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THESIS TOPIC

Replacement of the E-504 Heat Exchanger with an Air Cooler: Design and Thermal Calculation

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Defended on .../06/2025 before the jury composed of:

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Dedication

To my beloved **parents**, your love, sacrifices, and endless support have been the guiding light in my life. This work is a small reflection of all that you've given me, and I owe every step of this journey to you.

To my dear **brothers and sister**, thank you for always being there, for your encouragement, and for the strength your presence brings.

To my **extended family**, my uncle, my cousins, aunts, and maternal uncles your kindness and support have meant more to me than words can express.

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I dedicate this thesis to each of you, with all my heart.

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NOMENCLATURE

ASME: American Society of Mechanical Engineers
API661: Air-Cooled Heat Exchangers for General Refinery Service
BSI: British Standards Institute
CODAP: Design Code for Pressure Vessels Not Subject to Flame
DTLM: Logarithmic Mean Temperature Difference Method
EDM: Seawater
LNG: Liquefied Natural Gas
ISO: International Organization for Standardization
ISPELS: National Institute for Occupational Safety and Prevention
MEA: Monoethanolamine
MCR: Multi-Component Refrigerant
USA: United States of America

C: Specific heat capacity (J/kg·°C)
di: Inner diameter of the tube (m)
de: Outer diameter of the tube (m)
h: Convective heat transfer coefficient (kcal/h·m^2·°C)
K: Overall heat transfer coefficient (kcal/h·m^2·°C)
L: Tube length (m)
ma: Air mass flow rate (kg/h)
Nr: Number of rows
Nu: Nusselt number
Nt: Total number of tubes
Pr: Prandtl number
ΔPi: Pressure drop inside the tube (Pa)
ΔPT: Total pressure drop (Pa)
Re: Reynolds number
Sins: Installed heat transfer surface area (m^2)
Sp: Fluid passage cross-sectional area (m^2)
Φ: Heat flux (W)

GREEK SYMBOLS

λ: Thermal conductivity (k·s)
μ: Dynamic viscosity (Pa·s)
ρ: Density (kg/m^3)
δ: Fin spacing
ν: Flow velocity
f: Friction factor

EQUIPMENT NAMES

- **X01-E-506**: Gas Preheater
- **X01-E-503A and E-503B**: MEA Solution Exchanger
- **X01-E-504**: MEA Solution Cooler
- **X01-E-505**: MEA Purifier
- **X01-E-501**: Overhead MEA Condenser
- **X01-F-501**: MEA Regeneration Column
- **X01-F-502**: MEA Absorption Column
- **X01-G-507**: MEA Flash Drum
- **X01-G-502**: Acid Gas Separator

Abstract

Due to the recurring unreliability of the heat exchanger in the decarbonation unit of the GNL1/Z complex in Arzew, a study was proposed to assess the feasibility of replacing the seawater heat exchanger with an air-cooled alternative.

Following a comprehensive bibliographic review, we opted for a forced-draft dry air cooler. The sizing of this alternative was carried out using the Logarithmic Mean Temperature Difference (ΔTML) method. The proposed air cooler is presented as a potential substitute for the shell-and-tube exchanger, with a thorough evaluation of its advantages and limitations.

Keywords: sizing, air cooler, finned tubes, ΔTML method.

Résumé

En raison du manque de fiabilité de l'échangeur de chaleur de l'unité de décarbonatation du complexe GNL1/Z d'Arzew, une étude a été proposée afin d'évaluer l'opportunité de remplacer l'échangeur à eau de mer par un aéroréfrigérant.

Après une analyse bibliographique approfondie, notre choix s'est porté sur un aéroréfrigérant sec à tirage forcé. Son dimensionnement a été réalisé à l'aide de la méthode du ΔTML (différence moyenne logarithmique de température). L'aéroréfrigérant est proposé comme solution alternative à l'échangeur à calandre et à tubes, en tenant compte de ses avantages et inconvénients.

Mots clés : dimensionnement, aéroréfrigérant, tubes à ailettes, méthode ΔTML.

الملخص

نظرًا لعدم موثوقية المبادلات الحرارية في وحدة نزع الكربون التابعة لمركب GNL1/Z بأرزيو، تم اقتراح دراسة حول إمكانية استبدال المبادلات التي تعمل بمياه البحر بمبردات هوائية.

وبعد تحليل مرجعي شامل، تم اختيار مبرد هواء جاف من النوع المدفوع قسريًا. تم اعتماد طريقة فرق درجة الحرارة الوسطى اللوغاريتمية (ΔTML) كمنهجية للحسابات التصميمية. يُقترح هذا المبرد كحل بديل محتمَل للمبادل الحراري من نوع القشرة والأنابيب، مع الأخذ بعين الاعتبار مزاياه وعيوبه.

الكلمات المفتاحية :الحساب التصميمي، مبرد هوائي، أنابيب مزعنفَة، طريقة.ΔTML

INTRODUCTION

The global natural gas trade is rising exponentially because of growing energy requirements and the trend towards utilizing cleaner energy. Natural gas, its enormous global reserves and smaller environmental impact compared to other fossil fuels, is taking center stage of the world energy transition.Natural gas reserves standing at approximately 3,000 billion cubic meters position Algeria as the fourth largest country in the world after the United States, Russia, and Iran. As the country's economy relies largely on hydrocarbon exports, liquefied natural gas (LNG) is Algeria's industrial and economic priority.

Gas treatment, i.e., the decarbonation stage, marks the beginning of natural gas liquefaction, and it ends with the cryogenic stage, where the gas is chilled to extremely low temperatures and converted into liquid form to be shipped and stored. However, in certain situations, there are production shortfalls, whereby the amount of LNG produced is lower than forecasted. The deficits in the majority of instances are due to problems in the operation of various units in the process.Though there are several operational issues that may be repeated, one recurring issue in the GL1/Z natural gas liquefaction complex is the E-504 heat exchanger. The exchanger is important for cooling the regenerated lean MEA (monoethanolamine) solution down to a terminal temperature of 38°C. The MEA passes on the shell side, and corrosive seawater, used as the cooling medium, is on the tube side. Fouling and corrosion have increased in severity with the passage of time and have affected the efficiency of this exchanger by causing inefficient heat transfer and low process efficiency. To counteract this issue, a replacement proposal has been made replace the shell-and-tube exchanger with an air-cooled heat exchanger (aerocooler). Air coolers transfer heat from the process fluid in the tubes to surrounding air, offering an available solution to fouling reduction and equipment longevity.

This thesis is organized as follows:

- **Chapter I:** Description of the GL1/Z Complex
- **Chapter II:** Decarbonation and problem of Fouling E.504 HEAT Exchangers
- **Chapter III:** Air Cooler Heat Exchanger Technology (ACHEs)
- **Chapter IV :** Design and Thermal Sizing of Air-Cooled Heat Exchangers

A conclusion and recommendations will complete this thesis.

CHAPTER I

CHAPTER I: DESCRIPTION OF THE GL1/Z COMPLEX

I.1. Introduction

The GL1/Z complex is a significant industrial facility in Algeria, operated by SONATRACH, and plays a vital role in the hydrocarbon transformation sector. The complex is designed to process and liquefy natural gas transported from the Hassi R'mel gas fields through a 42 inch diameter, 500 km long gas pipeline.

The natural gas liquefaction process is crucial for enabling the transport of natural gas over long distances. The GL1/Z complex, along with other complexes like GL2/Z and GL3/Z, is a key part of the region's petrochemical and gas industry.

The construction of the GL1/Z complex was a multi-stage endeavor. The first stone was laid on June 16, 1973, by President Houari Boumediene, with the American company BECHTEL initiating the construction work on February 20, 1978. The complex began production in February 1978, with the first LNG deliveries being made to the United States. A major renovation project took place in January 1993 to enhance the complex's reliability, safety, and production capacity, targeting a 110% increase in production.

The GL1/Z complex is located north of Bethioua, about 7 km from Arzew, and covers an area of 72 hectares. The complex includes several zones, such as utilities, process, storage, and pumping.

Technical Data Sheet

- **Area:** 72 hectares
- **Constructed by:** Chemical Incorporation and Bechtel
- **Start-up date:** February 1978
- **Natural gas supply:** From HASSI RMEL
- **Process:** APCI
- **Number of trains:** 06
- **Storage capacity:** 300,000 m³ of LNG
- **Product loading temperature:** -162°C
- **Gas pipeline diameter:** 42 inches
- **Gas pipeline length:** 500 km

Sources and related content

The Complex Has:

- Six identical production trains.
- A utility unit which carries out the generation of steam, electric power, instrument air, pumping of cooling water, and supply of nitrogen to the trains.
- An end product storage and loading facility, known as the terminal.

I.2. Organization of the Complex

The GL1/Z complex is segmented into four main areas:

1. Utility Area
2. Process Area
3. Storage Area
4. Pumping Area

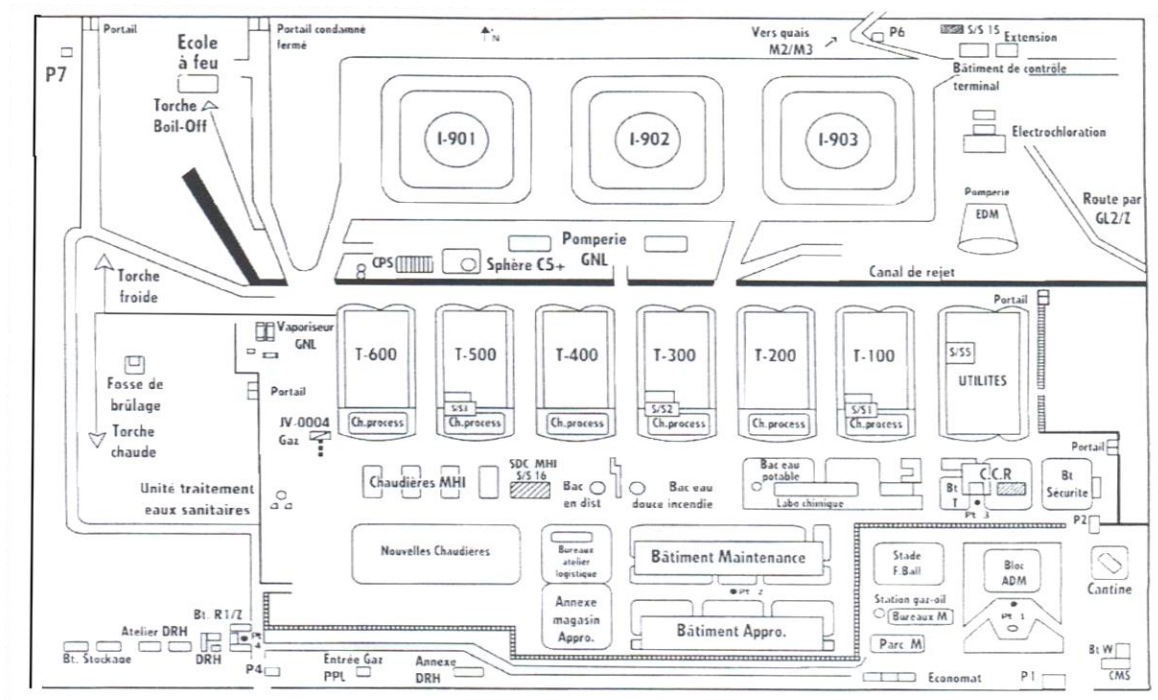

Fig I.1. General Layout of the GL1/Z complex.

I.2.1. Utility Area

This space addresses the provision of all the utilities required for electric power generation, instrument air, distilled water, and nitrogen.

Principal items to be found within this space include:

- Boilers

- Desalination plants

- Nitrogen plant installations

I.2.1.1. Steam Production

Production of steam is significantly essential as an energy source to various equipment including:

- Turbo generators (for power generation)
- Turbo compressors (for KT gas compression: 110/120/121/130)
- Heaters (for heat exchange)
- Turbo pumps (for pumping operation)

The following units generate steam:

- 17 boilers, each with a capacity to produce 115 tons/hour of steam at 62 bar and approximately 442°C.

- 4 boilers with capacities of 400 tons/hour at 62 bar and approximately 442°C.

- 3 boilers with capacities of 91 tons/hour at 62 bar and approximately 442°C.

- 1 low-pressure boiler with a capacity of 51 tons/hour at 27 bar.

I.2.1.2. Electricity Generation

Electricity is generated in three turbo generators (TG) with capacities of 18 MW each.

The electric load of the complex is around 36 MW. In addition to the TGs, the complex benefits from a backup power supply from the SONELGAZ grid in case of energy needs.

I.2.1.3. Seawater Circuit

Seawater is utilized in the complex as a cooling agent in various units. Seawater is used as a source of heat for refrigeration cycles and as a source of cold for steam cycles. Seawater is also used in desalination for the production of fresh water.

I.2.1.4. Production of Desalinated Water

Six parallel desalination units each produce 45 m^3 of fresh water. The plant has been recently equipped with a new type of desalination unit called an ejecto-compressor.

The water is used to feed boilers for the production of steam and is also used as a cooling agent for certain mechanical equipment, such as pumps.

I.2.1.5. Electro-Chlorination

Chlorination of the seawater is used to prevent marine life (e.g., mussels) from developing, which can double and plug seawater condensers.

I.2.1.6. Compressed Air Production

Instrumentation pieces like regulators and valves are actuated pneumatically.

Instrument air is dried ahead to avoid corrosion or deterioration of instrumentation pieces.

The complex utilises about 4600 m^3 of air provided by five centrifugal compressors at the pressure of 10 bar for discharge and stocked in receiver vessels.

I.2.2. Process Area

It consists of six identical liquefaction trains, referred to as trains that are identical in design and operation.

Their consistent industrial design explains their independent operation in liquefying natural gas. Each liquefaction train consists of eight sections and a cooling circuit:

- Decarbonation section (CO_2 removal)
- Dehydration section (H_2O removal)
- Mercury removal section (Hg removal)
- Cooling section
- Separation and wash tower section
- Fractionation section
- Propane cooling section
- Mixed refrigerant circuit
- Liquefaction section

I.2.3. Storage Area

Situated on the seaside, in front of the trains, the storage and shipping area for LNG is made up of three above-ground tanks, each with a capacity of 100,000 m^3.

The complex has two berths to load LNG carriers with a capacity of between 50,000 m^3 and 125,000 m^3.

LNG is stored under a pressure of 1.03 bar and a temperature of -162°C.

I.2.4. Pumping Area

The pumping area has the responsibility of moving LNG from storage tanks to the loading arms.

I.3. Description of the Liquefaction Process

Before getting into the detailed process of liquefaction, it is essential to understand what natural gas is made up of.

Table I.1: Chemical composition of natural gas.

COMPONENT	NOMENCLATURE	AVERAGE MOLAR %
N_2	**Nitrogen**	5,80
He	**Helium**	0,19
CO_2	**Carbon dioxide**	0,21
C_1	**Methane**	83,00
C_2	**Ethane**	7,10
C_3	**Propane**	2,25
iC_4	**Isobutane**	0,40
nC_4	**Butane**	0,40
iC_5	**Isopentane**	0,12
nC_5	**Pentane**	0,15
C_{6+}	**Gasolines**	0,18
TOTAL		100,00

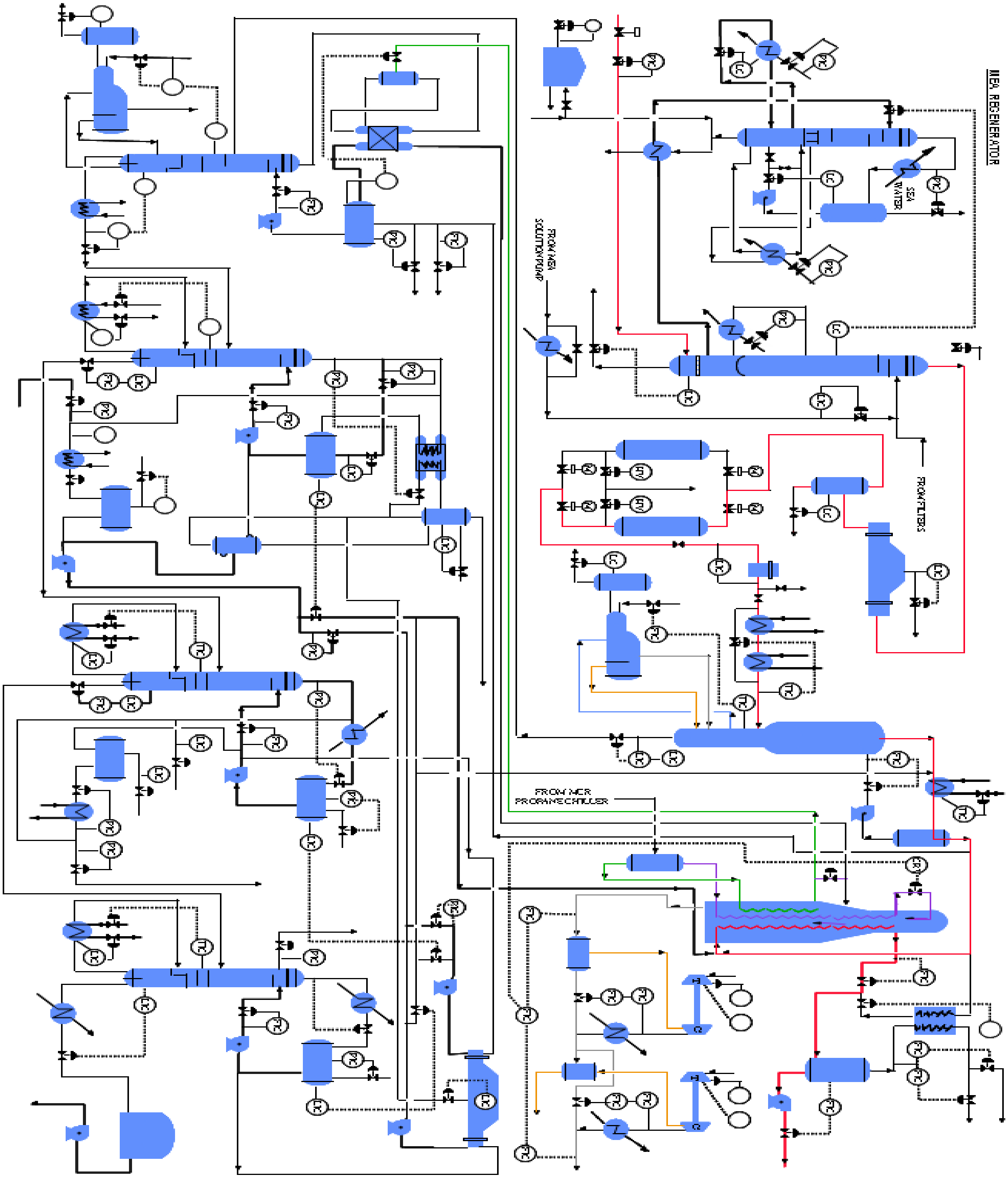

Fig I.2 General Diagram of the Liquefaction Process

I.3.1. Decarbonation Section

The function of this section is to eliminate the CO_2 contained in the natural gas, as CO_2, when brought to a low temperature around -70°C, solidifies and forms ice plugs that can obstruct service pipelines. Moreover, CO_2 exhibits a rather particular behavior regarding corrosion, where the partial pressure of CO_2 controls the corrosion phenomenon. For the decarbonation treatment, an amine called MEA (Monoethanolamine, $C_2H_5ONH_2$) is used. This amine has the ability to capture the CO_2 present in the natural gas.
This section involves two important steps

- Absorption
- Regeneration

This section will be detailed in Chapter II

I.3.2.DehydrationSection

After the decarbonation section, the natural gas is directed to a second section: dehydration. This section involves removing the water contained in the natural gas. For this, two molecular sieve dryers are used—one dryer operates in normal service while the second is in regeneration.

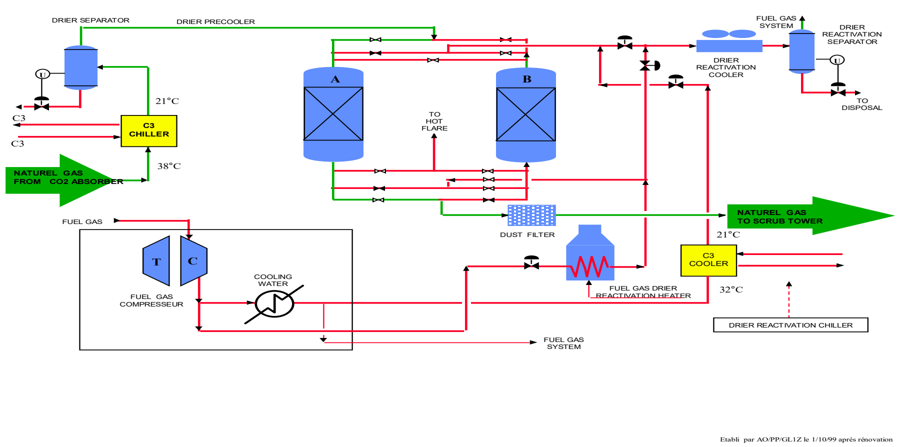

Fig I.3 Dehydration Section

I.3.3. Mercury Removal Section

The final treatment step is mercury removal. It consists of trapping the mercury present in the gas to prevent corrosion of aluminum equipment, such as the main exchanger in the liquefaction section. The natural gas passes through a sulfur-based activated carbon mercury removal unit and then through two filters. It exits this section with a mercury content lower than 7 nanograms per Nm^3 and is sent to the cooling section.

I.3.4. Cooling Section

Before introducing the natural gas into the separation and wash tower section, it undergoes **pre-cooling** to lower its temperature to -34°C using two exchangers: E522 and then E524. This pre-cooling is carried out in cryogenic exchangers commonly called **Schiller** exchangers. These are standard heat exchangers, but thermal insulation is critically important to avoid cold loss. Pre-cooling is performed using propane compressed by a turbocompressor K01.10. The natural gas is then directed into the wash tower after being mixed with less cold gas at a temperature of -26°C via TV442.

II.3.5. Separation and Wash Tower Section

Separation is a crucial step in the natural gas liquefaction cycle. After pre-cooling, the gas is sent to a **separation tower F711**. At this stage, the separation is a conventional distillation process that divides the product into two distinct phases: **heavy products** and **light products**.The heavy product is directed to the **fractionation section**, and the light product (gas) continues on to the **liquefaction section**.

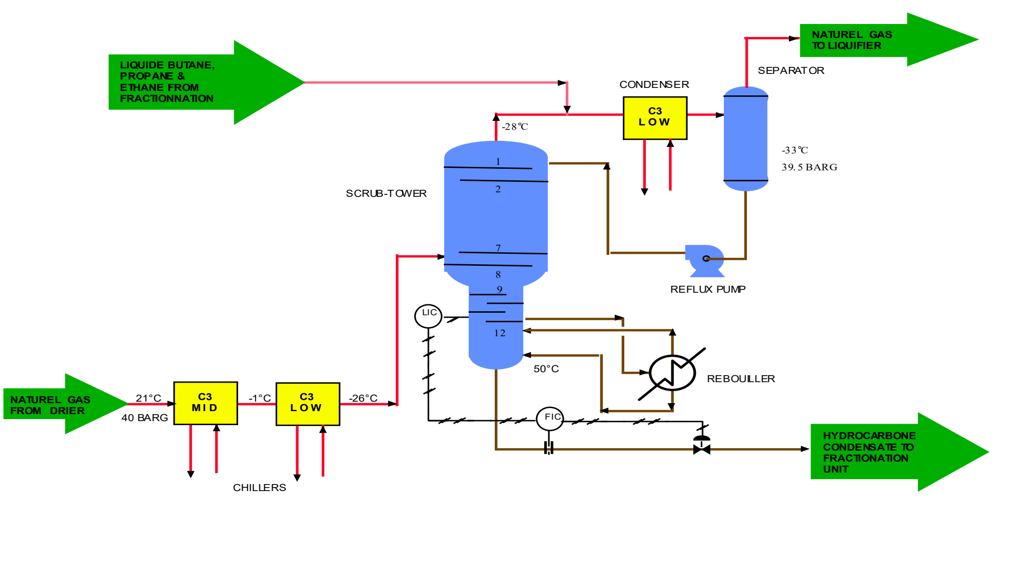

Fig I.4 Separation and Wash Tower Section

I.3.6. Fractionation Section

The purpose of this section is to fractionate the bottom product from the wash tower to supply the refrigeration and makeup circuits in the various process sections. This section consists of four distillation columns, each with a reboiler, a condenser, and a reflux drum.

I.3.6.1. De-ethnization (F721)

Light components, mainly methane, are separated from the heavier hydrocarbons and pass through the column counter-current to the reflux flow.
The top vapors are partially condensed in a condenser by the action of a refrigerant and then separated in the reflux drum. The non-condensed vapors provide an alternative methane makeup source for the MCR (Multi-Component Refrigerants) compression system.

I.3.6.2. De-ethnization (F731)

The heavy hydrocarbon flow coming from the de-methanization column is then fractionated in the Deethanizer to primarily produce ethane as the top product. The obtained ethane is used as a makeup for the MCR, the wash tower, and for LNG quality control—particularly for enhancing its Higher Heating Value (HHV).
Higher Heating Value (HHV): This is the thermal energy released by the combustion of one kilogram of fuel. It includes both the sensible heat and the latent heat of vaporization of the water, which is usually produced during combustion.

I.3.6.3. De-propanization (F741)

The De-propanization column is fed by a continuous hydrocarbon stream from the De-ethanization column. Its purpose is to produce propane used as a HHV makeup and as a supply for the propane refrigeration circuit.

I.3.6.4. Debutanization (F751)

The heavy fractions from the Depropanization column feed the final distillation column in the fractionation section. The bottom debutanized product is cooled and then sent to gasoline storage in a sphere, composed of C_5+ components. The butane produced is used for HHV makeup and boiler fuel supply.

Table I.2: Chemical Composition of the Multi-Component Refrigerant (MCR)

MCR Component	Component Percentage (%)
Nitrogen	3
Methane	40
Ethane	54
Propane	3

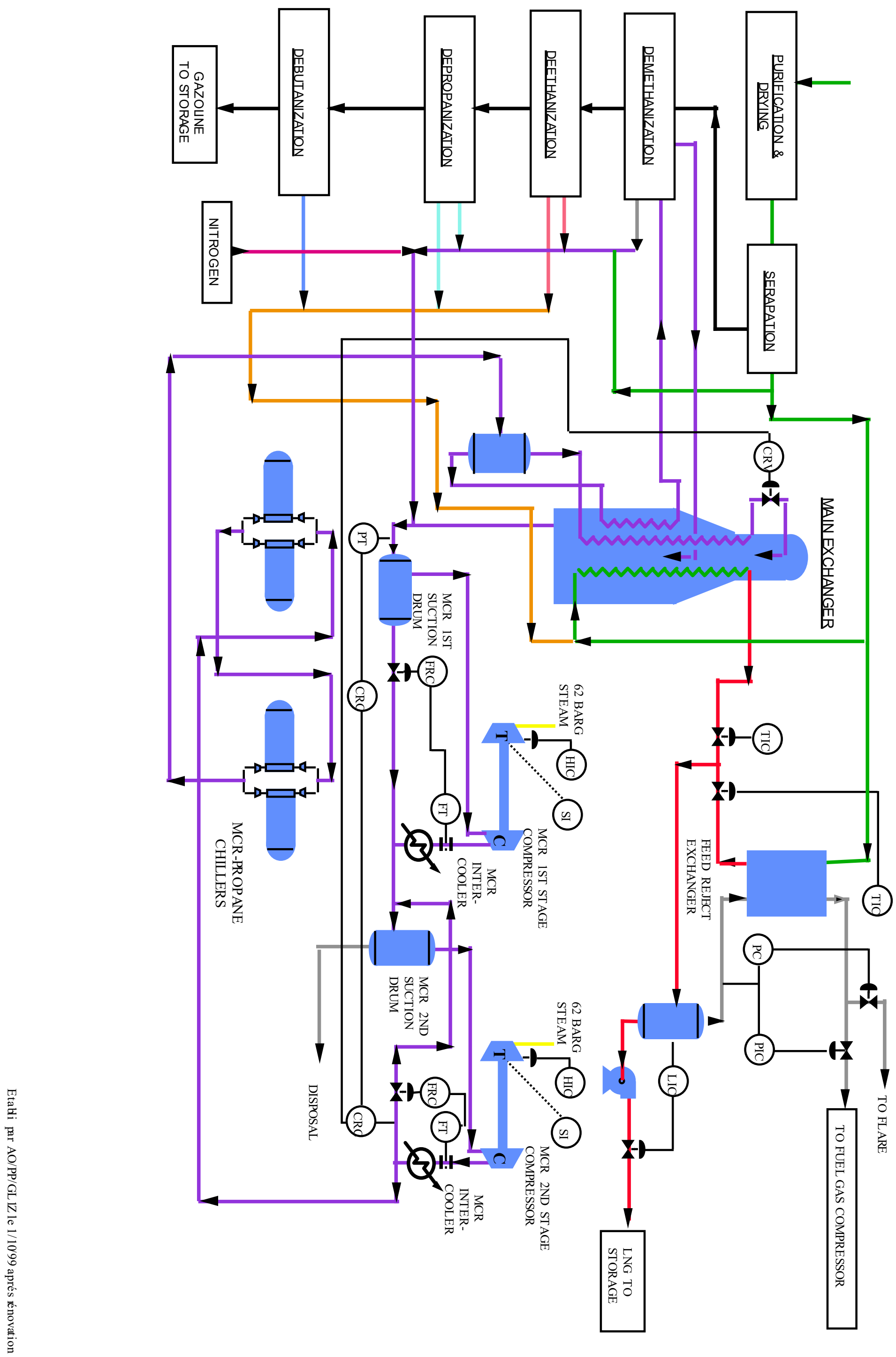

Fig I.5 Liquefaction Section

CHAPTER II

CHAPTER II: DECARBONATION AND PROBLEM OF Fouling E.504 HEAT Exchangers

II.1. Introduction

In natural gas processing, the removal of acid gases such as carbon dioxide (CO_2) is a critical step to ensure the safety, efficiency, and environmental compliance of the final product. This step, commonly referred to as decarbonation, relies heavily on the performance of amine-based absorption systems. Within the GL1/Z gas liquefaction complex, monoethanolamine (MEA) is used to absorb CO_2 from natural gas, followed by a regeneration phase where the CO_2 is stripped and the MEA is recycled.

A key element in this process is the E.504 heat exchanger, which serves to cool the regenerated lean MEA before it re-enters the absorber column. The effectiveness of this exchanger directly impacts the absorption performance, energy efficiency, and operational stability of the decarbonation unit.

However, over the years, the E.504 exchanger has faced persistent operational challenges due to fouling. This phenomenon, involving the buildup of unwanted materials such as scale, corrosion products, and biological matter, has significantly reduced the thermal efficiency of the unit. Fouling not only compromises the heat transfer performance but also leads to increased pressure drops, maintenance frequency, and operational costs. In severe cases, it contributes to environmental risks and production downtime.

This chapter explores the decarbonation process and provides an in-depth analysis of the fouling problems affecting the E.504 heat exchanger. It examines the types of fouling observed, their root causes, performance implications, and operational consequences, setting the stage for technical recommendations and design improvements in subsequent sections.

Table II.1. The Main Equipment in the Decarbonation Section

Equipment Name	Tag/Code	Function
Absorber Column	F-502	Removes CO_2 from the natural gas via absorption using MEA solution.
Regeneration Column	F-501	Regenerates CO_2-rich MEA by desorption under high temperature and low pressure.
MEA Flash Drum	G-507	Reduces pressure and separates liquid/vapor phases.
MEA Purifier	E-505	Cleans the MEA solution by removing contaminants.
MEA Heat Exchangers (Series)	E-503/A, B	Recover heat from the rich and lean MEA streams.
Activated Cartridge Filter	P-501	Filters fine solid particles from the MEA stream.
Cartridge Filter	P-502	Provides general filtration of the MEA.
Overhead Condenser	E-501	Condenses vapors at the top of the regenerator column.
MEA Reboiler	E-502	Supplies heat to strip CO_2 from the MEA solution in the regeneration column.
Acid Gas Drum	G-502	Collects the stripped CO_2 before disposal or further treatment.
Wash Pumps	J-510/511	Circulate wash liquid to the absorber's wash section.
Turbo MEA Pump	J-503	Ensures high-capacity circulation of MEA through the system.
MEA Pump	J-504	Pumps MEA to the absorber column.
Sump	G-504	Collects liquids draining from various parts of the system.

II.2. The Decarbonation Section

The purpose of the decarbonation section is to prevent system blockage caused by the solidification of CO_2 in the low-temperature sections downstream of the liquefaction unit.

CO_2 removal to a final concentration of 90 ppm is achieved via an absorption process in a high-pressure, low-temperature column (Absorber F-502), operating at 42 bar and 38°C. The column is equipped with 27 theoretical stages.

II.2.1 CO_2 Absorption Process

The feed natural gas enters the decarbonation unit at a controlled pressure of 42 bar. Upon entry, it passes through a rapid separation section located at the base of the absorber column F-502, where any entrained liquids are efficiently separated from the gas stream. These separated liquids are then routed to the hydrocarbon decanting system for further treatment.

After this initial separation, the dry gas is preheated via the heat exchanger E-506 before entering the lower section of the absorber column at a temperature of 38°C. The absorption process occurs within 27 valve trays, where the gas ascends counter-currently to the descending solvent.

The solvent used is a 15% aqueous solution of Monoethanolamine (MEA), specifically formulated for CO_2 capture. This lean MEA, containing a minimal amount of CO_2, is introduced at the top of the absorber at the same operating temperature of 38°C and at a slightly higher pressure than the incoming gas, ensuring optimal mass transfer conditions.

The MEA flow rate is precisely regulated at 54,000 kg/h, providing sufficient solvent volume to achieve the target CO_2 concentration reduction down to 90 ppm at the absorber outlet. As the natural gas rises, it comes into intimate contact with the MEA solution, allowing for effective absorption of carbon dioxide before being sent to the dehydration section for further purification.

II.2.2 MEA Solution Regeneration

The regeneration of monoethanolamine (MEA) is a critical step in maintaining the efficiency and continuity of the CO_2 removal process. After absorbing carbon dioxide in the absorber column (F-502), the rich MEA solution now laden with CO_2 is routed through a heat exchanger where it is preheated by thermal exchange with the lean MEA returning from the regeneration column. This energy recovery step not only improves process efficiency but also reduces the overall energy consumption of the system.

Once preheated, the rich MEA is introduced at the top of the regeneration column F-501, which operates at atmospheric pressure (approximately 1 bar) and elevated temperature (~110°C). Within this column, the absorbed CO_2 is separated from the MEA solution through a process of thermal stripping using steam. The upward flow of steam promotes desorption by shifting the chemical equilibrium, releasing CO_2 from the rich solution.

The regenerated lean MEA, now depleted of CO_2, is collected at the bottom of the column. Before being recycled to the absorber column, it is first used to preheat the incoming rich MEA (as mentioned above), then cooled down to the operating absorption temperature of approximately

38°C. This cooled, lean MEA is then reinjected into the absorber to begin a new cycle of CO_2 capture.

This continuous regeneration and reuse of MEA not only ensures consistent performance of the decarbonation process but also significantly minimizes solvent consumption and operational costs.

II.3 Characteristics of the Chemical Reaction

The removal of carbon dioxide from natural gas is achieved by counter-current washing in the absorber column using an aqueous solution of monoethanolamine (MEA), at a concentration of 15–20%.

The primary chemical reaction involved in CO_2 absorption with MEA is:

$$2RNH_2 + H_2O + CO_2 \rightleftharpoons (RNH_3)_2CO_3 + Q \qquad \text{(II.1)}$$

Where RNH_2 represents the MEA molecule and $R = C_2H_4OH$.

An intermediate step involves the formation of carbonic acid from water and carbon dioxide:

$$H_2O + CO_2 \rightleftharpoons 2H^+ + CO_3^{2-} \qquad \text{(II.2)}$$

The reaction equilibrium (Equation II.1) favors absorption at high pressure and low temperature, while the reverse reaction (desorption) is promoted under low pressure and high temperature conditions typically used in the MEA regeneration process.

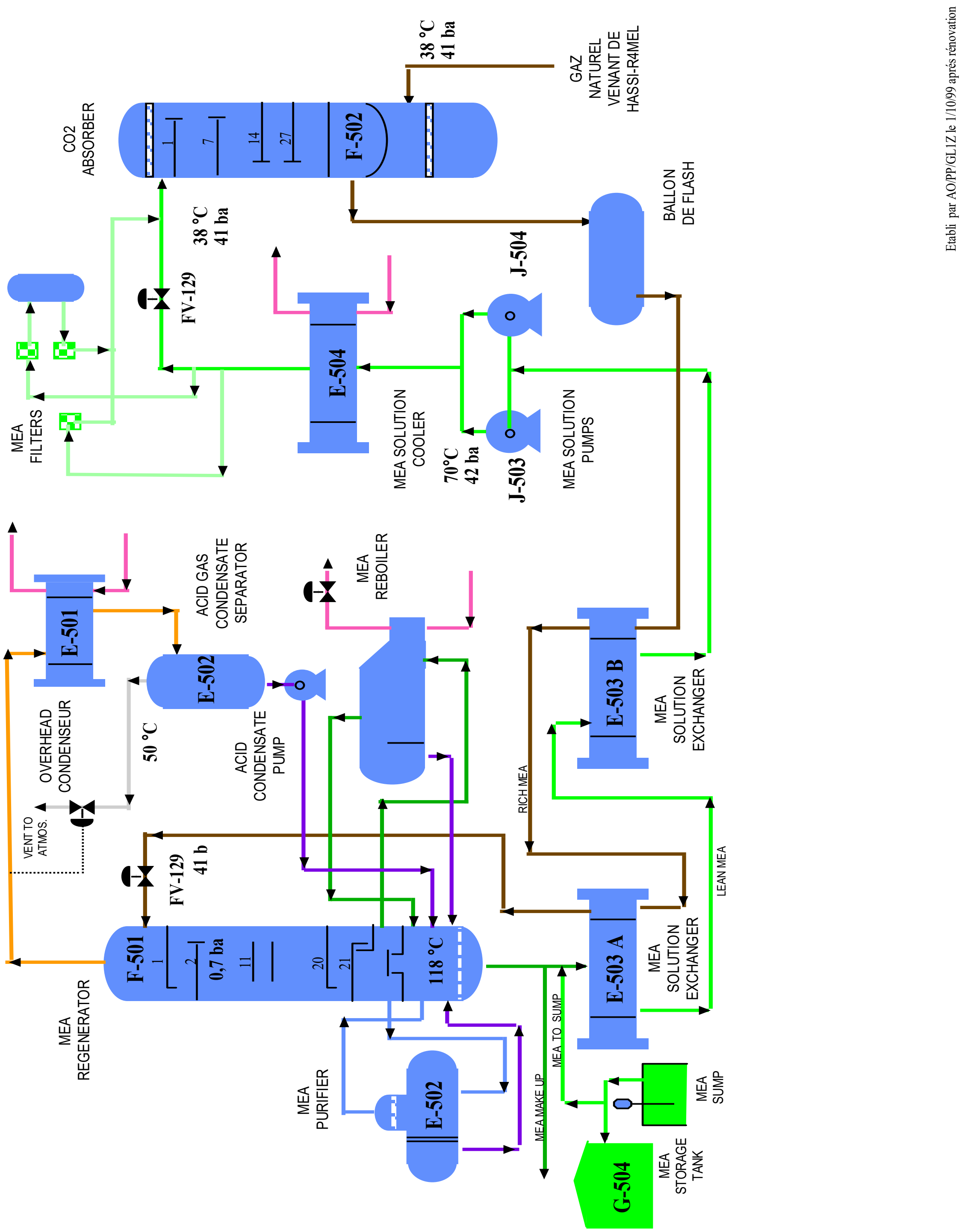

Fig.II.1: Schematic Diagram of the Decarbonation Section

II.4 Functional Role of E.504

CHAPTER II: DECARBONATION AND PROBLEM OF Fouling E.504 HEAT Exchangers

The E.504 heat exchanger plays a critical role in the amine-based gas treatment process at the GL1/Z liquefaction complex. Specifically, it is part of the MEA regeneration loop, where its main function is to cool the lean MEA solution before it returns to the absorber column for further CO_2 removal.

After completing the regeneration phase, the lean MEA exits the stripper column at high temperatures typically around 72 to 73°C. Before re-entering the absorber, this solution must be cooled to approximately 38°C to optimize the absorption of acidic gases. E.504 achieves this by using seawater as the cooling medium, operating as a shell-and-tube exchanger in a cross-flow configuration:

- Lean MEA flows through the shell side
- Seawater circulates through the tube side

This cooling stage is essential for several reasons:

- It maintains optimal absorption efficiency in the absorber column
- It prevents thermal degradation of MEA, which would otherwise reduce solvent life and increase corrosion
- It ensures thermal balance across the gas treatment system
- It contributes to overall process energy efficiency

E.504 must operate reliably under demanding conditions high pressure on the MEA side, corrosive seawater on the cooling side, and exposure to variable flow and temperature conditions. Its performance directly affects gas treatment efficiency, MEA solvent integrity, and overall plant stability. Any decline in its heat exchange capability can lead to increased energy consumption, reduced gas purity, and costly shutdowns.

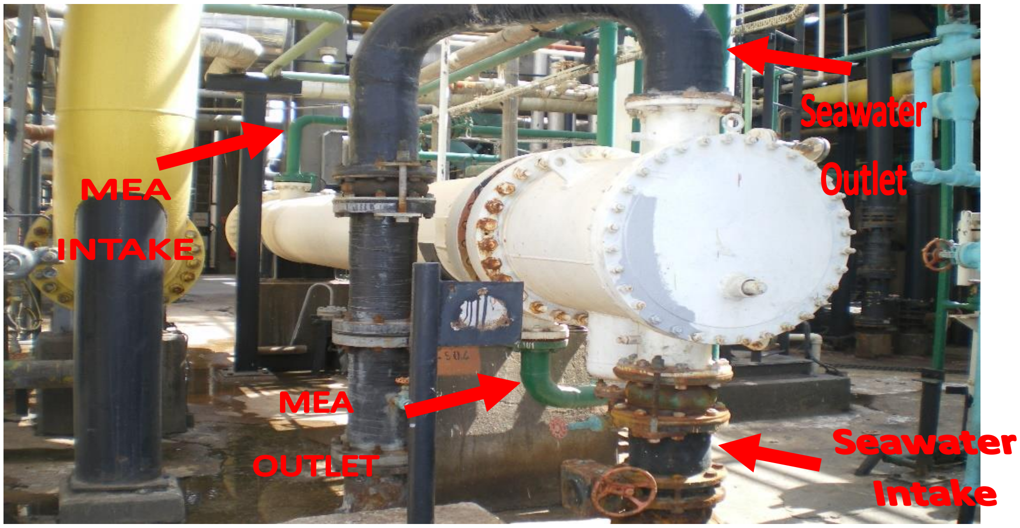

Fig.II.2 : Position of the E.504 heat exchanger in the decarbonation section.

II.4.1. Symptoms of Fouling in E.504

Over years of continuous operation, the E.504 heat exchanger has shown clear signs of performance degradation, primarily due to the progressive effects of fouling. These symptoms became increasingly evident in both operational data and physical inspections, signaling the need for corrective actions.

The key symptoms observed include:

II.4.1.1. Elevated Outlet Temperature of MEA

One of the earliest and most critical indicators of fouling was the rise in the outlet temperature of the lean MEA. Designed to cool the MEA down to 38°C, the exchanger at times delivered the solution at temperatures as high as 50°C, particularly after 2002. This deviation from design values suggests a decline in thermal performance, likely due to an increase in thermal resistance caused by fouling deposits.

Table II.2. Operating Parameters of the E.504 Heat Exchanger (Current)

	MEA (Shell Side)	Sea water (Tube Side)
Inlet Temperature (°C)	72.9	25
Outlet Temperature (°C)	43	38
Flow Rate (Kg/h)	42780	103079.6
Inlet Pressure (bar)	63	3.5

II.4.1.2. Reduction in Seawater Flow Efficiency

Operational records show a decline in the seawater mass flow rate through the tube side. The design flow of around 147,000 kg/h dropped to approximately 103,000 kg/h, indicating partial blockage of flow channels. This was further supported by a reduction in pressure drop across the exchanger from the design value of 0.689 bar to 0.35 bar a sign that resistance to flow had changed, likely due to internal obstructions.

II.4.1.3. Increased Differential Pressure and Local Blockages

Routine inspections revealed localized blockages at critical points:

- Up to 80% clogging of seawater distributor box strainers by marine debris
- Partial blockage of orifice plates, further limiting flow
- Accumulation of tartareous and gelatinous materials in internal surfaces and headers

These blockages not only restricted flow but also contributed to uneven heat distribution and increased mechanical stress on the exchanger.

Fig.II.3: Tube Blockage in the E.504 Heat Exchanger

II.4.1.4. Visual Signs of Corrosion and Deposits

Maintenance interventions consistently reported:

- Visible scaling on tube sheets and inner walls (mainly calcium carbonate)
- Pitting and metal loss on tube surfaces, particularly on the MEA side
- Gel-like biological layers, possibly from seawater bioactivity and insufficient chlorination

These physical indicators confirmed the presence of multiple fouling mechanisms, including scaling, corrosion, biofouling, and particulate fouling

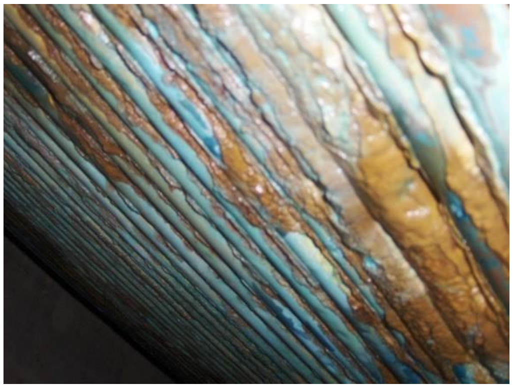

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Fig.II.4 : Effect of MEA on the Tube Surface

II.4.1.5. Performance Instability and Maintenance Frequency

The exchanger required frequent maintenance, especially between 1998 and 2008, with interventions to remove deposits, replace tubes, and address corrosion damage. These interventions were time-consuming and costly, resulting in reduced availability of the unit and occasional train shutdowns.

Together, these symptoms strongly indicated a progressive decline in exchanger performance directly attributable to fouling. In the following sections, we will examine the specific types of fouling that affected E.504 and analyze their root causes and consequences.

CHAPTER III

III.1. Introduction

Heat exchangers are fundamental components in a vast array of industrial processes, facilitating the transfer of thermal energy between two or more fluids at different temperatures. Among the diverse types of heat exchangers, air coolers (also known as fin-fan coolers, air-cooled heat exchangers, or aero-refrigerants) play a critical role, particularly in environments where water resources are scarce, expensive, or subject to strict environmental regulations. Unlike traditional shell-and-tube or plate heat exchangers that often rely on water as a cooling medium, air coolers leverage ambient air, making them a highly versatile and often preferred solution for dissipating heat.

The increasing global focus on water conservation, coupled with the rising costs associated with water treatment and discharge, has further propelled the adoption of air cooling technology across various sectors, including petrochemicals, power generation, oil and gas, and refrigeration. This chapter will delve into the principles, design considerations, operational aspects, and emerging innovations surrounding air cooler technology, highlighting its advantages, limitations, and its indispensable role in modern industrial thermal management.

III.2. Basic Working Principle

Air coolers, also known as air-cooled heat exchangers (ACHEs), operate based on the principle of forced convection, wherein heat is transferred from a process fluid (usually a hot liquid or gas) flowing inside the tubes to ambient air circulating over the external surface of the tubes. This process eliminates the need for water-based cooling systems, making air coolers a preferred choice in areas with limited water availability or where environmental concerns demand dry cooling solutions.

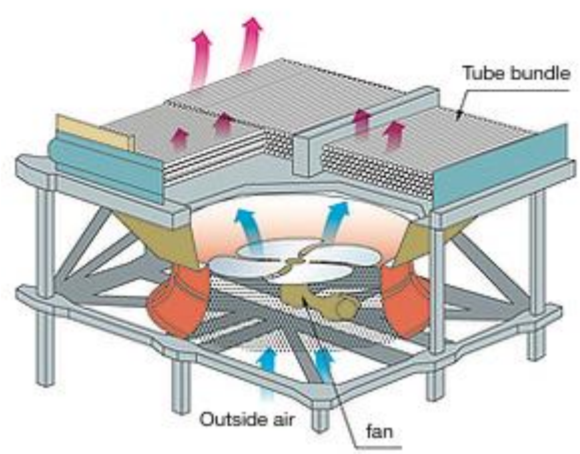

Fig.III.1: air-cooled heat exchangers (ACHEs)

III.2.1 Mechanism of Heat Transfer

The fundamental principle governing air coolers is the transfer of thermal energy from the hot process fluid to the ambient air via conduction and convection:

- **Conduction** occurs through the tube walls, where heat passes from the inner surface (in contact with the hot fluid) to the outer surface (in contact with air).
- **Convection** takes place on both sides:
  - **Internal convection** between the fluid and the inner wall of the tube.
  - **External forced convection** where ambient air, moved by fans, absorbs the heat from the outer surface of the tube.

The overall heat transfer rate (Q) is governed by the equation:

$$\boldsymbol{Q = \alpha S \Delta T_{Lm}} \qquad \text{(III.1)}$$

Where:

- $\boldsymbol{Q}$: Rate of heat transfer (W)
- $\boldsymbol{\alpha}$: Overall heat transfer coefficient ($W/m^2 \cdot K$)
- S: Effective surface area for heat exchange (m^2)
- $\boldsymbol{\Delta T_{Lm}}$: Log Mean Temperature Difference (K)

This formula highlights the importance of both material properties and design parameters in maximizing performance.

III.2.2. Hot Fluid Inside Tubes, Ambient Air Outside

In a typical air cooler:

- The hot process fluid (e.g., hydrocarbon gas, steam, or chemical vapor) enters the tube bundle and flows through the internally hollow tubes.
- The tubes are externally finned (to increase surface area) and exposed to ambient air blown or drawn across them by large industrial fans.
- As the hot fluid flows inside the tubes, it loses heat to the tube walls. This heat is then transferred to the cooler ambient air outside the tubes.
- The now cooled fluid exits the exchanger at the outlet header.

The design ensures that there is no direct contact between the process fluid and air, preserving process purity and enabling pressure containment.

III.2.3 Role of Fans (Forced or Induced Draft)

Fans are critical components that enable forced convection, which significantly enhances the rate of heat transfer by continuously supplying cool air to absorb the heat from the tubes. Depending on their configuration, air coolers are classified into:

- **Forced Draft Coolers:**
  - Fans are located below the tube bundle.
  - They push air upward through the tubes.
  - This setup protects fans from hot discharge air but may suffer from recirculation of heated air in windy conditions.
- **Induced Draft Coolers:**
  - Fans are mounted above the tube bundle.
  - They draw air through the tubes and exhaust it vertically.
  - This arrangement minimizes hot air recirculation, improves air distribution, and ensures more uniform cooling.
  - However, fans are exposed to hot outlet air, which may affect motor longevity and require special insulation or remote motor placement.

Fan speed, blade design, and placement are carefully engineered to ensure optimal airflow and energy efficiency while limiting noise and vibration.

III.2.4 Importance of Surface Area and Material Conductivity

Two key parameters that significantly affect the thermal performance of an air cooler are:

Surface Area (S):

- The larger the surface area in contact with air, the greater the potential for heat dissipation.
- Extended surfaces, such as fins, are added to the tubes to increase external surface area without enlarging the cooler footprint.
- Fin configurations (e.g., helical, plate, extruded) are selected based on fluid type, fouling tendencies, and maintenance access.

Thermal Conductivity of Materials:

- The thermal conductivity (k) of tube and fin materials directly influences the heat transfer efficiency.
- Aluminum and copper have high thermal conductivities and are often used for fins, while carbon steel, stainless steel, and copper-nickel alloys are used for tubes depending on corrosion resistance, cost, and pressure requirements.
- Material selection balances thermal performance, durability, and economic feasibility.

III.3. Types of Air Coolers

Air coolers, or air-cooled heat exchangers (ACHEs), come in various configurations based on fan arrangement, airflow direction, and cooling mechanisms. The selection of air cooler type depends on factors such as thermal performance requirements, site conditions (temperature, humidity, wind), space availability, energy consumption, and process criticality.

III.3.1 Forced Draft vs. Induced Draft

III.3.1.1 Forced Draft Air Coolers

- **Configuration:** Fans are installed below the tube bundle and push air upward through the finned tubes.
- **Advantages:**
  - Fans are exposed to cool ambient air, resulting in longer fan motor life and less need for insulation.
  - Easier maintenance due to ground-level fan placement.
- **Disadvantages:**
  - Susceptible to hot air recirculation in windy conditions, especially if layout and spacing are not optimized.
  - Non-uniform air distribution in large bundles may occur.
- **Applications:** Common in small to medium-sized installations and where motor protection from high temperatures is critical.

III.3.1.2 Induced Draft Air Coolers

- **Configuration:** Fans are placed above the tube bundle and draw air upward through the tubes before discharging it to the atmosphere.
- **Advantages:**
  - **Minimizes recirculation of hot air, especially in congested or high-temperature areas.**
  - **Promotes uniform airflow through the bundle.**
- **Disadvantages:**
  - Fans operate in a hot air stream, which may require heat-resistant materials or remote-drive fan systems.
  - Maintenance can be more complex due to fan elevation.
- **Applications:** Preferred for high-performance and high-temperature processes where thermal efficiency is a priority.

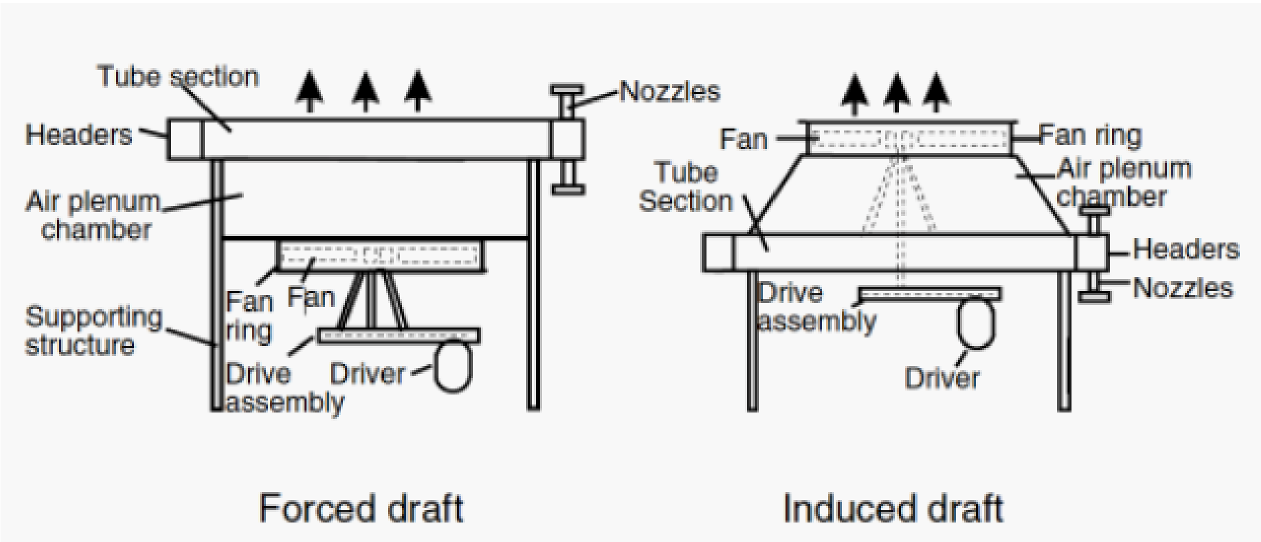

Fig.III.2: Forced Draft vs. Induced Draft

III.3.2 Vertical vs. Horizontal Layout

III.3.2.1 Horizontal Layout (Most Common)

- Tube bundles are arranged horizontally, and air flows vertically through the bundle (either upward or downward).
- Fan position determines if it's forced or induced draft.
- **Advantages:**
  - Easier to access for cleaning and maintenance.
  - Compatible with large-scale industrial installations.
  - Stable structure with less wind sensitivity.
- **Disadvantages:**
  - Requires more ground area (footprint).
- Applications: Widely used in oil refineries, gas processing plants, and petrochemical facilities.

III.3.2.2 Vertical Layout

- Tube bundles are mounted vertically, and air flows horizontally.
- Fans can be located at the side or behind the bundles.
- **Advantages:**
  - Space-saving, ideal for congested sites or offshore platforms.
  - Reduced footprint and better protection from wind in narrow installations.
- **Disadvantages:**
  - More challenging maintenance due to vertical orientation.
  - Uneven air distribution is more difficult to control.
- **Applications:** Often found in modular units, portable skid-mounted systems, and marine/offshore environments.

III.3.3 Dry Air Coolers vs. Adiabatic Coolers

III.3.3.1 Dry Air Coolers

- Rely entirely on ambient air for cooling — no water used.
- 100% dry operation, making them ideal in water-scarce regions.
- Environmentally sustainable and low-maintenance.
- Performance can drop significantly in hot climates, as cooling capacity depends on the ambient dry bulb temperature.
- Applications: General-purpose cooling where water use is undesirable or restricted.

III.3.3.2 Adiabatic Coolers (Hybrid Air Coolers)

- Use a water spray or wetting pad system to cool the incoming air before it reaches the finned tubes.
- Reduces the dry bulb temperature of the air, mimicking evaporative cooling and increasing heat transfer efficiency.
- Operate in dry mode most of the time, and switch to wet mode during high ambient temperatures.
- **Advantages:**
    - Up to 20-40% improved cooling performance in hot weather.
    - Water consumption is lower than in traditional wet cooling towers.
- **Disadvantages:**
    - Slightly more complex due to water handling, filtration, and scaling risks.
- **Applications:** Power plants, data centers, and industries in hot, arid regions requiring consistent cooling.

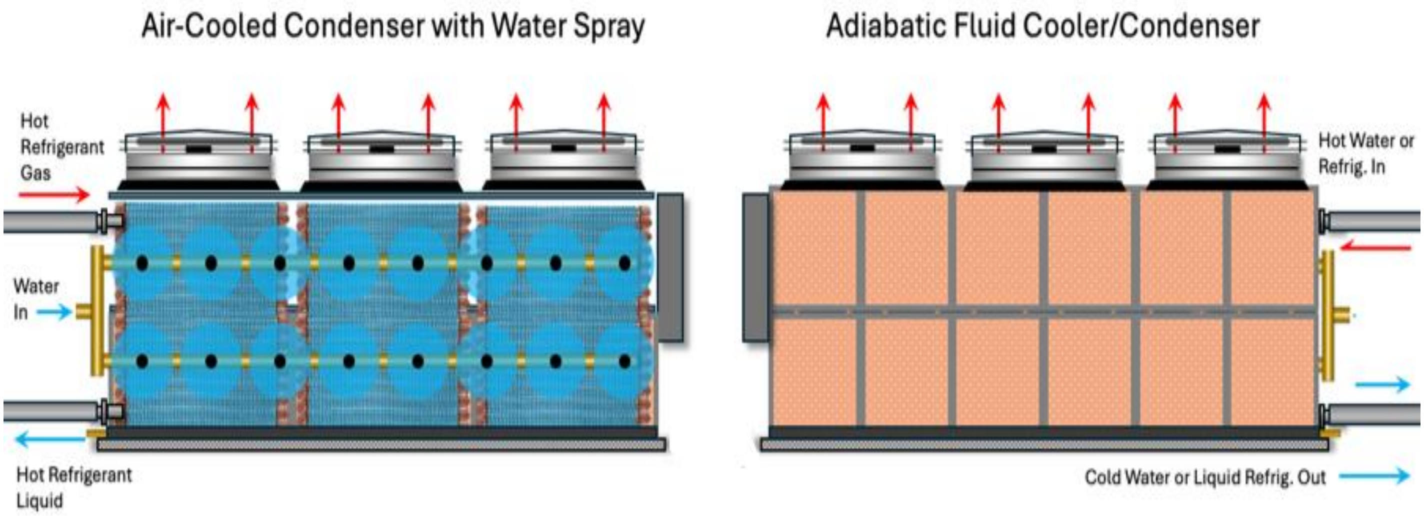

Fig.III.3: Adiabatic Coolers

III.4.Construction and Components

III.4.1 Tube Bundle

The tube bundle is the heart of an air-cooled heat exchanger. It defines the heat transfer performance and the physical limits of the unit. Its design must balance thermal efficiency, mechanical strength, and cleanability.

- Material Selection:
  - Carbon Steel: Cost-effective, but susceptible to corrosion.
  - Stainless Steel (304/316): Corrosion-resistant, ideal for aggressive fluids.
  - Aluminum-Brass or Cu-Ni: For seawater or corrosive environments.
- Tube Diameter:
  - Common outer diameters: 19 mm, 25 mm, or 38 mm.
  - Smaller diameters improve heat transfer but increase pressure drop.
- Wall Thickness (BWG):

- Balanced between heat transfer and mechanical strength.[1]
- Thinner walls improve thermal conduction but reduce pressure tolerance.

Fig.III.4: Tube Bundle

III.4.2. Fin Design – Surface Area Multiplier

- Purpose: Increase external surface area to enhance air-side heat transfer.
- Materials: Mostly aluminum, sometimes copper (for higher conductivity).
- Fin Types:

a. Extruded Fins

Extruded fins represent the most thermally efficient fin type due to the monolithic integration between the fin material and the base tube. These fins are manufactured by forcing a composite tube comprising a core tube (typically carbon steel, stainless steel, or copper) encased in an outer aluminum sheath through an extrusion die, forming integral fins from the outer layer of the material.

Advantages:

- The intimate metallurgical bond between the tube and fin ensures minimal thermal resistance, resulting in superior heat transfer performance.
- The absence of crevices or gaps between the fin and tube provides excellent resistance to corrosion, particularly in aggressive environments such as coastal, offshore, or chemically reactive atmospheres.

[1] **API Standard 661**, *Air-Cooled Heat Exchangers for General Refinery Service*, American Petroleum Institute, 8th Edition, 2013

- The mechanical strength and rigidity of extruded fins make them highly resistant to vibration and physical damage.

Limitations:

- The manufacturing process is relatively complex and costly.
- Applicable only to specific material combinations due to the nature of the extrusion process.

Extruded fins are preferred in applications where durability and heat transfer efficiency are critical, such as in petrochemical plants, offshore oil platforms, and high-salinity environments.

Fig.III.5 Extruded Fins

b. Embedded Fins (G-Type)

Embedded fins, also known as grooved or G-type fins, are formed by helically winding an aluminum or copper strip into a pre-machined groove on the outer surface of the tube. The displaced tube material is then mechanically deformed over the base of the fin to secure it in place, forming a tight mechanical bond.

Advantages:

- This configuration provides a relatively strong mechanical attachment with enhanced thermal contact, making it suitable for a wide range of industrial applications.
- The process allows the use of various tube materials, making it adaptable to different service conditions.
- Offers a balance between performance and cost, making it an efficient choice for non-corrosive environments.

Limitations:

- Over time, mechanical stresses and thermal cycling may degrade the bond between the fin and tube.
- It exhibits lower corrosion resistance compared to extruded fins due to partial exposure of the tube wall.

Embedded fins are typically used in general-purpose air coolers operating under moderate thermal and environmental conditions, such as refineries, power generation units, and gas processing plants.

Fig.III.6 Embedded Fins (G-Type)

c. **L-Fins and LL-Fins**

L-type and LL-type fins are classified as mechanically wrapped fins. In the L-fin configuration, the aluminum or copper fin strip is helically wound around the base tube with an L-shaped foot to maintain contact. The LL-fin is a modified version wherein the fins are overlapped to completely enclose the tube, thereby offering marginally improved corrosion protection.

Advantages:

- These fins are cost-effective and allow high manufacturing throughput.
- Suitable for applications where cost constraints are significant and environmental conditions are not severe.

Limitations:

- The thermal contact between the fin and tube is solely maintained by mechanical pressure, resulting in reduced heat transfer efficiency.

- These fins are more susceptible to loosening under mechanical vibration or thermal cycling.
- Their mechanical and corrosion resistance is significantly lower compared to extruded or embedded fins.

As a result, L and LL-type fins are predominantly employed in HVAC systems, mobile heat exchangers, and non-critical industrial applications where budget constraints outweigh performance demands.

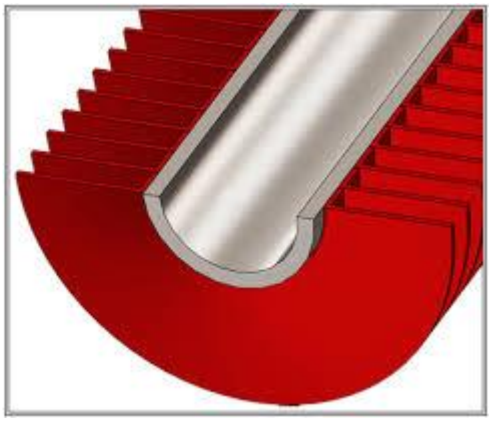

Fig.III.7: L-Fins and LL-Fins

- Fin Density:
  - Measured in fins per inch (FPI).
  - Higher FPI = greater surface area, but increases air-side pressure drop and fouling risk.
- Fin Height and Thickness:
  - Typical height: 10–20 mm.
  - Thicker fins last longer in harsh environments but slightly reduce heat transfer rate.

III.4.3 Fans and Drive System

The fan system is what makes an air cooler "air-cooled." Its role is to force or pull ambient air across the finned tubes to enable heat rejection.

- ✓ Axial Fans (most common):
  - High airflow, low pressure.

  - Diameters up to 10 meters in large exchangers.
  - Efficiencies typically between 60% to 85%.
  - Blade count: usually 4–10 blades.

- ✓ Centrifugal Fans (rare in ACHEs):
  - Used when static pressure is high (e.g., ducted systems).
  - Lower flow rates than axial but higher pressure.

Fan blades are typically manufactured from:

- Aluminum alloys: Lightweight and corrosion-resistant, suitable for general industrial applications.
- Glass-reinforced plastic (GRP): Offers superior corrosion resistance, particularly in corrosive or marine environments.
- Advanced composites: These materials combine low weight with high mechanical strength, making them suitable for large-diameter fans (often exceeding 9 meters or 30 feet in diameter), commonly used in power and petrochemical facilities.

Fan blade design especially pitch, curvature, and number of blades is optimized for maximum airflow efficiency while minimizing energy consumption and mechanical stress.

III.4.4 Headers (Manifolds)

The headers, also known as manifolds, are critical components located at both ends of the tube bundle in an air-cooled heat exchanger. Their principal function is to distribute the process fluid uniformly into the individual tubes during entry and to collect the fluid after it has undergone heat exchange with ambient air.

There are several header configurations, each selected based on operational, maintenance, and cost considerations:

- Plug-Type Headers: These are designed with individual access plugs for each tube row, allowing straightforward inspection, mechanical cleaning, and maintenance. This type is particularly advantageous in services where fouling or scaling is common, or where frequent access to the tube interiors is necessary.

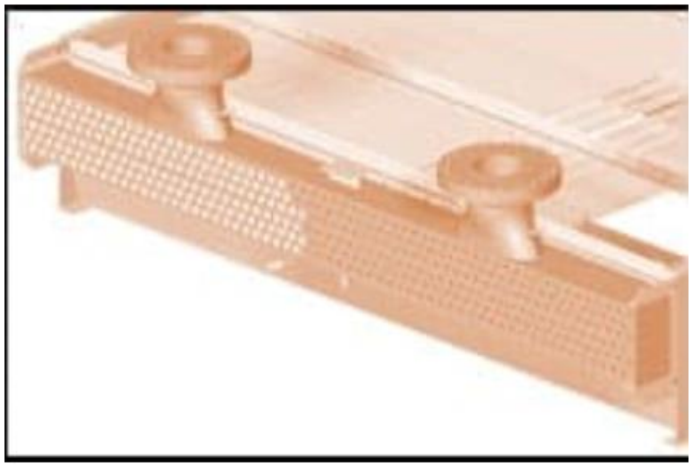

Fig.III.8: Plug-Type Headers

- Welded Bonnet Headers: In this design, the header is sealed with a welded cap, offering a compact and leak-resistant solution, ideal for applications with space constraints or where minimal maintenance is expected. However, they are less accessible, and any internal inspection requires complete removal or cutting, limiting their suitability in high-maintenance environments.

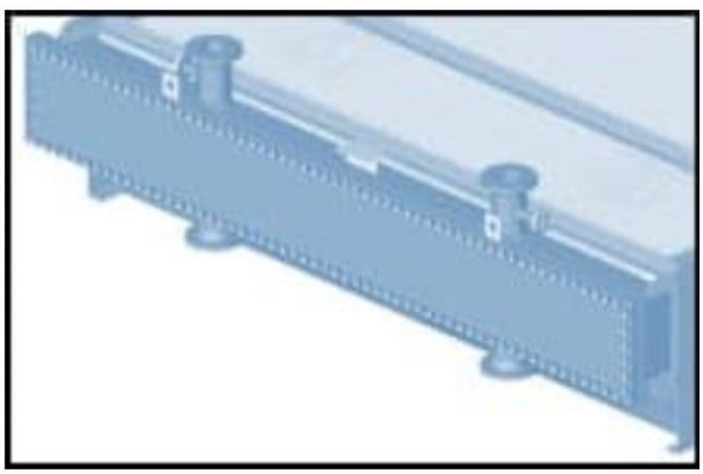

Fig.III.9: Welded Bonnet Headers

- Removable Cover Headers: These include bolted flanges that allow the entire end cover to be removed. This configuration offers easy access for inspection, maintenance, and cleaning without disturbing the tube bundle. They are widely used in applications where periodic servicing is required.

Fig.III.10: Removable Cover Headers

III.4.5 Fans and Drive System

The fan and drive system is a core element in the functioning of an air-cooled heat exchanger, responsible for providing the necessary airflow across the tube bundle to facilitate forced convection heat transfer. Depending on the application, different drive configurations may be employed:

- Electric Motors: This is the most commonly used drive system due to its simplicity, reliability, and ease of control. Electric motors can be either fixed-speed or variable-speed (via Variable Frequency Drives), allowing for flexibility in thermal load management.
- Hydraulic Drives: These systems are particularly advantageous in variable-speed applications and environments requiring precise speed modulation, such as in extreme ambient conditions. Hydraulic drives also offer the benefit of remote control and positioning, especially in hazardous or space-restricted areas.

Transmission Modes

- Belt Drive Systems: These allow for a degree of mechanical isolation between the motor and the fan, reducing vibrations transmitted to the motor. Belt drives are also cost-effective and offer flexibility in adjusting fan speed by changing pulley ratios. However, they require regular maintenance and tension checks.
- Direct Drive Systems: In this configuration, the motor is coupled directly to the fan shaft, offering higher mechanical efficiency, lower maintenance, and reduced energy losses. Direct drives are favored in high-performance systems where continuous operation and efficiency are critical.

III.4.6 Support Structure

The support structure forms the mechanical backbone of the air cooler unit. Constructed primarily from structural steel, it provides the necessary framework to support the tube bundles, headers, fans, motors, and associated instrumentation.

The design of the support structure must take into account several mechanical and environmental factors, including:

- Wind loads and seismic forces, especially for installations in exposed or high-risk areas.
- Vibrational stresses generated by fan rotation and fluid dynamics.
- Thermal expansion and contraction, which must be accommodated without imposing excessive stress on the tubes or connections.

Moreover, the structural design must ensure easy accessibility for maintenance personnel, compliance with local safety codes, and resistance to corrosion through surface treatment or material selection.

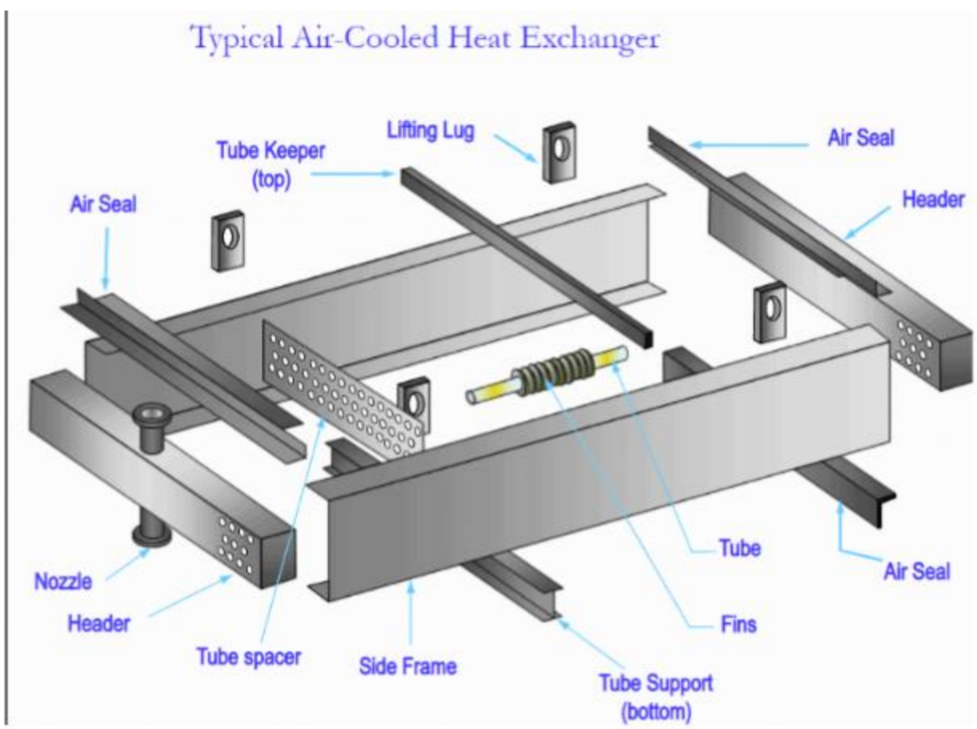

Fig.III.11: Structural Components of a Typical Air-Cooled Heat Exchanger

CHAPTER IV

IV.1. Introduction

The design of an air cooled heat exchanger entails the evaluation and determination of the amount of thermal energy that must be removed from a hot process fluid to meet a specified outlet temperature. This requirement is often dictated by the operational constraints of downstream equipment, such as gas turbines or processing units.

The core objective in such a design process is to maximize the heat transfer rate while minimizing both the heat transfer surface area and the associated pressure losses. Achieving this balance is essential to ensuring optimal capital expenditure and reduced operational costs.

Although thermal performance is the principal consideration, a range of additional criteria must be taken into account to ensure an effective and reliable design. These include:

- Allowable pressure drops on the fluid sides;
- Susceptibility to fouling and maintenance requirements;
- Physical space constraints and equipment footprint;
- Maximum permissible wall temperatures to avoid material degradation;
- The thermophysical properties of the working fluids, especially density;
- The nature and suitability of construction materials.

These factors collectively influence the selection and sizing of the heat exchanger and must be carefully integrated into the design methodology to ensure both performance and durability under actual operating conditions.

IV.2. Design Methodology of the Air-Cooled Heat Exchanger

The design of an air-cooled heat exchanger can be approached using two primary thermal calculation methods, each depending on the available design data and project objectives:

- **Thermal Sizing Approach**:
  This method focuses on determining the required heat transfer surface area based on known inlet and outlet temperatures, fluid properties, and mass flow rates. It is typically used during the preliminary design phase to define the geometric and thermal characteristics of the exchanger.
- **Performance Evaluation Approach**:

In contrast, this method assumes the geometry and surface area of the exchanger are already known. The analysis then focuses on predicting the outlet temperatures and flow conditions, thereby evaluating the exchanger's ability to meet process requirements.[1]

In this study, the objective is to design an air-cooled heat exchanger intended to cool a monoethanolamine (MEA) solution. This solution is used to absorb carbon dioxide (CO_2) from natural gas as part of the gas sweetening process in the decarbonation section. The cooling medium is a stream of ambient air that flows across the heat exchanger.

To complete the design process, several intermediate calculations are required. These include the determination of the air outlet temperature (T_2), the logarithmic mean temperature difference (LMTD), and the total number of tubes (n_T) in the exchanger.The type of air-cooled heat exchanger under consideration is a **dry-type unit with forced draft configuration**. In this setup, the hot MEA solution flows inside finned tubes, while cooling air is drawn across the tube bundle by mechanical fans to facilitate heat removal. A schematic representation of a finned tube is provided in Figure IV.1, and the thermophysical properties of both fluids are summarized in Table IV.2.

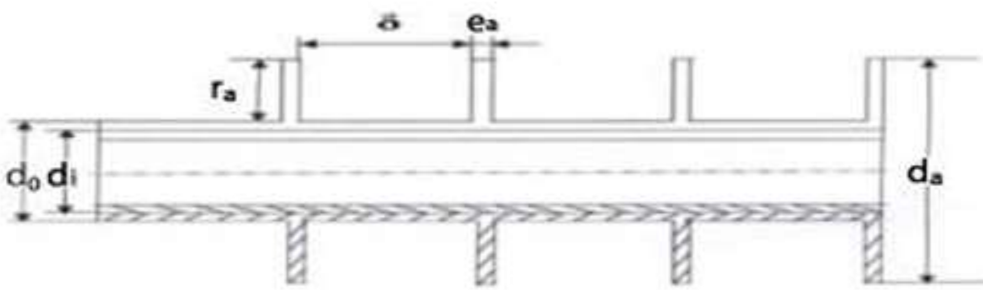

Fig.IV.1: Schematic Diagram of a Finned Tube

[1] **Kuppan, T.**, *Heat Exchanger Design Handbook*, CRC Press, 2nd Edition, 2013.

Table IV.1: Symbol Diagram of a Finned Tube

Symbol	Description
d_i	Inside diameter of the bare tube
d_0	Outside diameter of the bare tube
d_a	Outside diameter of the finned tube (including fins)
e_a	Thickness of the fin
δ	Fin pitch (distance between two successive fins)
R_a	Radius from the tube center to the base of the fin

Table IV.2: Characteristics of the Finned Tube

Property	Tube	Fin
Standard	BWG14	
Material	Steel	Aluminum
Thermal conductivity (kcal/h·m·°C)	43.172	–
Type	–	Integrated G-type
Number per meter	–	433
Outer diameter (mm)	25.4	57
Inner diameter (mm)	21.2	–
Thickness (mm)	2.1	0.3556

Table IV.3: Properties of the Two Fluids

Property	Hot Fluid (MEA Solution)	Cooling Fluid (Air)
Mass flow rate (kg/h)	48,000	–
Heat duty (kcal/h)	1.30064×10^6	–
Inlet temperature (°C)	73	30
Outlet temperature (°C)	46	–
Design pressure (bar)	10	–
Allowable pressure drop (bar)	0.5	–
Density at inlet (kg/m³)	981.4	–
Density at outlet (kg/m³)	998	–
Dynamic viscosity at inlet (cP)	0.4505	0.0186
Dynamic viscosity at outlet (cP)	0.8929	–
Specific heat capacity (J/kg·°C)	4199	1005
Thermal conductivity at inlet (kcal/h·m·°C)	0.5955	0.022966
Thermal conductivity at outlet (kcal/h·m·°C)	0.5640	–
Minimum temperature (°C)	–	-1
Altitude (m)	–	0

IV.3. Thermal Analysis

The initial step in the design of an air-cooled heat exchanger involves determining the overall heat transfer coefficient. This requires an estimation of the convective heat transfer coefficients on both the air side and the process fluid side (MEA solution). These coefficients are then used within the framework of the Briggs and Young method to estimate the global heat transfer performance. Prior to applying this approach, the relevant convective coefficients must be evaluated independently.[2]

1. [2] **Perry, R.H., & Green, D.W.**, *Perry's Chemical Engineers' Handbook*, 8th Edition, McGraw-Hill, 2008.

IIV.3.1. Fin Spacing Calculation

Based on the linear fin density provided in Table IV.2, the spacing between two consecutive fins, denoted as δ, can be calculated using the following expression:

$$\delta=\frac{1}{nfin}-ea \tag{IV.1}$$

Where $nfin$ is the number of fins per meter of tube length.

IV.3.2 Estimation of the Overall Heat Transfer Coefficient

As a first assumption, the fin efficiency is taken to be ideal (**η_0 = 1).** The overall heat transfer coefficient K_0 accounts for multiple thermal resistances, including:

- Convective resistance on the air side;
- Convective resistance on the MEA side;
- Conduction through the tube and fin materials;
- Fouling resistance inside the tubes and, to a lesser extent, on the air side.

The global heat transfer coefficient is computed using the following expression:

$$\frac{1}{K_0}=\frac{1}{h_a\frac{A_T}{A_0}}+r_m+\frac{1}{h_i\frac{A_i}{A_0}}+\left(\frac{e}{\lambda}\right)_t+R_{FOULING} \tag{IV.2}$$

Where:

- **h_a**: convective heat transfer coefficient of air (external side)
- **h_i**: convective heat transfer coefficient of MEA (internal side)
- **r_m**: effective thermal resistance due to fin geometry and material
- **A_T**: total external surface area including fins
- **A_0:** external surface area of the bare tube
- **$R_{fouling}$**: internal fouling resistance

The number of fins per unit length and the ratio of finned surface area to bare tube surface area are specified in Table IV.4

Table IV.4 : Fin Count and Fin-to-Bare Tube Surface Ratio

Fins per inch	7	8	9	10	11
Fins per meter	275	315	354	394	433
Fin surface / bare tube surface ratio	17.1	19.2	21.2	23.2	23.2

The fouling resistance on the MEA side is given by:

$$R_{\text{fouling total}} = R_{\text{fouling internal}} \times \frac{d_0}{d_i} \quad \text{(IV. 3)}$$

Using the data from Table IV.1 and the following table, the spacing between successive fins, the total fouling resistance, and the overall heat transfer coefficient are calculated.

Table IV.5: Air Cooler Performance Input Parameters

Parameter	**Value**
rm (m²·h·°C/kcal)	0.000151
hi (kcal/h·m²·°C)	3500
ha (kcal/h·m²·°C)	60
$Re_{\text{fouling internal}}$ (m²·h·°C/kcal)	0.0004
St/S0	23.2

$$\frac{1}{K0} = \frac{1}{60 * 23.2} + 0.000151 + \frac{1}{3500 * \frac{21.2}{25.4}} + 0.0004 + \frac{2.1 * 10 - 3}{43.172} = 0.001660351$$

$$K0 = \frac{1}{0.001660351} \approx 602 Kcal/hm^2 C^{\circ}$$

Table IV.6: Evaluation of δ, K₀, and Total Fouling Resistance

Results		
δ (mm)	**K0 (kcal/h.m2.°C)**	**R fouling_total (m2 h °C / Kcal)**
19.54	602	0.0004428

IV.3.3 Air Outlet Temperature

To determine the outlet temperature of the cooling air after heat exchange with the hot MEA fluid, a diagram from Appendix1 is consulted. This diagram provides the required number of tube rows (nr) to derive the following ratios: $\frac{\Delta T_{air}}{\Delta T_{moy}}$ and $\frac{T_1 - T_2}{T_1 - t_1}$

Where:
T_1: Inlet temperature of the hot fluid (MEA)
T_2: Inlet temperature of the cooling air
t_1: Outlet temperature of MEA
t_2: Outlet temperature of air
Based on the diagram, from Appendix1 a number of tube rows *n = 5* was selected, enabling the calculation of the following ratio:

$$\frac{\Delta T_{air}}{\Delta T_{moy}} = 1.11 \tag{IV.4}$$

Next, we calculate the following temperature ratio based on the inlet and outlet temperatures of the hot fluid (MEA):

$$\frac{T_1 - T_2}{T_1 - t_1} = \frac{73 - 46}{73 - 30} = 0.62 \tag{IV.5}$$

Using from Appendix 3, and based on the two ratios above, we can determine the temperature rise of the air as it passes through the heat exchanger:

$$\frac{\Delta T_{air}}{T_1 - t_1} = 0.48 \tag{IV.6}$$

Then, the air temperature rise is:

$$\Delta T_{air} = 0.48 \times 43 = 20.64\ °C$$

Now, the average air temperature difference (mean driving force) is:

$$\Delta T_m = \frac{\Delta T\ air}{1.11} = 20.64 / 1.11 = 18.59\ °C$$

To verify the above calculation, we estimate the air outlet temperature:

$$t_2 = T_2 + \Delta T_{air} = 30 + 20.64 = 50.64\ °C$$

The logarithmic mean temperature difference (LMTD) is then calculated as:

$$\Delta T_{lm} = \frac{(73 - 50.64) - (46 - 30)}{Ln\frac{(73-50.64)}{(46-30)}} = 19.002 C^{\circ} \tag{IV.7}$$

Table IV.7: System Temperatures

Fluid	MEA	Air
Inlet Temperature (°C)	73	30
Outlet Temperature (°C)	46	50.64
Temperature Difference (°C)	27	20.64

This table summarizes the inlet and outlet temperatures of the MEA solution and cooling air, along with their respective temperature differences, as used in the estimation of the mean temperature difference and air outlet temperature.

Estimated Mean Temperature Difference: $\Delta T_{mean} \approx 19$ °C

IV.3.4. Calculation of Total Bare Tube Surface and Air Mass Flow

First, we estimate the total bare tube surface area using the formula:

$$S_T = \frac{\Phi}{K_0 \Delta T_{lm} f_c} = \frac{1.30064 \times 10^6}{602 \times 19 \times 1} = 113.62 m^2 \qquad \text{(IV.8)}$$

The correction factor fc is taken from charts. For a number of rows greater than 3, fc =1.

Next, we estimate the face area:

$$S_F = 0.9 \times (S_0/N_R) \qquad \text{(IV.9)}$$

$$S_F = \mathbf{20.45\ m^2}$$

- **Geometric verification**:

Assuming a tube length L = 6.8 m

$$\text{Width } \mathbf{l = S_F / L} \qquad \text{(IV.10)}$$

l= 20.45 /6.8 = 3.007 m

Standard width = 10' = 3.05 m, thus SF = 6.8 × 3.05 m

Tubes arranged in a triangular pattern with pitch = 60.5 mm (2.375")

Number of tubes per row: (n + 0.5) × pitch = 3.05

$n = (3.05 / (60.5 \times 10^{-3})) - 0.5 = 49$ tubes

Total number of tubes: $n_T = 49 \times 5 = 245$ tubes

Installed heat exchange surface area:

$$\mathbf{S_{ins} = S0 \times n_T} \qquad \text{(IV.11)}$$

$$S_{ins} = \pi \times d_0 \times L \times n_T = 3.14 \times 0.0254 \times 6.8 \times 245 = 132.87 \text{ m}^2$$

- **Thermal Verification**
  - Air mass flow rate:

$$m_a = \frac{\Phi}{C_a(T_{FS} - T_{FE})} \quad \text{(IV.12)}$$

$$m_a = \frac{1.30064 \times 10^6}{0,24 \times (50.64 - 30)} = 262564.599 \, Kg/h$$

C_a given by **1005 J/kg·°C ≈ 0.24 kcal/kg·°C**.

  - Mass velocity:

$$G = \frac{m_a}{S_F} \quad \text{(IV.13)}$$

$$G = \frac{262564.599}{20.45 \times 3600} = 3.56 \text{Kg/m}^2\text{S}$$

Table IV.8: Summary Table Geometric and Thermal Results

Parameter	Value
Estimated bare tube surface area (ST)	113.62m²
Correction factor (fc)	1
Estimated face area (SF)	20.45 m²
Tube length (L)	6.8 m
Width (l)	3.02 m
Number of tubes per row (n)	49
Total number of tubes (nT)	245
Installed exchange surface (S_{ins})	132.87 m²
Air mass flow rate (ṁa)	262564.599kg/h
Mass velocity (G)	3.56 kg/m²·s

IV.3.5 Calculation of Convective Heat Transfer Coefficients

IV.3.5.1 Air-Side Convection Coefficient (ha): Briggs and Young Method

$$h_a = \frac{0.134\lambda_a}{d0}\left(\frac{d_0 G_{max}}{\mu_a}\right)^{0.681}\left(\frac{C_a \mu_a}{\lambda_a}\right)^{1/3}\left(\frac{\delta}{l}\right)^{0.2}\left(\frac{\delta}{t}\right)^{0.1134} \quad \text{(IV.14)}$$

Given: The following table summarizes the input data used in calculating the air-side convection coefficient.

Table IV.9: Finned Tube Geometry and Air Viscosity Parameters

Parameter	Value
Outer tube diameter (D_o)	0.0254 m
Fin height (l)	0.0159 m
Fin thickness (T)	0.000355 m
Fin spacing (δ)	0.01955 m (1/434 – 0.000355)
Finned tube diameter (d_a)	0.05722 m
Dynamic viscosity at 20 °C (μ_a)	1.82×10^{-5} Pl
Dynamic viscosity at 60 °C (μ_a)	2.00×10^{-5} Pl
Thermal conductivity at 20 °C (λ)	0.0224 kcal/h·m·°C
Thermal conductivity at 60 °C (λ)	0.0254 kcal/h·m·°C

$$h_a = \frac{0.134 \times 0.0239}{0.0254}\left(\frac{0.0254 \times 3.56}{1.91 \times 10^{-5}}\right)^{0.681}\left(\frac{1005 \times 1.91.10^{-5}}{0.0239}\right)^{\frac{1}{3}}\left(\frac{0.01955}{0.0159}\right)^{0.2}\left(\frac{0.01955}{0.000355}\right)^{0.1134}$$

$$h_a = 58.84 Kcal/hm^2C^{\circ}$$

IV.3.5.2. MEA-Side Convection Coefficient (hi)

Fluid properties at average temperature:

Thermal conductivity: λ = 0.5844 W/m·K

Dynamic viscosity: μ = 0.5739 cP

Specific heat capacity: C = 4122 kJ/kg·°C

- **Step 1**: Calculate flow cross-sectional area

$$S_P = \frac{n_T \times \pi . d_i^{\ 2}}{n_{ptubes} \times 4} = \frac{245 \times 3.14 \times 21.2^2 . 10^{-6}}{5 \times 4} = 0.01728 m^2 \quad \text{(IV.15)}$$

- **Step 2**: Calculate flow velocity of MEA

$$V_i = \frac{m.C}{\rho.S_P} = 0.785\,\frac{m}{s} \quad \text{(IV.16)}$$

- **Step 3**: Calculate Reynolds number

$$R_e = \frac{\rho.v_i.d_i}{\mu} = \mathbf{36254.07} \quad \text{(IV.17)}$$

- **Step 4**: Calculate Prandtl number

$$p_r = \frac{\mu.C}{\lambda} = 2.725 \quad \text{(IV.18)}$$

- **Step 5**: Calculate Nusselt number

$$N_u = 0.023 \times {R_e}^{0.8} \times P_r^{\frac{1}{3}} = 142.67 \quad \text{(IV.19)}$$

- **Step 6**: Calculate internal convection coefficient, hi

$$h_i = \frac{142.19 \times 0.5950}{21.2 \times 10^{-3}} = 3990.84 Kcal/hm^2C^\circ \quad \text{(IV.20)}$$

IV.3.5.3. Global Heat Transfer and Pressure Drop Calculations

$$K_0 = \frac{1}{\frac{1}{58.84*23.2} + 0.000151 + \frac{1}{3990.84*\frac{21.2}{25.4}} + 0.0004 + \frac{2.1*10-3}{43.172}} = 612.59 Kcal/hm^2C^\circ \quad \text{(IV.21)}$$

Relative Difference:

$$\varepsilon = \frac{612 - 602}{612} = 0.01 = 1.10^{-2}$$

The error is sufficiently small and can be considered acceptable. However, an additional iteration may be carried out for improved precision, if required.

IV.3.5.4. Pressure Drop and Fan Power Calculations

1. **Pressure Drop Inside Tubes**

We use the Darcy–Weisbach equation to calculate the pressure drop inside the tubes:

$$\Delta P_i = n_p \times \rho \times v^2 (f \times \frac{L}{d_i} + 2) \quad \text{(IV.22)}$$

2. **Pressure Drop Outside Tubes**

The pressure drop across the tube bundle is given by:

$$\Delta P_a = f_a \times \frac{G^2.N_R}{\rho_{ma}} \quad \text{(IV.23)}$$

Briggs-Robinson Equation

$$f_a = 18.893\left(\frac{d_r.G}{\mu_a}\right)^{-0.316}\left(\frac{\delta}{d_r}\right)^{-0.927} \quad \text{(IV.25)}$$

$f_a = 1.681$

3. Fan Power Calculation

Number of fans:

$n_V = \frac{S_F}{S_V} = 2\ to\ 2.5$ $\quad S_V = \frac{20.45}{2.5} = 8.18m^2$ (→ 2 fans)

Fan surface area: $S_V = \frac{8.18}{2} = 4.09m^2$

Fan diameter: $D_V = \sqrt{\frac{4\times 4.09}{3.14}} = 2.28m$

This corresponds to 7.51 feet, that is, a fan with a diameter of 8 feet (2.43 m).
And a cross-sectional area of: Sv = 4.63 m²

IV.3.5.5. The variation of dynamic pressure in the fan ring is given by:

$$\Delta P_d = \rho \frac{v^2}{2} \quad \text{(IV.26)}$$

- **Forced draft:** $\rho = \rho_{20}\frac{293}{273+t_1}\times F_a$ (IV.27)
- **Induced draft:** $\rho = \rho_{20}\frac{293}{273+t_2}\times F_a$ (IV.28)

But we opted for **forced draft**: $\rho = 1.2\frac{293}{303}\times 1 = 1.160Kg/m^3$

- **Velocity at the fan ring:** $V_{ring} = \frac{\text{Air volumetric flow rate}}{surface\ fan}$ (IV.29)
- **Air volumetric flow rate per fan**
  $Q_V = \frac{flow\ rate\ of\ air}{n_V} = 131108.51m^3/h$ (IV.30)

Fan air velocity: $V_{\text{velocity}} = \frac{Q_V}{S_V} = \frac{36.41}{4.63} = 7.86m/s$

Dynamic pressure loss: $\Delta P_d = \frac{\rho_V . v^2}{2} = 1.160.\frac{7.86^2}{2} = 35.83 Pa$ (IV.31)

Total pressure loss: $\Delta P_T = \Delta P_a + \Delta P_d$ (IV.32)

Power per fan: $P_V = \frac{Q_V . \Delta P_T}{\eta_m . \eta_v}$ (IV.33)

Installed power with 10% safety factor: $P_{inst} = P_V .1.1.\frac{\rho_{Thiver}}{\rho_{Tété}}$ (IV.34)

$P_{inst} = 9.91.1.1.\frac{1.288}{1.088} = 12.90 \approx 13CV$ per fan

Total installed power: 13 × 2 = 26 CV

Table IV.10: Summary of Key Results

Parameter	Result
Pressure drop inside tubes (ΔP_i)	47734 Pa[3]
Pressure drop outside tubes (ΔP_a)	87.72 Pa
Friction coefficient (f_a)	1.681
Fan air velocity	7.86 m/s
Dynamic pressure loss (ΔPd)	35.83 Pa
Total pressure loss (ΔPT)	123.55 Pa
Fan power per unit	9.91 CV
Installed fan power (per unit)	13 CV
Total installed power	26 CV

IV.3.5.6. Fan Diameter Verification

$$\frac{S_F}{S_V} = \frac{20.45}{4.09 \times 2} = 2.20 < 2.5 \quad \text{(IV.35)}$$

The selected fan diameter is confirmed to be appropriate based on the calculations.

IV.4 Comparison with Simulation Results

In order to validate the results obtained through analytical calculations, a simulation was carried out using the HTRI Xchanger Suite, a specialized software widely used for the thermal design

3 **Coulson, J.M., Richardson, J.F.**, *Chemical Engineering Volume 6: Design*, 6th Edition, Butterworth-Heinemann, 2019.

and performance evaluation of heat exchangers.
This step allowed for a comparison between the manually calculated values and those generated by the software, particularly in terms of heat transfer area, outlet temperatures, and overall thermal duty.
[4]The comparison aims to assess the accuracy of the design approach and identify any significant deviations or confirmations between the two methods.

IV.4.1 HTRI Xchanger Suite – Simulation Steps

- Opened the Xace module in HTRI.
- Selected the fluid type and defined operating conditions.
- Entered process and air side parameters.
- Chose tube type, fin configuration, and layout.
- Defined fan arrangement and enabled louvers.
- Ran the simulation and reviewed key results.
- Used results to compare with manual calculations.

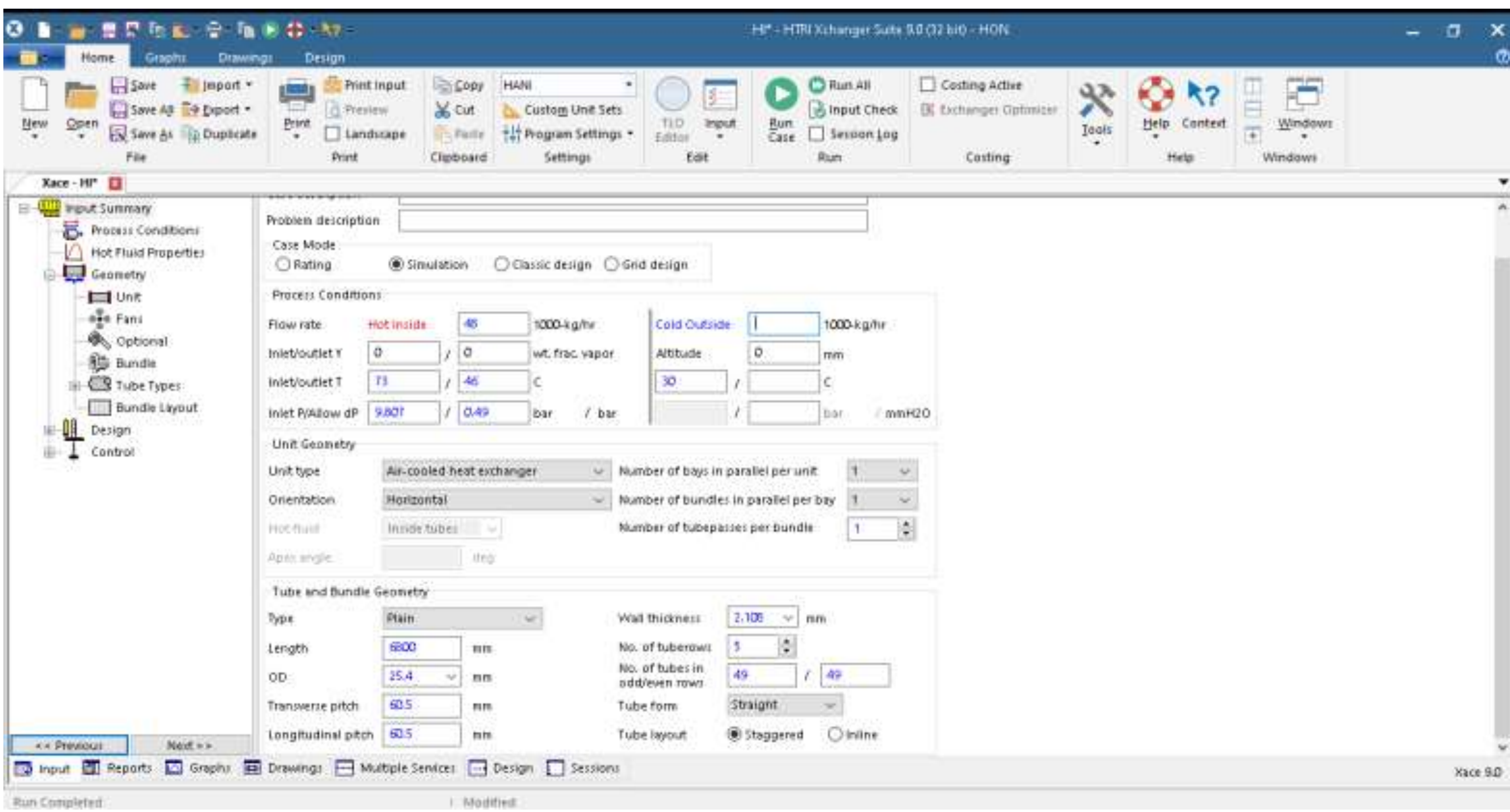

Fig IV.2: HTRI Xchanger Suite – Input Data Interface

As shown in Figure IV.2, the HTRI Xchanger Suite required the following input parameters to simulate the air-cooled heat exchanger:

Process Conditions

- Flow rate of hot fluid (kg/h)
- Inlet and outlet temperatures of hot fluid (°C)
- Inlet pressure / allowable pressure drop (bar)
- Cold side ambient temperature (°C)
- Altitude (m)

Unit Geometry

- Unit type: Air-cooled heat exchanger
- Orientation: Horizontal
- Hot fluid location: Inside tubes

Tube and Bundle Geometry

- Tube length (mm)
- Tube outer diameter (OD) (mm)
- Wall thickness (mm)
- Number of tube rows (odd/even)
- Total number of tubes per row

IV.4.2 HTRI Results Summary

After entering all the required input parameters into the HTRI Xchanger Suite, the simulation was successfully executed. The results obtained were consistent and aligned well with the manual calculations, as shown in **Table IV.11**, confirming the reliability of the design.

Table IV.11: Comparison of key thermal design outputs between manual calculation and HTRI simulation results.

Parameter	Manual Calculation	HTRI Simulation Output
Heat Duty (MMkcal/h)	1.3	1.300
Outlet Air Temperature (°C)	50.64	50.59
Total Air Flow Rate (m^3/min)	262.564	262.564
Required Overall Heat Transfer Coefficient U (kcal/m^2·h·°C)	612.84	610.36
Heat Transfer Area (m^2)	132	130.46
Tube Count	245	245

IV.4.3 Air Cooler Design Overview

The air-cooled heat exchanger selected for this application was configured using the HTRI Xchanger Suite software. Based on the thermal and process input conditions, the software generated a detailed schematic of the exchanger, which includes essential mechanical and geometrical features such as the number of tube rows, bundle dimensions, fan arrangement, and tube layout.

As shown in **Figure VI.3**, **Figure VI.4**, the air cooler is of horizontal configuration with forced-draft fans located below the tube bundle. The hot process fluid (MEA solution) flows inside plain, finned tubes arranged in a staggered layout, while ambient air is drawn across the external surface of the tubes to remove heat by forced convection. The exchanger includes a single bundle with one pass, five tube rows, and a total of 245 tubes. The tube outer diameter and pitch were selected to ensure optimal heat transfer while minimizing pressure losses.

This visual representation not only confirms the feasibility of the thermal design but also assists in evaluating the physical integration of the unit within the plant. The geometry, fan position, and layout contribute significantly to the air distribution and overall cooling performance of the exchanger.

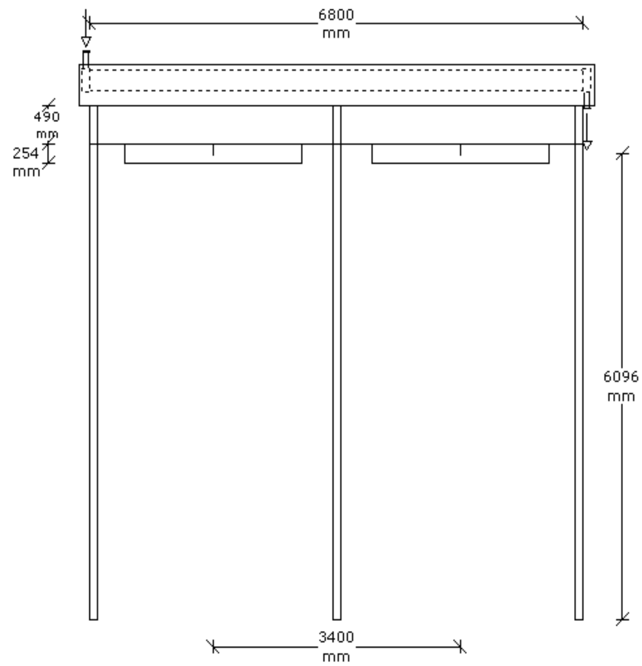

Fig VI.3: 2d exchnger side view

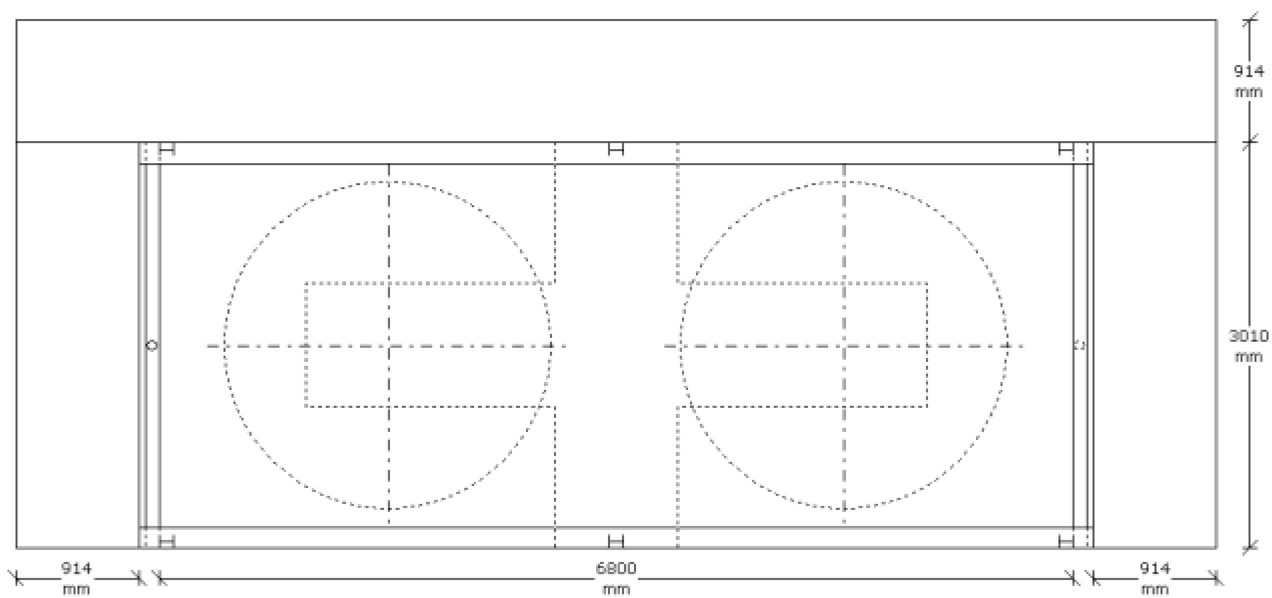

Fig VI.4: 2D exchnger top view

Conclusion and Recommendations

Conclusion

This thesis has examined the operational challenges faced by the E.504 shell-and-tube heat exchanger in the decarbonation section of the GL1/Z natural gas liquefaction complex, particularly the persistent fouling and corrosion issues resulting from prolonged exposure to seawater and MEA solution. The reduction in heat transfer efficiency, increased maintenance requirements, and process inefficiencies directly trace back to the thermal and mechanical degradation of this exchanger over time.

To address this critical problem, the study explored the potential of replacing the existing exchanger with an Air-Cooled Heat Exchanger (ACHE). A detailed analysis of air cooler technology including its design principles, operating mechanisms, components, and benefits was presented. Furthermore, a full thermal sizing calculation was performed to determine the technical feasibility of integrating an ACHE into the GL1/Z process.

The results indicate that air-cooled heat exchangers present a viable, efficient, and environmentally favorable alternative. By leveraging ambient air as the cooling medium, ACHEs mitigate the risks of corrosion, reduce fouling tendencies, and enhance long-term reliability, especially in coastal or seawater-exposed installations.

Recommendations

Based on the findings of this study, the following recommendations are proposed:

1. Immediate Replacement Strategy:

   Begin the engineering and procurement process for the replacement of E.504 with a properly sized ACHE. The design provided in Chapter IV should serve as the baseline configuration.

2. Material Selection Optimization:

   Use corrosion-resistant materials such as stainless steel for the tube side and aluminum fins to ensure long-term durability in outdoor, humid environments.

3. Implement Preventive Maintenance Protocols:

   Establish a scheduled inspection and cleaning regime for the ACHE fans, fin surfaces, and headers to maintain high thermal efficiency and prevent dust or debris accumulation.

4. Adopt Modular Configuration:

   Consider using a modular air cooler design with multiple fans and tube bundles, which provides operational flexibility and allows partial shutdowns for maintenance without interrupting the entire process.

5. Fan Speed Control:

   Integrate variable frequency drives (VFDs) for the fan motors to optimize energy consumption according to cooling demand, especially under varying ambient conditions.

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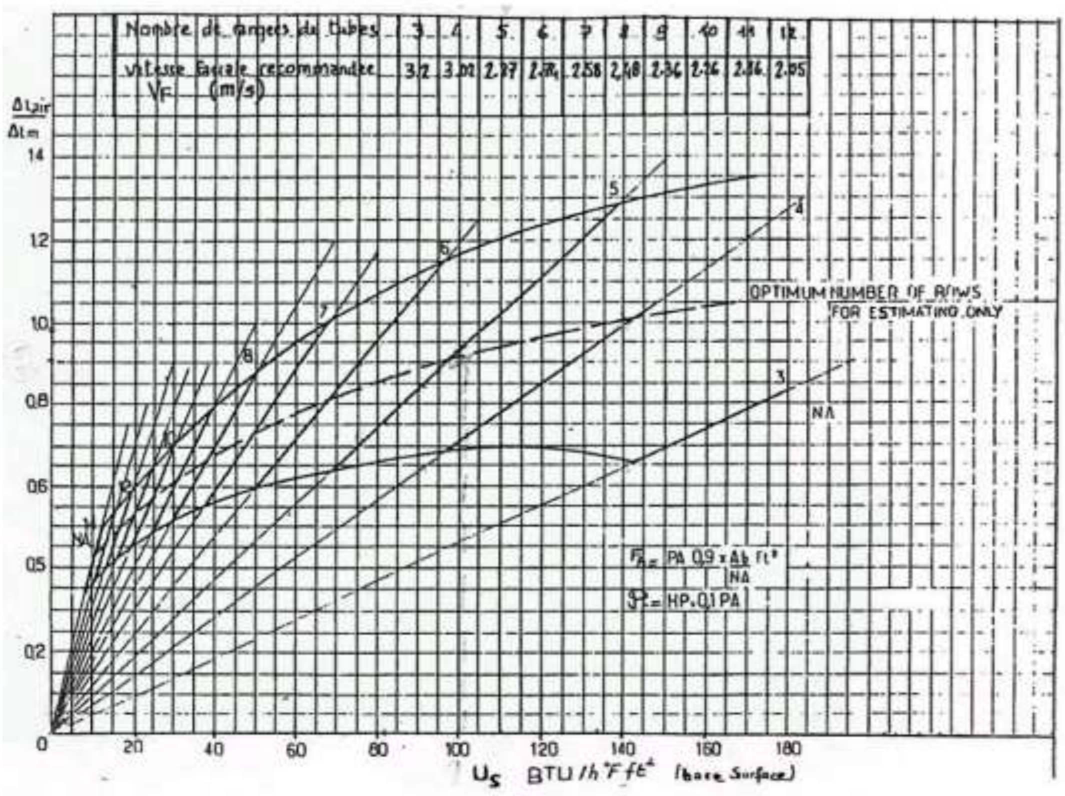
Appendix



Estimation of the Global Heat Transfer Coefficient

HEAT EXCHANGER SPECIFICATION SHEET
(S.I. UNITS)

REQ'N No 20784-6-014
ITEM No X01-E-504
PLANT

Customer SONATRACH
Project location GL1-Z PROJECT RENOVATION — Manufacturer STRUTHERS INDUSTRIES
Item No
Service of Unit REPLACEMENT TUBE BUNDLES FOR M.E.A. SOLUTION COOLERS
Size: x — Type: — Posn: — Connected in: Series Parallel
Surf./Unit (Eff) m^2 — No. of Shells/Unit — Surf/Shell (Eff) m^2

PERFORMANCE OF ONE UNIT								
	Shellside				Tubeside (No OF UNITS REQ'D)			
Fluid Allocation								
Fluid Circulated	15% MEA LEAN SOLUTION				SEAWATER (CHLORINATED)			
Total Fluid Entering Kg/S								
	IN		OUT		IN		OUT	
Liquid Kg/S								
Vapour Kg/S/MW								
Noncond Kg/S/MW								
Steam Kg/S								
Water Kg/S								
Fluid Vap./Cond. Kg/S								
	LIQ.	VAP.	LIQ.	VAP.	LIQ.	VAP.	LIQ.	VAP.
Density Kg/m^3								
Viscosity cP								
Therm. Cond. W/m °C								
Specific Heat KJ/Kg °C								
Temperature °C								
Operating Press. bar A								
No. passes / Shell								
Velocity m/sec								
Press. Drop, Allow / Calc bar		/				/		
Fouling Resistance m^2°C/W								
Heat Exchanged MW					MTD °C			
Transfer Rate W/m^2°C	Service				Clean			

CONSTRUCTION				
	Shellside		Tubeside	
Design / Test Press. bar G	63.8	TO CODE	5.17 & F.VAC	TO CODE
Design Temperature (Max / Min) °C	93 / –		/	
Corrosion Allowance mm				
Connections Inlet ins	No.	N.B.	No.	N.B.
Connections Outlet ins	No.	N.B.	No.	N.B.
Rating (ANSI) / Facing Ref.	LB		LB	

Tubes No: 410 OD: 19.05 mm THK(MIN) (1) mm Length: 6.096 M Pitch: 25.4 mm Flow →
Tube Material: CS / 70-30 CuNi BIMETALLIC (2) — Tube-Tubesheet Joint: EXPANDED
Shell: ID: mm — Shell Cover:
Channel / Bonnet: — Channel Cover:
Tubesheet Stationary: FORGED CS. + 10mm AL BRZE CLAD. — Tubesheet Floating: FORGED CS + 10mm AL BRZE CLAD.
Floating Head Cover: — Impingement Plate:
Baffles - Cross: C.S. — Type: HORIZ SEG — % Cut (DIA) 20% — Spacing 152 mm
Baffles - Long: — Seal: — Pairs: — Tube supports:
Insul. Thk.: Shell: mm — Chan: mm — Expansion Joint:
Gaskets: Shellside: STEEL DOUBLE JACKETED - NON ASBESTOS FILLED — Tubeside: COPPER DBL JKTD - NON ASBESTOS FILLED
Code Requirements: ASME 8, DIV 1 — Stamp: NO — TEMA Class: 'R' — Specs: SEE M/R
Weight: Each Shell: Kg — Bundle Kg — Full Of Water Kg

Remarks: (1) BIMETALLIC TUBES, OUTER TUBE CS 1.65mm THK, INNER TUBE 70/30 CuNi x 1.65 mm THICK.
(2) OUTER TUBE TO BE FITTED WITH SUITABLE 70/30 CuNi FERRULE FOR FITTING TO TUBESHEET.
(3) BUNDLES SHALL BE CONSTRUCTED IN ACCORDANCE WITH THE DRAWINGS REFERENCED IN MATL REQUISITION, EXCEPT TUBE MATL SHALL BE AS ABOVE.

No	DATE	REVISION	BY	CHK	APP
3	23/6/93	ISSUED FINAL	SK	[illegible]	[illegible]
2	2/2/93	ISSUED FOR PURCHASE	S.K	[illegible]	[illegible]
1	7/8/92	ISSUED FOR PURCHASE	[illegible]	[illegible]	[illegible]
0	13/DEC/91	ISSUED FOR QUOTATION	[illegible]	[illegible]	[illegible]

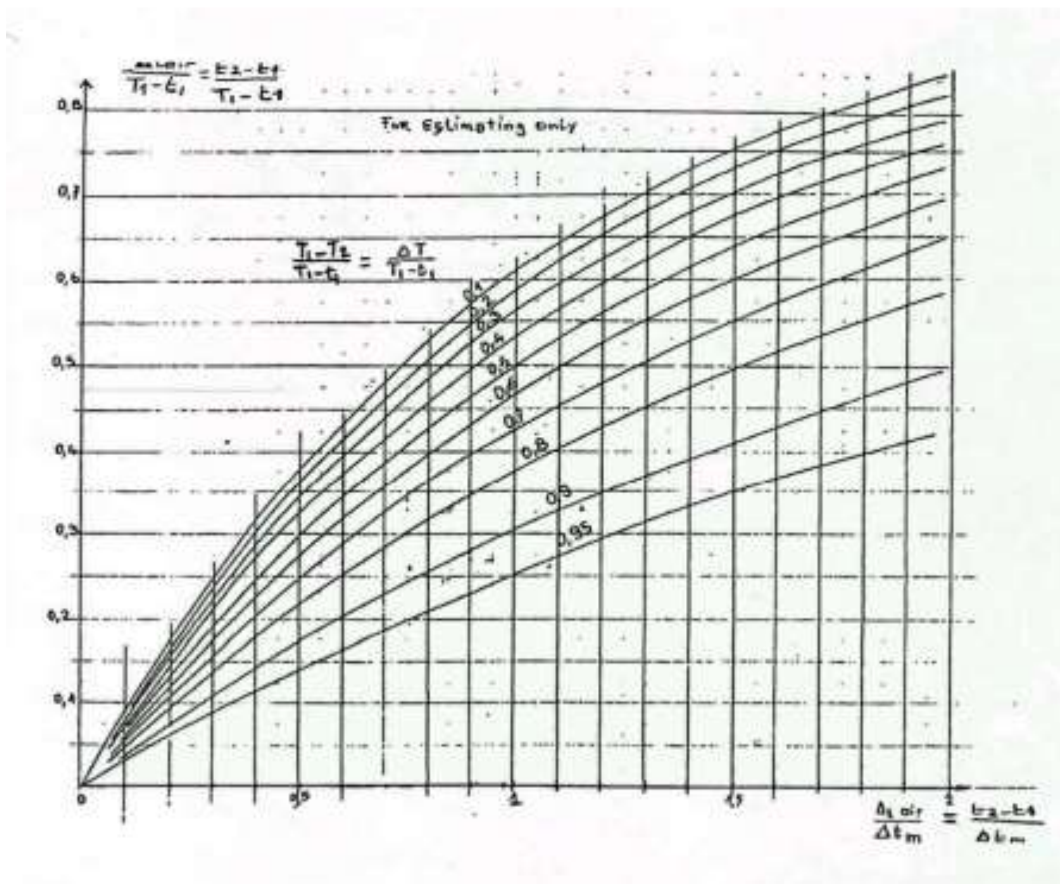
JOB NUMBER: 20784
DRAWING No DS-X01-E-104 REV 3
SHEET 1 OF 1

DATASHEET PREVIOUSLY ISSUED AS X01-DS-E-001

(LON) E-138 (S.I) FEB'70

Sonatrach Approval — Revision

Specification Sheet



Estimation of the Air Outlet Temperature

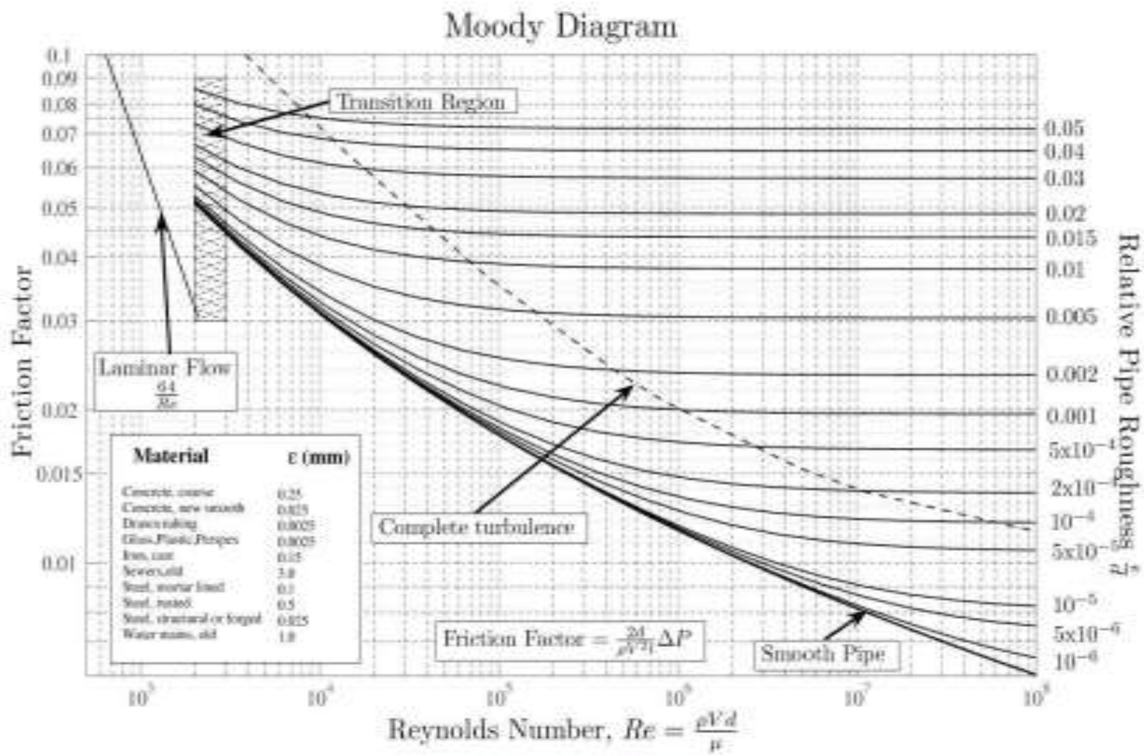


Diagramme de Moody

Moody Diagram

DIAMÈTRE EXTÉRIEUR (in et mm)	BWG	ÉPAISSEUR (mm)	DIAMÈTRE INTÉRIEUR (cm)	SECTION (cm^2)	SURFACE (m^2/m)		POIDS (kg/m)
					extérieure	intérieure	
1/2 in (12,7 mm)	14	2,10	0,848	0,565	0,0399	0,0266	0,600
	16	1,65	0,940	0,694		0,0295	0,490
	18	1,24	1,021	0,819		0,0321	0,384
3/4 in (19,05 mm)	10	3,40	1,224	1,177	0,0598	0,0384	1,436
	12	2,77	1,351	1,434		0,0424	1,216
	14	2,10	1,483	1,727		0,0466	0,963
	16	1,65	1,575	1,948		0,0495	0,774
	18	1,24	1,656	2,154		0,0520	0,597
1 in (25,4 mm)	10	3,40	1,859	2,714	0,0798	0,0584	2,024
	12	2,77	1,986	3,098		0,0624	1,696
	14	2,10	2,118	3,523		0,0665	1,324
	16	1,65	2,210	3,836		0,0694	1,057
	18	1,24	2,291	4,122		0,0720	0,811
1 1/4 in (31,75 mm)	10	3,40	2,494	4,885	0,0997	0,0783	2,604
	12	2,77	2,616	5,375		0,0822	2,158
	14	2,10	2,743	5,909		0,0862	1,682
	16	1,65	2,845	6,357		0,0894	1,340
	18	1,24	2,291	6,701		0,0918	1,024
1 1/2 in (38,1 mm)	10	3,40	3,124	7,665	0,1197	0,0981	3,185
	12	2,77	3,251	8,300		0,1021	2,634
	14	2,10	3,378	8,962		0,1061	2,039
	16	1,65	3,480	9,512		0,1093	1,622
	18	1,24	3,556	9,931		0,1171	1,237

BWG (Birmingham Wire Gauge) : Norme caractérisant l'épaisseur des tubes selon le diamètre et correspondant à une pression maximale d'utilisation.

Characteristics of the Heat Exchanger Tubes

Propriétés physiques de l'air sec ($B = 760$ mm Hg $= 1{,}01 \cdot 10^5$ Pa) [13]

t, °C	ρ, kg/m³	c_p, kJ/(kg·°C)	$\lambda \cdot 10^2$, W/(m·°C)	$a \cdot 10^6$, m²/s	$\mu \cdot 10^6$, Pa·s	$\nu \cdot 10^6$, m²/s	Pr
−50	1,584	1,013	2,04	12,7	14,6	9,23	0,728
−40	1,515	1,013	2,12	13,8	15,2	10,04	0,728
−30	1,453	1,013	2,20	14,9	15,7	10,80	0,723
−20	1,395	1,009	2,28	16,2	16,2	12,79	0,716
−10	1,342	1,009	2,36	17,4	16,7	12,43	0,712
0	1,293	1,005	2,44	18,8	17,2	13,28	0,707
10	1,247	1,005	2,51	20,0	17,6	14,16	0,705
20	1,205	1,005	2,59	21,4	18,1	15,06	0,703
30	1,165	1,005	2,67	22,9	18,6	16,00	0,701
40	1,128	[illegible]	2,76	24,3	19,1	16,96	0,699
50	1,093	1,005	2,83	25,7	19,6	17,95	0,698
60	1,060	1,005	2,90	26,2	20,1	18,97	0,696
70	1,029	1,009	2,96	[illegible]	20,6	20,02	0,694
80	1,000	1,009	3,05	[illegible]	21,1	21,09	0,692
90	0,972	1,009	3,13	31,9	21,5	22,10	0,690
100	0,946	1,009	3,21	33,6	21,9	23,13	0,688
120	0,898	1,009	3,34	36,8	22,8	25,45	0,686
140	0,854	1,013	3,49	40,3	23,7	27,80	0,684
160	0,815	1,017	3,64	43,9	24,5	30,09	0,682
180	0,779	1,022	3,78	47,5	25,3	32,49	0,681
200	0,746	1,026	3,93	51,4	26,0	34,85	0,680
250	0,674	1,038	4,27	61,0	27,4	40,61	0,677
300	0,615	1,047	4,60	71,6	29,7	48,33	0,674
350	0,566	1,059	4,91	81,9	31,4	55,46	0,676
400	0,524	1,068	5,21	93,1	33,0	63,09	0,678

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Physical Properties of Air

Appendix A: Estimation of the Global Heat Transfer Coefficient

Appendix B: Specification Sheet

Appendix C: Estimation of the Air Outlet Temperature

Appendix D: Moody Diagram

Appendix E: Physical Properties of Air