | $\dot{\mathrm{Ex}}_{\mathrm{D,k}}^{\mathrm{UN}}[MW]$ | $\varepsilon^{*}$(%) |
|---|---|---|---|
| **Waste Heat Boiler** | 0.176 | 0.158 | 62.8 |

# 6. DISCUSSION & CONCLUSION

## PART -1

### First Method: Increasing the heat transfer area of existing Air Preheater

Table 5.2 shows that, furnace requires 42.12 Gcal heat per hour to vaporize aromatic feed stream consists 95% of benzene and 5% of toluene and it operates with 77.7% of thermal energy efficiency without air preheater. The stack gas (combustion gas) that leaves the furnace at 408°C still carries lots of thermal energy, hence, an APH was placed next to the furnace by the company to decrease the combustion gas temperature and increase the temperature of combustion air going to furnace.

With existing APH, thermal efficiency of total system was increased up to 82.67% and the temperature of the combustion gas discarded to atmosphere was decreases to 259 °C (still hot!). This air preheater has a heat transfer area of 614 $m^2$ and overall heat transfer coefficienct of this unit was calculated indirectly as 24.4 $W/m^2K$. Since hot and cold fluids in the APH are both in gas phase, this low value of overall heat transfer coefficient was acceptable and consistent with literature values.

As the first method of the Part-1, increasing the heat transfer area of APH in the plant was investigated. By using MATLAB R2012-b, required percentage area increase and corresponding combustion gas temperature decrease were computed by increasing combustion air temperature that leaves the APH. As previously mentioned, if the combustion reaction in the furnace has taken place with higher air temperature than required amount of fuel that should be supplied to heat the feed would decrease. This is very important especially when the high prices of natural gas is taken into account.

It can clearly seen from energy balance in Appendix that, in terms of heat transfer there is no certain limits for decreasing the exit combustion gas temperature to 120-140°C. If these desired stack gas values are achieved, reduction in $CO_2$ and $NO_x$ emmissions will eventually decrease and environmental needs are satisfied.

To accomplish this, extra heat transfer area in air preheater must be placed to increase the heat load. However, MATLAB simulation results shows in Table 5.4

that, only %1.26 fuel saving can be achieved by 61% of area increase. At this point, stack gas leaving the APH decreases from 259 °C to 227°C, only while combustion air leaving the APH increases up to 235°C and total thermal energy efficiency of the system was increased up from 82.67% to just 83.74%. This “insufficient” situation arises from the low value of overall heat transfer coefficient (U) of APH. When U value in a heat exchanger is low, increase in required area should be very high to increase the heat load of the exchanger, remarkably.

**Second Method: Production of low-medium pressure steam with a waste heat boiler**

Since steam is a crucial resource in today’s industrial world, it can be more effective to produce steam from flue gas for the companies which need steams as a source of power for the production of electricity in the plant or heating purposes.

Placing a waste heat boiler next to APH, to produce low-medium pressure steam, because of relatively low temperature of combustion gases, using the thermal energy of combustion gas leaving the APH stack, was considered as second alternative.

In this second way, the overall heat transfer coefficient in boiler will be considerably higher (see Table 5.5-5.9) than that of in APH. The underlying reason is that, during the steam production, boiling takes place in the water side of the tubes and heat transfer coefficient of boiling fluids is very high.

A MATLAB program was written that gives the amount of steam produced at various pressures and required boiler area by decreasing the combustion air temperature to various values. It was seen in the tables through Tables 5.5-5.9 that relatively smaller heat transfer areas are required in this method to increase the total thermal efficiency of the system.

## Conclusion

In the first alternative, it was concluded that, a reasonable increase in heat transfer area of APH could not provide a remarkable fuel saving and a decrease in combustion gas temperature in the APH unit. From economical point of view, although increasing heat load of available APH may enhance the thermal performance of the system, due to preheater's high capital and maintanence costs this first method is not economically justifiable.

On the other hand, with second alternative system (waste heat boiler); it is observed that, pretty good remediations can be provided. If a waste heat boiler is placed after APH, exit temperature of stack gas can be reduced under 200 °C with relatively smaller areas.

## PART-2

### Conventional exergy analysis

In the second part of the study, exergy analysis was applied to the furnace-air preheater coupled system. Total exergy rates of streams were computed as the sum of physical and chemical exergy rates. As expected, the major exergy destruction was obtained in the furnace. Because, combustion reaction that takes place in furnace burner's is one of the biggest source of entropy generation and exergy destructions. Exergy destruction in the furnace is also affected by the excess air fraction and the temperature of the air at the inlet. The thermodynamic inefficiencies of combustion can be reduced by preheating the combustion air and avoiding from high air–fuel ratio.

On the other hand, exergetic efficiency of APH was calculated as 37% with an exergy destruction of 1.02 MW. The significant part of exergy destruction within a heat exchanger arises from heat transfer with finite temperature difference. Increasing temperature difference of fluids results in high entropy generation and exergy destruction in accordance with Gouy-Stodola theorem $(Ex_d = T_{surr} S_{gen})$.

Exergy analysis of WHB, however, was performed for different cases. In each cases, the outlet temperature of combustion gases leaving from the boiler and produced steam pressure were varied, hence, amount of steam produced and the temperature of the steam were changed. Thus, for each cases calculated exergetic efficiency changed as in Table 5.35. In compliance with these results, in the further calculations a reasonable operating conditions for waste heat boiler were chosen to be used in advanced exergetic analysis. According to chosen data, combustion gas temperature leaving from the boiler was 189 °C and 1908.8 kg/h steam at 3 bar was being produced in this unit.

### Advanced exergy analysis

In advanced exergy analysis, the destruction of the exergy was split into avoidable and unavoidable parts. To calculate the amount of unavoidable exergy destruction, firstly unavoidable conditions of process units were well-identified and shown in Table 5.19.

As shown in the table, high reactant and product temperatures and adiabatic combustion were stated as unavoidable (best) conditions in the furnace. At the same time, for heat exchangers (APH, WHB) entropy generation, hence exergy destruction, can be minimized by assuming no pressure drop, no heat loss and temperature differences as low as possible. These are also the unavoidable conditions for APH unit in this study.

According to these specifications, avoidable and unavoidable exergy destructions of the two units were calculated and modified exergetic efficiency values were found as 52.4% , 83.5%, 62.8% for furnace, air preheater and waste heat boiler respectively.

## Conclusion

The energy and the exergy values of streams were calculated with the aid of mass, energy and exergy balances for overall system and for each unit of the system and exergetic efficiencies of each unit were determined by both conventional and advanced exergy analysis techniques.

It can be seen in the Table 5.18 that exergy destruction in the furnace (32.09 MW) was much higher than 1.02 MW in APH. Almost 97% of total exergy destruction arises from furnace irreversibilities, because entropy generation during combustion is very high and it results in high irreversibilities. It was determined by advanced exergy analysis that 64% of this destruction can be avoided by improving working conditions.

In APH, it was found that 87% of total exergy destruction can be avoided and exergetic efficiency can be increased from 37% to 83.5% with suitable modifications and only the part of the irreversibilities, that can be avoided, should be taken into consideration for increasing the potential improvement of the system.

Since 97% of total exergy destruction occurs in the furnace, enhancement of the system from exergy point of view should be especially focused on furnace. The main reason for high entropy production is to heat aromatic feed stream with high temperature difference. It is inevitable (this is for this reason high combustion gas temperature and eventually high amount of fuel required) to transfer fixed amount of heat for vaporization of aromatics

Here, as third alternative method, we can recommend to increase heat transfer area of the tubes in radiant and convection sections of the furnace. Therefore, to transfer the same amount of heat, less temperature difference and and as a result of lower temperature combustion gases will be sufficient. This will lead us less amount of fuel consumed, finally.

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## RESUME

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2. S.Sert, F.Balkan, Effect of Some Parameters on Exergetic Efficiency of a Combustion System in a Petrochemical Plant, 1st International Conference on Engineering Technology and Applied Sciences, ICETAS'2016, 21-22 April, 2016, Afyonkarahisar, Türkiye.

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# APPENDICES

## Natural gas heat of cumbustion calculation

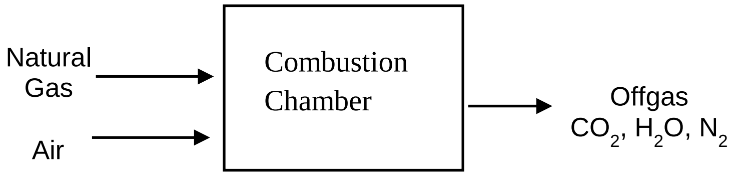


**Table 7.1.** Natural gas properties

| Species | Molecular Mass | Volume [%] | Mass [%] | Mass [g] | $\Delta H_{comb}$ [kJ/mol] |
|---|---|---|---|---|---|
| **Methane** | 16.04 | 0.97 | 0.91 | 15.24 | -762.21 |
| **Ethane** | 30.07 | 0.03 | 0.09 | 1.50 | -71.39 |

$$\Delta H_{comb_{NG}} = x_{CH_4}\left(\Delta H_{comb_{CH_4}}\right) + x_{C_2H_6}\left(\Delta H_{comb_{C_2H_6}}\right)=$$

833.6 kJ/mol=50001 kJ/kg

## Calculation of the heat load of aromatic feed

## Energy balance around the furnace

$$\dot{m}_{NG}Cp_{NG}\left(T_{NG,in} - T_{ref}\right) + \dot{m}_{Air}Cp_{Air}\left(T_{Air,in} - T_{ref}\right) + \dot{m}_{NG}\Delta H_{comb} - \dot{m}_{CG}Cp_{CG}\left(T_{CG,out} - T_{ref}\right) - \dot{m}_{feed}\Delta H_{feed} = 0$$

$$\Delta H_{feed} = \frac{(3960 \times 2.2 \times 293) + (61602 \times 1.032 \times 468) + (3960 \times 50001) - (65562 \times 1.15 \times 681)}{998700} = 176.1 \times 10^6\ kJ/h = 42.1\ \ Gcal/h$$

## Energy balance around APH

$$\dot{Q}_{APH} = \dot{Q}_{air} = \ \dot{Q}_{CG}$$

$$\dot{Q}_{APH} = 65562 \times 1.1 \times (408 - 259) = 61602 \times 1.02 \times (195 - 12) = 11.3\, GJ/h = 2.6\ \ Gcal/h$$

$$\dot{m}_{CG}Cp_{CG}\left(T_{CG,out} - T_{CG,in}\right) = \dot{m}_{air}Cp_{air}\left(T_{air,out} - T_{air,in}\right) = UA_{APH}\Delta T_{lm}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

$$\Delta T_1 = 408 - 195 \; \& \; \Delta T_2 = 259 - 12$$

$$\Delta T_{lm} = \frac{213 - 247}{ln\left(\frac{213}{247}\right)} = 228\,°C = 502K$$

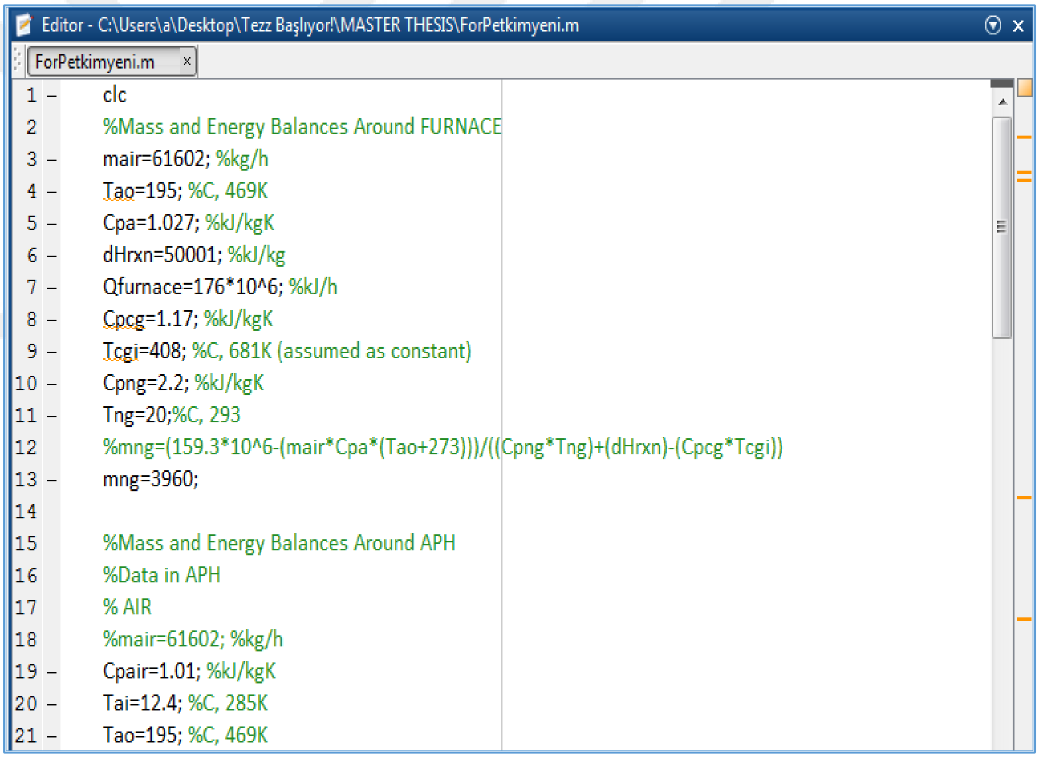
Cross flow correction factor is read as 0.93 from Figure 4.3.

$$\Delta T_{lm_{cross-flow}} = 0.92 \times 228 = 209K$$

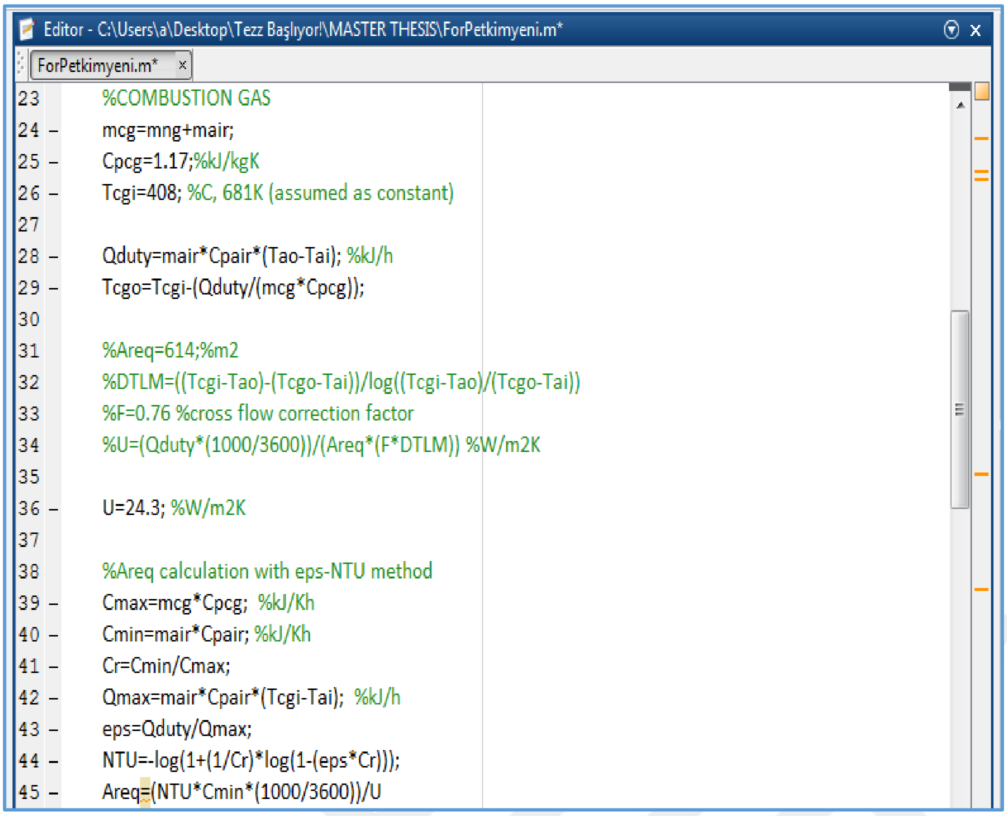
$$U = \frac{11.3 \times 10^6/3600}{614 \times 209} \times 1000 = 24.4\frac{W}{m^2K}$$

## Matlab Simulation Codes

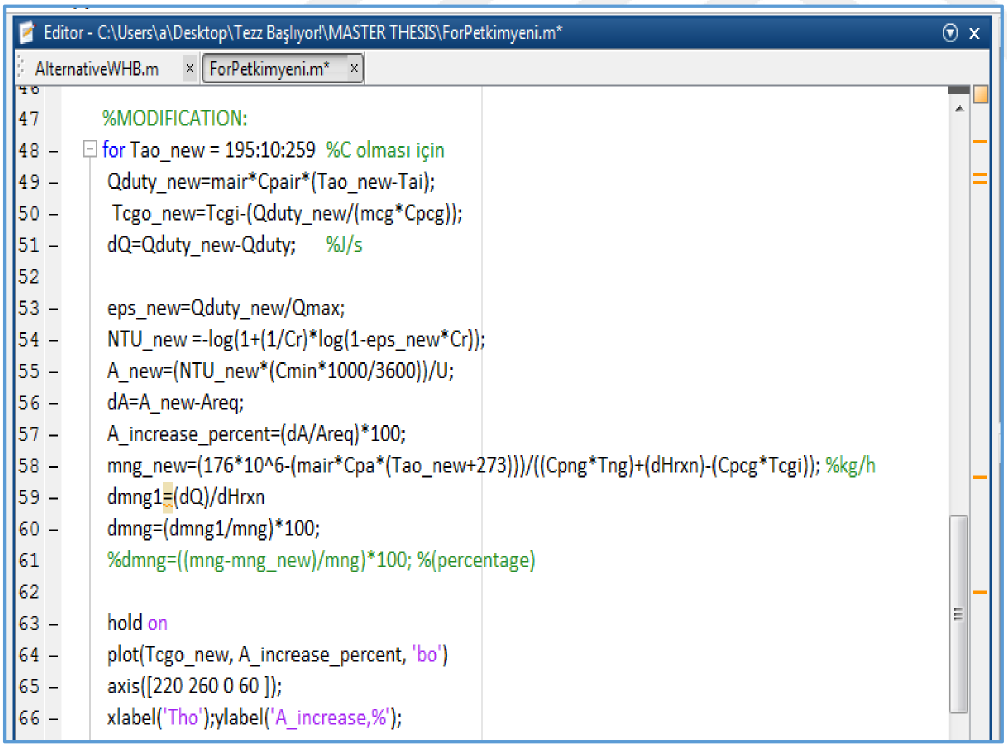
## APH Simulation



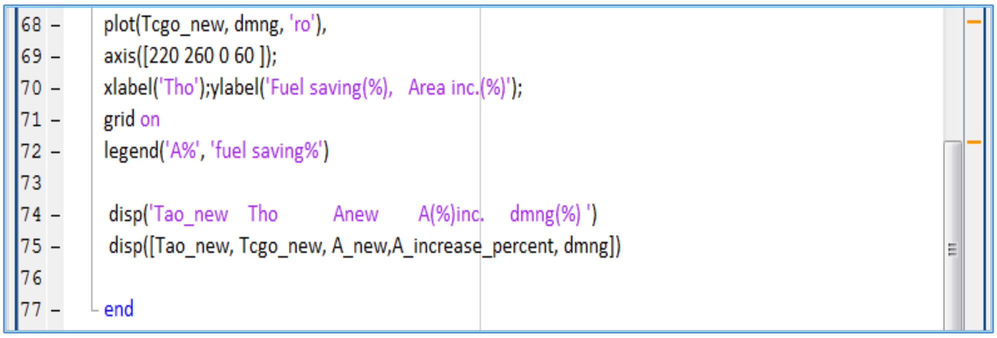
```
clc
%Mass and Energy Balances Around FURNACE
mair=61602; %kg/h
Tao=195; %C, 469K
Cpa=1.027; %kJ/kgK
dHrxn=50001; %kJ/kg
Qfurnace=176*10^6; %kJ/h
Cpcg=1.17; %kJ/kgK
Tcgi=408; %C, 681K (assumed as constant)
Cpng=2.2; %kJ/kgK
Tng=20;%C, 293
%mng=(159.3*10^6-(mair*Cpa*(Tao+273)))/((Cpng*Tng)+(dHrxn)-(Cpcg*Tcgi))
mng=3960;

%Mass and Energy Balances Around APH
%Data in APH
% AIR
%mair=61602; %kg/h
Cpair=1.01; %kJ/kgK
Tai=12.4; %C, 285K
Tao=195; %C, 469K
```



```
%COMBUSTION GAS
mcg=mng+mair;
Cpcg=1.17;%kJ/kgK
Tcgi=408; %C, 681K (assumed as constant)

Qduty=mair*Cpair*(Tao-Tai); %kJ/h
Tcgo=Tcgi-(Qduty/(mcg*Cpcg));

%Areq=614;%m2
%DTLM=((Tcgi-Tao)-(Tcgo-Tai))/log((Tcgi-Tao)/(Tcgo-Tai))
%F=0.76 %cross flow correction factor
%U=(Qduty*(1000/3600))/(Areq*(F*DTLM)) %W/m2K

U=24.3; %W/m2K

%Areq calculation with eps-NTU method
Cmax=mcg*Cpcg;  %kJ/Kh
Cmin=mair*Cpair; %kJ/Kh
Cr=Cmin/Cmax;
Qmax=mair*Cpair*(Tcgi-Tai);  %kJ/h
eps=Qduty/Qmax;
NTU=-log(1+(1/Cr)*log(1-(eps*Cr)));
Areq=(NTU*Cmin*(1000/3600))/U
```



```
%MODIFICATION:
for Tao_new = 195:10:259  %C olması için
Qduty_new=mair*Cpair*(Tao_new-Tai);
 Tcgo_new=Tcgi-(Qduty_new/(mcg*Cpcg));
dQ=Qduty_new-Qduty;     %J/s

eps_new=Qduty_new/Qmax;
NTU_new =-log(1+(1/Cr)*log(1-eps_new*Cr));
A_new=(NTU_new*(Cmin*1000/3600))/U;
dA=A_new-Areq;
A_increase_percent=(dA/Areq)*100;
mng_new=(176*10^6-(mair*Cpa*(Tao_new+273)))/((Cpng*Tng)+(dHrxn)-(Cpcg*Tcgi)); %kg/h
dmng1=(dQ)/dHrxn
dmng=(dmng1/mng)*100;
%dmng=((mng-mng_new)/mng)*100; %(percentage)

hold on
plot(Tcgo_new, A_increase_percent, 'bo')
axis([220 260 0 60 ]);
xlabel('Tho');ylabel('A_increase,%');
```



```
plot(Tcgo_new, dmng, 'ro'),
axis([220 260 0 60 ]);
xlabel('Tho');ylabel('Fuel saving(%),   Area inc.(%)');
grid on
legend('A%', 'fuel saving%')

 disp('Tao_new   Tho        Anew      A(%)inc.    dmng(%) ')
 disp([Tao_new, Tcgo_new, A_new,A_increase_percent, dmng])

end
```

## Energy balance around WHB

**Table 7.2.** Physical properties of water at average temperature (T=60 °C) and saturated steam

| **Property** | **Liquid Phase** | **Gas Phase** |
|---|---|---|
| Viscosity, μ (Pas) | 0,000547 | 0,000015 |
| Density, ρ (kg/m$^3$) | 998 | 2.16 |
| Thermal conductivity, k (W/mK) | 0.65 | 0.0184 |
| Specific heat, Cp (kJ/kgK) | 4.185 | 1.901 |

**Table 7.3.** Chemical composition of combustion gas and physical properties of combustion gases at average temperature (T=200 °C).

| Gas Name | Composition (mol %) | Viscosity, μ (Pas) | Thermal conductivity, k (W/mK) | Specific heat, Cp (kJ/kgK) |
|---|---|---|---|---|
| $N_2$ | 0.7368 | 0.0000206 | 0.0276 | 1.04 |
| $H_2O$ | 0.1000 | 0.000015 | 0.0228 | 2.01 |
| $O_2$ | 0.0330 | 0.0000240 | 0.0285 | 0.92 |
| $CO_2$ | 0.1202 | 0.0000182 | 0.0183 | 0.84 |
| $SO_2$ | 0.0008 | 0.0000158 | 0.0226 | 0.64 |
| Ar | 0.0092 | 0.0000158 | 0.0225 | 0.52 |

Antoine constants of water

$$A = 16.26, B = 3799.9, C = -46.8$$

Saturation temperature of waste water at 2 bar

$$T_{sat} = \frac{3799.9}{16.26 - \ln(2 \times 100)} - (-46.8) = 120.2℃$$

Other values of saturation temperature at different pressures were shown through Table 5.6 to 5.9.

Latent heat of vaporization of water:

$$h_{fg} = 2501.3 - 2.363T_{sat} - 0.000468T_{sat}^{2} - 0.00001266 \;\; [kJ/kg]$$

Amount of produced steam:

$$\dot{m}_{steam} = \frac{\dot{m}_{CG}Cp_{CG}(T_{CG,in} - T_{CG,out})}{\left[Cp_{w}\left(T_{sat} - T_{w,in}\right) + h_{fg}\right]} \; [kg/h]$$

Sensible and latent heating rates:

$$\dot{Q}_{sensible} = \dot{m}_{steam} * Cp_{w}\left(T_{sat} - T_{w,in}\right) = U_{sens}A_{sens}\Delta T_{lm,sens} \;\; [kJ/h]$$

$$\dot{Q}_{latent} = \dot{m}_{steam} * h_{fg} = U_{latent}A_{latent}\Delta T_{lm,latent} \; [kJ/h]$$

$$\dot{Q}_{total} = \dot{Q}_{sensible} + \dot{Q}_{latent} = U_{total}A_{total}\Delta T_{lm,total} \;\; [kJ/h]$$

Overall heat transfer coeffficient in sensible heating zone:

$$\frac{1}{U_{sens}} = \frac{1}{h_o} + \frac{1}{h_{i,sens}}\left(\frac{d_o}{d_i}\right)$$

Overall heat transfer coeffficient in latent heating zone:

$$\frac{1}{U_{latent}} = \frac{1}{h_o} + \frac{1}{h_{i,latent}}\left(\frac{d_o}{d_i}\right)$$

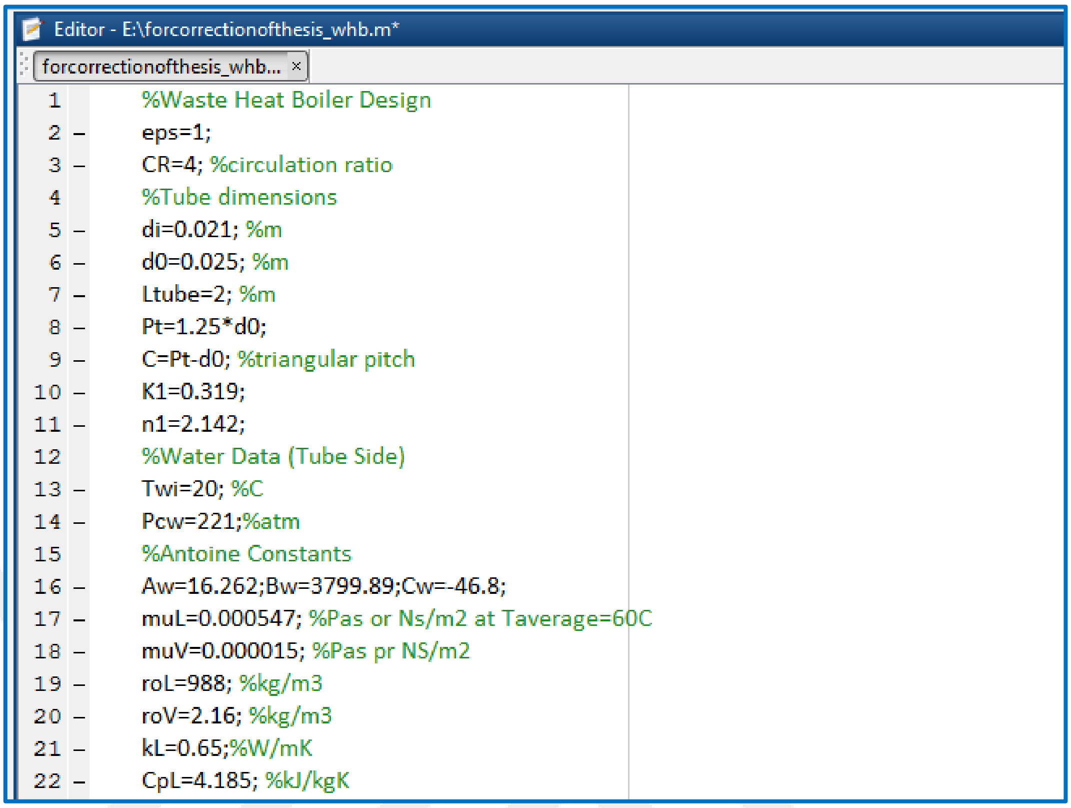
$$h_{i,latent} = h_{fc} + h_{nb} \; [W/m^{2}K]$$

where $h_{fc}$ indicates the heat transfer coefficient by forced convection and $h_{nb}$ by nucleate boiling in the tubes.

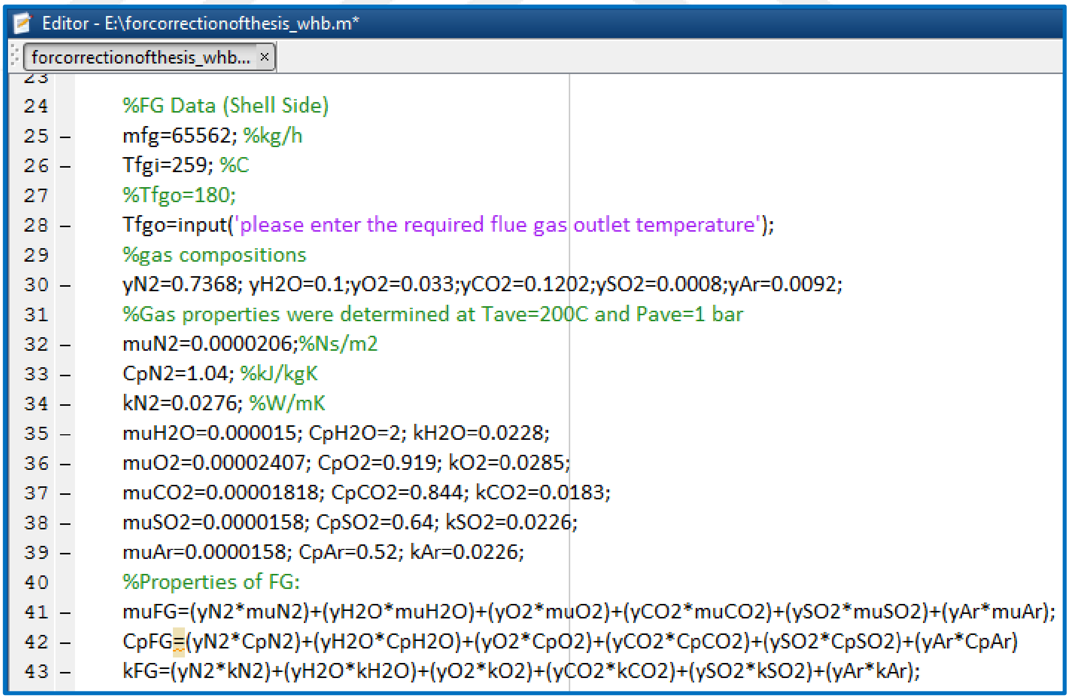
Total number of tubes:

$$N_{tube} = \frac{A_{total}}{\pi * d_o * l_{tube}}$$

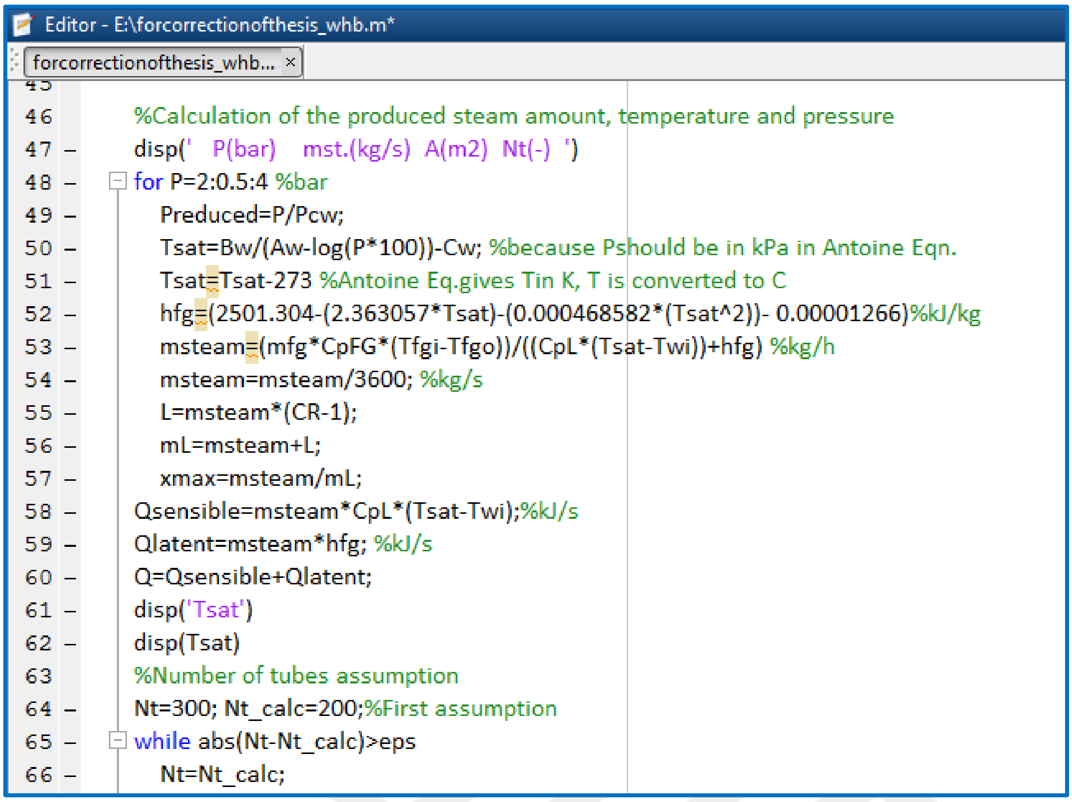
## WHB Simulation



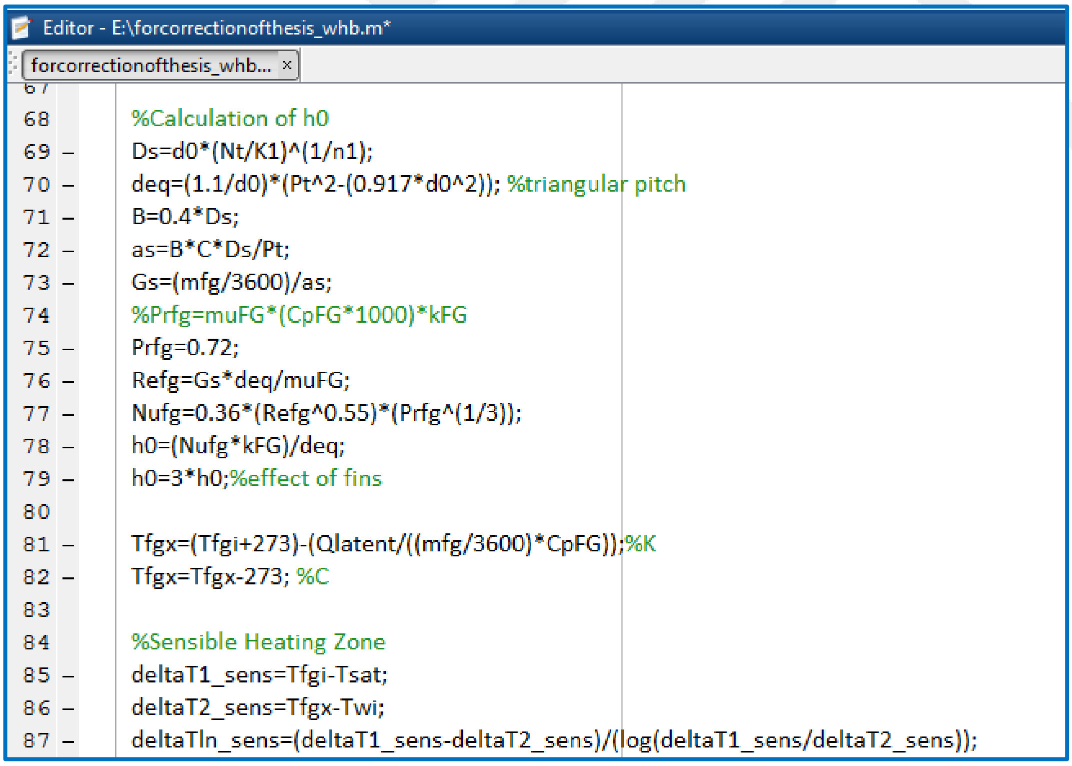
```
%Waste Heat Boiler Design
eps=1;
CR=4; %circulation ratio
%Tube dimensions
di=0.021; %m
d0=0.025; %m
Ltube=2; %m
Pt=1.25*d0;
C=Pt-d0; %triangular pitch
K1=0.319;
n1=2.142;
%Water Data (Tube Side)
Twi=20; %C
Pcw=221;%atm
%Antoine Constants
Aw=16.262;Bw=3799.89;Cw=-46.8;
muL=0.000547; %Pas or Ns/m2 at Taverage=60C
muV=0.000015; %Pas pr NS/m2
roL=988; %kg/m3
roV=2.16; %kg/m3
kL=0.65;%W/mK
CpL=4.185; %kJ/kgK
```



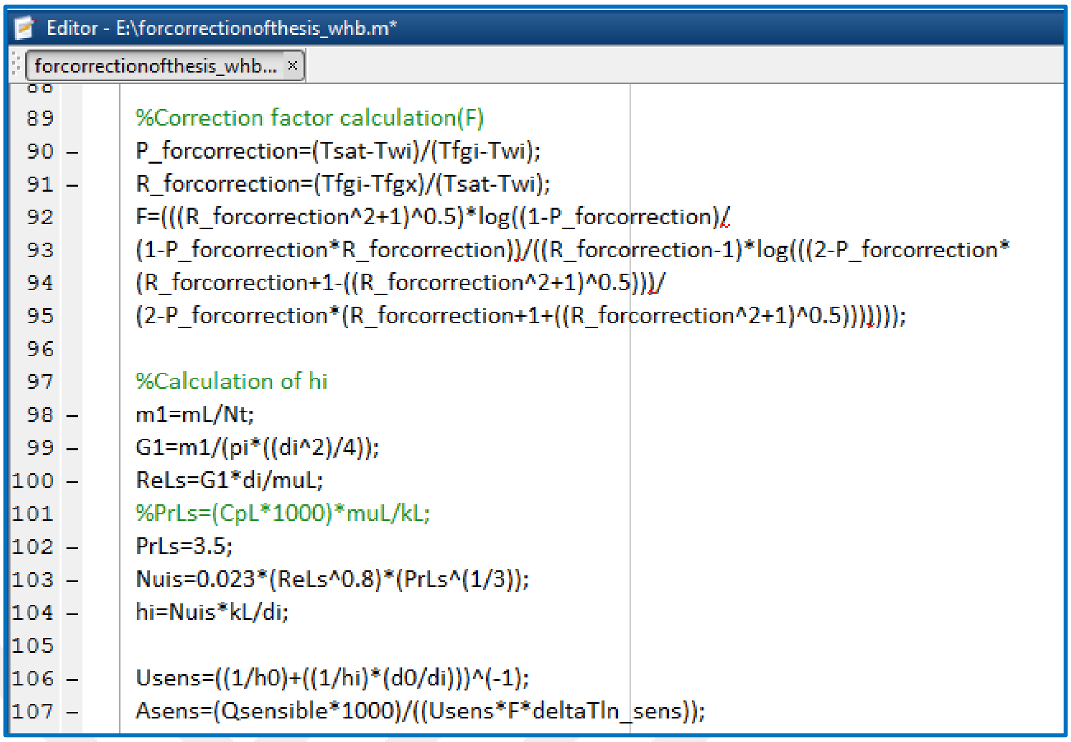
```
%FG Data (Shell Side)
mfg=65562; %kg/h
Tfgi=259; %C
%Tfgo=180;
Tfgo=input('please enter the required flue gas outlet temperature');
%gas compositions
yN2=0.7368; yH2O=0.1;yO2=0.033;yCO2=0.1202;ySO2=0.0008;yAr=0.0092;
%Gas properties were determined at Tave=200C and Pave=1 bar
muN2=0.0000206;%Ns/m2
CpN2=1.04; %kJ/kgK
kN2=0.0276; %W/mK
muH2O=0.000015; CpH2O=2; kH2O=0.0228;
muO2=0.00002407; CpO2=0.919; kO2=0.0285;
muCO2=0.00001818; CpCO2=0.844; kCO2=0.0183;
muSO2=0.0000158; CpSO2=0.64; kSO2=0.0226;
muAr=0.0000158; CpAr=0.52; kAr=0.0226;
%Properties of FG:
muFG=(yN2*muN2)+(yH2O*muH2O)+(yO2*muO2)+(yCO2*muCO2)+(ySO2*muSO2)+(yAr*muAr);
CpFG=(yN2*CpN2)+(yH2O*CpH2O)+(yO2*CpO2)+(yCO2*CpCO2)+(ySO2*CpSO2)+(yAr*CpAr)
kFG=(yN2*kN2)+(yH2O*kH2O)+(yO2*kO2)+(yCO2*kCO2)+(ySO2*kSO2)+(yAr*kAr);
```



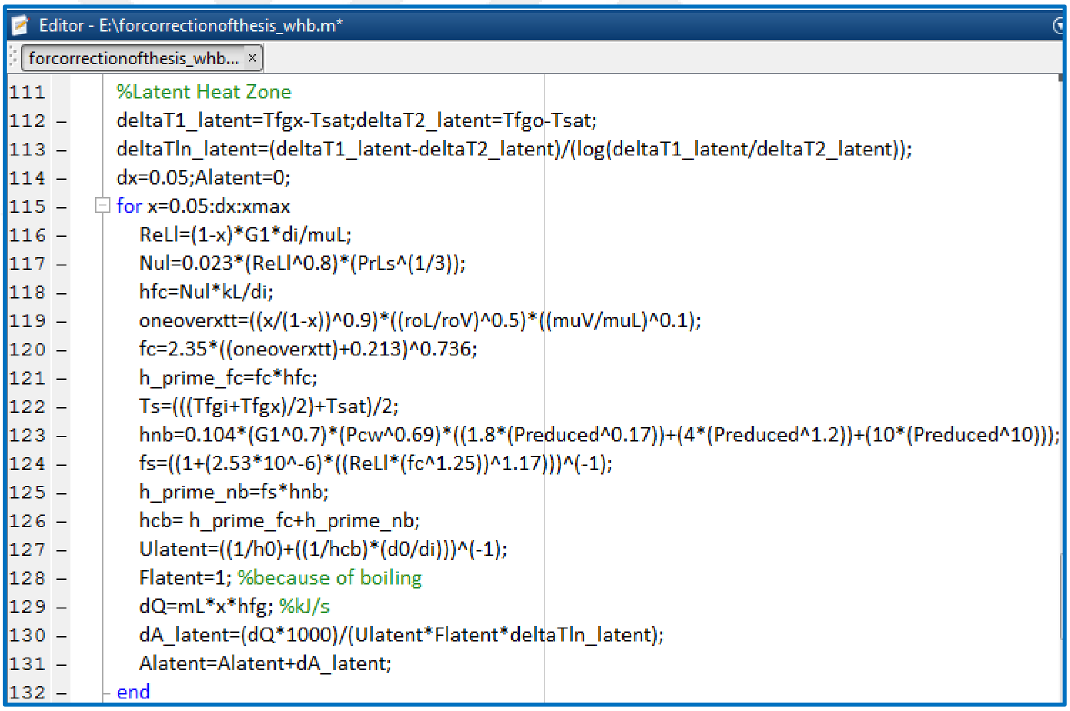
```
%Calculation of the produced steam amount, temperature and pressure
disp('   P(bar)    mst.(kg/s)  A(m2)  Nt(-)  ')
for P=2:0.5:4 %bar
    Preduced=P/Pcw;
    Tsat=Bw/(Aw-log(P*100))-Cw; %because Pshould be in kPa in Antoine Eqn.
    Tsat=Tsat-273 %Antoine Eq.gives Tin K, T is converted to C
    hfg=(2501.304-(2.363057*Tsat)-(0.000468582*(Tsat^2))- 0.00001266)%kJ/kg
    msteam=(mfg*CpFG*(Tfgi-Tfgo))/((CpL*(Tsat-Twi))+hfg) %kg/h
    msteam=msteam/3600; %kg/s
    L=msteam*(CR-1);
    mL=msteam+L;
    xmax=msteam/mL;
Qsensible=msteam*CpL*(Tsat-Twi);%kJ/s
Qlatent=msteam*hfg; %kJ/s
Q=Qsensible+Qlatent;
disp('Tsat')
disp(Tsat)
%Number of tubes assumption
Nt=300; Nt_calc=200;%First assumption
while abs(Nt-Nt_calc)>eps
    Nt=Nt_calc;
```



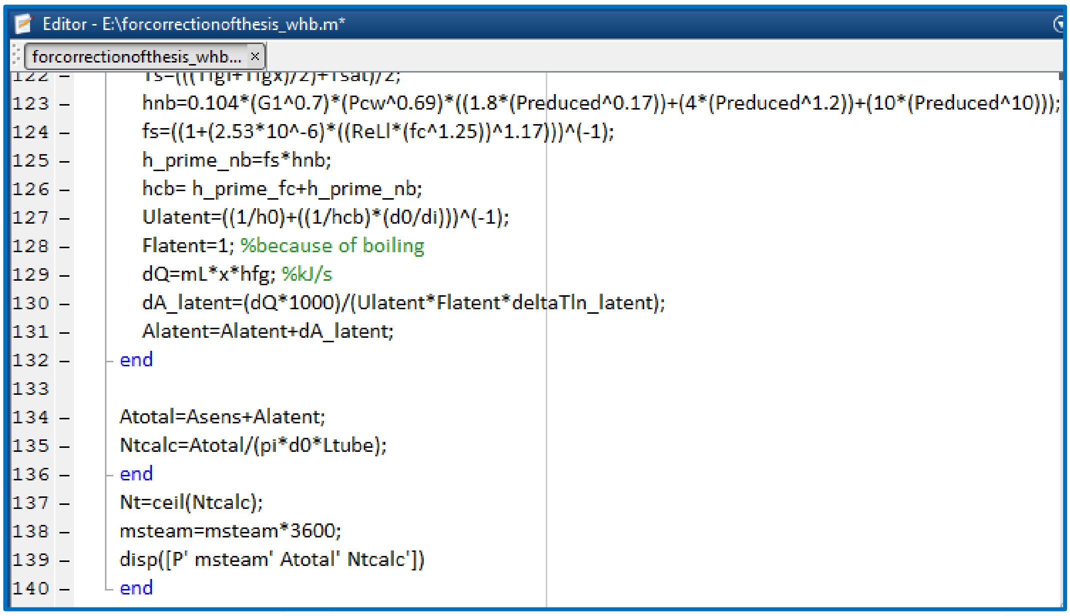
```
%Calculation of h0
Ds=d0*(Nt/K1)^(1/n1);
deq=(1.1/d0)*(Pt^2-(0.917*d0^2)); %triangular pitch
B=0.4*Ds;
as=B*C*Ds/Pt;
Gs=(mfg/3600)/as;
%Prfg=muFG*(CpFG*1000)*kFG
Prfg=0.72;
Refg=Gs*deq/muFG;
Nufg=0.36*(Refg^0.55)*(Prfg^(1/3));
h0=(Nufg*kFG)/deq;
h0=3*h0;%effect of fins

Tfgx=(Tfgi+273)-(Qlatent/((mfg/3600)*CpFG));%K
Tfgx=Tfgx-273; %C

%Sensible Heating Zone
deltaT1_sens=Tfgi-Tsat;
deltaT2_sens=Tfgx-Twi;
deltaTln_sens=(deltaT1_sens-deltaT2_sens)/(log(deltaT1_sens/deltaT2_sens));
```



```
%Correction factor calculation(F)
P_forcorrection=(Tsat-Twi)/(Tfgi-Twi);
R_forcorrection=(Tfgi-Tfgx)/(Tsat-Twi);
F=(((R_forcorrection^2+1)^0.5)*log((1-P_forcorrection)/
(1-P_forcorrection*R_forcorrection))/((R_forcorrection-1)*log(((2-P_forcorrection*
(R_forcorrection+1-((R_forcorrection^2+1)^0.5)))/
(2-P_forcorrection*(R_forcorrection+1+((R_forcorrection^2+1)^0.5))))));

%Calculation of hi
m1=mL/Nt;
G1=m1/(pi*((di^2)/4));
ReLs=G1*di/muL;
%PrLs=(CpL*1000)*muL/kL;
PrLs=3.5;
Nuis=0.023*(ReLs^0.8)*(PrLs^(1/3));
hi=Nuis*kL/di;

Usens=((1/h0)+((1/hi)*(d0/di)))^(-1);
Asens=(Qsensible*1000)/((Usens*F*deltaTln_sens));
```



```
%Latent Heat Zone
deltaT1_latent=Tfgx-Tsat;deltaT2_latent=Tfgo-Tsat;
deltaTln_latent=(deltaT1_latent-deltaT2_latent)/(log(deltaT1_latent/deltaT2_latent));
dx=0.05;Alatent=0;
for x=0.05:dx:xmax
    ReLl=(1-x)*G1*di/muL;
    Nul=0.023*(ReLl^0.8)*(PrLs^(1/3));
    hfc=Nul*kL/di;
    oneoverxtt=((x/(1-x))^0.9)*((roL/roV)^0.5)*((muV/muL)^0.1);
    fc=2.35*((oneoverxtt)+0.213)^0.736;
    h_prime_fc=fc*hfc;
    Ts=(((Tfgi+Tfgx)/2)+Tsat)/2;
    hnb=0.104*(G1^0.7)*(Pcw^0.69)*((1.8*(Preduced^0.17))+(4*(Preduced^1.2))+(10*(Preduced^10)));
    fs=((1+(2.53*10^-6)*((ReLl*(fc^1.25))^1.17)))^(-1);
    h_prime_nb=fs*hnb;
    hcb= h_prime_fc+h_prime_nb;
    Ulatent=((1/h0)+((1/hcb)*(d0/di)))^(-1);
    Flatent=1; %because of boiling
    dQ=mL*x*hfg; %kJ/s
    dA_latent=(dQ*1000)/(Ulatent*Flatent*deltaTln_latent);
    Alatent=Alatent+dA_latent;
end
```



```
        hnb=0.104*(G1^0.7)*(Pcw^0.69)*((1.8*(Preduced^0.17))+(4*(Preduced^1.2))+(10*(Preduced^10)));
        fs=((1+(2.53*10^-6)*((ReLl*(fc^1.25))^1.17)))^(-1);
        h_prime_nb=fs*hnb;
        hcb= h_prime_fc+h_prime_nb;
        Ulatent=((1/h0)+((1/hcb)*(d0/di)))^(-1);
        Flatent=1; %because of boiling
        dQ=mL*x*hfg; %kJ/s
        dA_latent=(dQ*1000)/(Ulatent*Flatent*deltaTln_latent);
        Alatent=Alatent+dA_latent;
    end

    Atotal=Asens+Alatent;
    Ntcalc=Atotal/(pi*d0*Ltube);
    end
    Nt=ceil(Ntcalc);
    msteam=msteam*3600;
    disp([P' msteam' Atotal' Ntcalc'])
    end
```

## Calculations of Conventional Exergy Analysis

**Table 7.4.** Stream compositions

| | **Molar Composition,%** | | | | | |
|---|---|---|---|---|---|---|
| | $N_2(g)$ | $O_2(g)$ | $CO_2(g)$ | $H_2O(g)$ | $CH_4(g)$ | $C_2H_6(g)$ |
| **Air** | 0.7902 | 0.2095 | 0.0003 | - | - | - |
| **Natural gas*** | - | - | - | - | 0.97 | 0.03 |
| **Combustion gas** | 0.745 | 0.0844 | 0.056 | 0.0113 | - | - |

*It was assumed that natural gas consists of only methane.

**Table 7.4.** Standart entalphies of compounds (For reference state: T=298K & P=1bar)

| **Compound** | **$Cp^0$[kJ/kmolK]** | **$h^0$[kJ/kmol]** | **$s^0$[kJ/kmolK]** |
|---|---|---|---|
| **$N_2$ (g)** | 28.49 | 0 | 191.6 |
| **$O_2$ (g)** | 28.92 | 0 | 205.1 |
| **$CO_2$ (g)** | 35.91 | -393521 | 213.8 |
| **$H_2O$ (g)** | 31.96 | -241856 | 188.8 |
| **$H_2O$ (l)** | 75.79 | -285829 | 69.9 |
| **$CH_4$(g)** | 35.05 | -74872 | 186.0 |

For 298<T<Tmax, Pref=1 bar, y=10-3T

**Table 7.2.** Entalphy and entropy constants of compounds

| **Compound** | **H**$^+$ | **S**$^+$ | **a** | **B** | **c** | **d** |
|---|---|---|---|---|---|---|
| **$N_2$ (g)** | -9.982 | 16.203 | 30.418 | 2.540 | -0.238 | 0.000 |
| **$O_2$ (g)** | -9.589 | 36.16 | 29.154 | 6.470 | -0.184 | -1.017 |
| **$CO_2$ (g)** | -413.886 | -87.07 | 51.128 | 4.360 | -1.469 | 0.000 |
| **$H_2O$ (g)** | -253.87 | -11.75 | 34.376 | 7.840 | -0.423 | 0.000 |
| **$CH_4$(g)** | -81.242 | 96.731 | 11.933 | 77.647 | 0.142 | -18.414 |

Calculated entalphy and entropy values of the streams are shown in Table 5.11 and Table 5.12.

Determined physical, chemical and total exergy values of streams according to Eq.36 and Eq.37 are tabulated through Table 5.13-5.17.

To compute exergetic efficiency of the units, fuel and product of each unit were identified as follows and by using Eq.40 and Eq.42, calculated values of exergetic efficiency were shown in Table 5.18.

**Table 7.3.** Identification of fuel and product of the units

| | **Furnace** | **APH** | **WHB** |
|---|---|---|---|
| **Fuel Exergy** | $Ex_1 + Ex_2$ | $Ex_4 - Ex_3$ | $Ex_4 - Ex_7$ |
| **Product Exergy** | $Ex_3 + \Delta Ex_{feed}$ | $Ex_2 - Ex_5$ | $Ex_9 - Ex_8$ |

## Calculations of Advanced Exergy Analysis

Figure7.1 and Figure 7.2 shows the flow chart of the furnace with unavoidable conditions stated in Table 5.19.

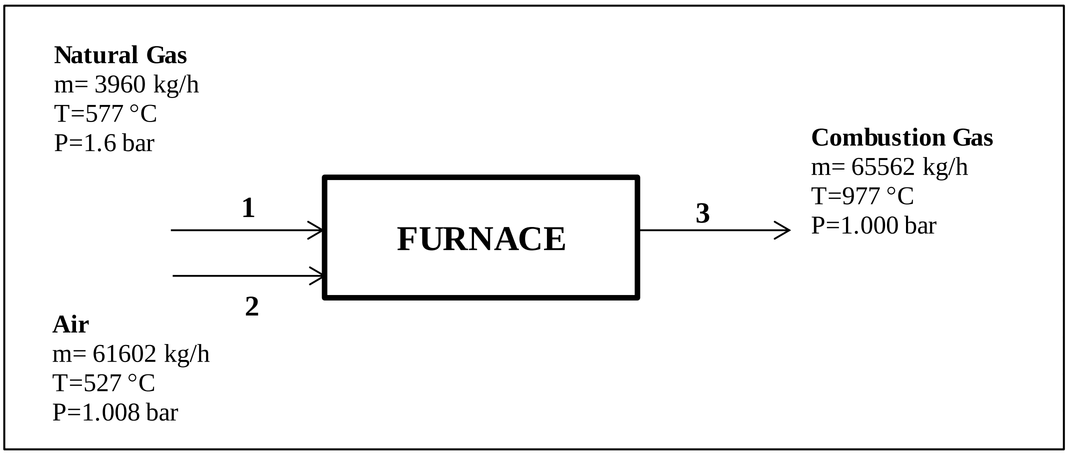


**Figure 7.1** Furnace operating data with unavoidable conditions

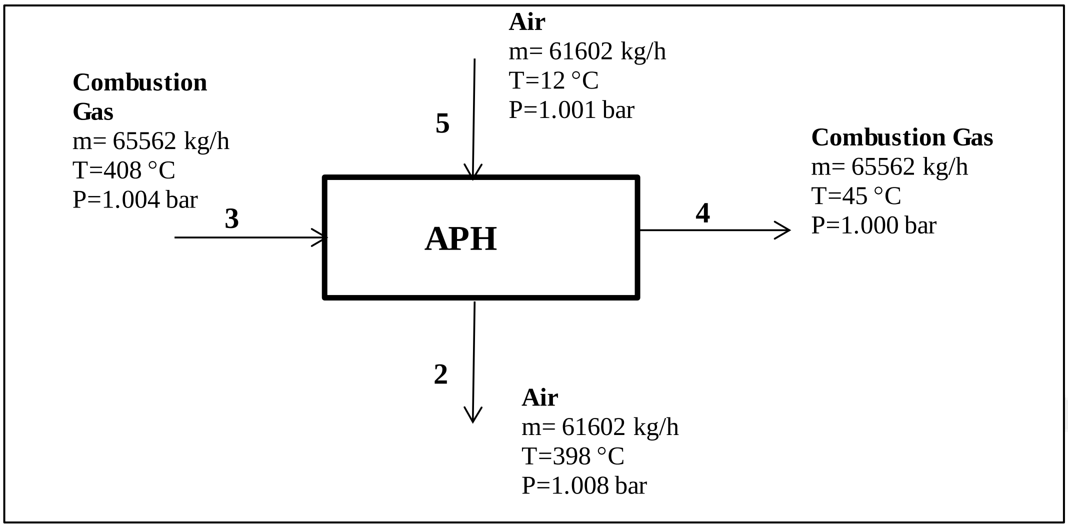


**Figure 7.2** APH operating data with unavoidable conditions

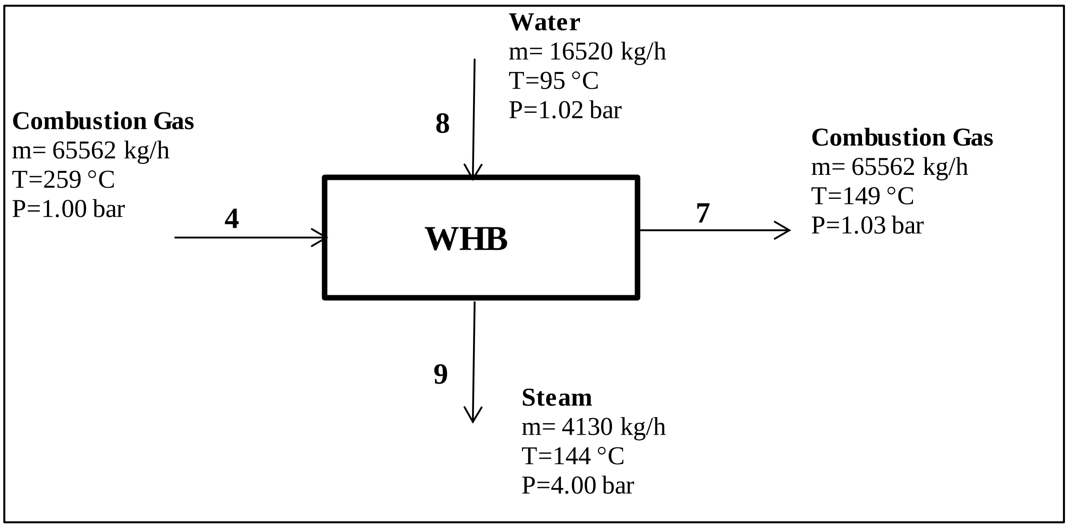


**Figure 7.3** WHB operating data with unavoidable conditions

Calculated physical, chemical and total exergy rates of streams with unavoidable conditions were shown between Table 5.20-5.25 and Table 5.28-5.32.

**Determination of unavoidable exergy destructions of the units**

Eq.44 helps us to determine the amount of unavoidable destruction within a system component. To obtain this, amount of fuel, product and destructed exergy values of units for real and best conditions were summarized in Table 7.6.

**Table 7.4.** Amount of exergy values with real and best conditions

| | **FURNACE** | | **APH** | | **WHB** | |
|---|---|---|---|---|---|---|
| | Real | Best | Real | Best | Real | Best |
| **Fuel Exergy** | 54.831 | 59.49 | 1.65 | 2.87 | 0.60 | 0.84 |
| **Product Exergy** | 22.74 | 31.18 | 0.62 | 2.39 | 0.26 | 0.53 |
| **Destructed Exergy** | 32.091 | 28.30 | 1.02 | 0.48 | 0.34 | 0.31 |

Unavoidable exergy desturction of furnace:

$$\dot{E}x_{D,furnace}^{UN} = \dot{E}x_{P,k}\left(\frac{\dot{E}x_{D,k}}{\dot{E}x_{P,best,k}}\right)^{UN} = 22.74\left(\frac{28.30}{31.18}\right) = 20.39\ MW$$

Unavoidable exergy desturction of APH:

$$\dot{E}x_{D,APH}^{UN} = \dot{E}x_{P,k}\left(\frac{\dot{E}x_{D,k}}{\dot{E}x_{P,best,k}}\right)^{UN} = 0.62\left(\frac{0.48}{2.39}\right) = 0.12\ MW$$

Unavoidable exergy desturction of WHB:

$$\dot{E}x_{D,WHB}^{UN} = \dot{E}x_{P,k}\left(\frac{\dot{E}x_{D,k}}{\dot{E}x_{P,best,k}}\right)^{UN} = 0.26\left(\frac{0.31}{0.53}\right) = 0.16\ MW$$

The value of avoidable destructions and modified exergetix efficiencies were shown in Table 5.32.