

CHAPTER 1

INTRODUCTION

1.1 RESEARCH BACKGROUND

This research has been inspired by a problem encountered at the Nigeria Liquefied Natural Gas (NLNG) plant in Bonny Island, Nigeria, in 2004 which led to the loss of about 1000 metric tons of liquefied natural gas (LNG) production daily and an associated revenue loss of about 150,000 USD daily! This problem was caused by the cooling water return temperature which gradually rose above the plant design threshold of 32°C¹ and started to constrain the propane condensers (partial condensation of propane vapour from the refrigerant compressor) and hence, the propane refrigerant compressor power (due to high discharge pressure and peak firing limit of turbine driver). The root cause of the problem was traced to a leaking propane sub-cooler² exchanger tube. This caused propane to leak into the cooling water returning to the cooling tower from the plant and hence constrained the cooling efficiency of the cooling tower by damaging the packing, leading to lack of adequate air-water intermingling in the tower and resulting in warmer cooling water supply from the cooling tower to the propane condensers and other heat exchangers in the plant (Figure 1.1 and Figure 1.2). This problem has inspired this current research on cooling tower optimisation in order to find a way to improve the thermal efficiency of this important equipment.

A cooling tower and one or more heat exchangers comprise an open recirculating water-cooling system. A cooling tower cools warm water by exposing it to the air. This temperature reduction is the result of two processes: sensible heat transfer and evaporation. Sensible heat transfer accounts for 10 to 20% of the cooling effect in a cooling tower, while evaporative cooling accounts for 80 to 90% of the cooling effect.

This evaporative cooling water system is the key to the efficient management of

¹ Under normal operating conditions, the cooling water temperature averages 31°C. The 32°C design threshold is only exceeded during plant upsets, e.g., loss of one or more cooling tower fans.

² The sub-cooler reduces the temperature of condensed propane below the bubble point (saturated liquid state).

energy and water. For example, an increase in cooling water temperature to a plant due to under-performance of the cooling tower may result in the inability of the cold water to exchange heat or to condense vapour at the optimum temperature level, resulting in the loss of production and an increase in operating costs. Consequently, the design and the manner in which cooling towers are operated can have a very significant financial bearing on a plant. Hence the need to optimize the design and performance of cooling towers to improve their efficiency and invariably, the efficiency of the plant they service. The main objective of this research is to achieve this by modifying the cooling tower design. This modification involves the development of a system to pre-condition the inlet air to cooling towers.



Figure 1.1: Picture of Damaged Packing of NLNG Cooling Tower



Figure 1.2: Picture of Propane Condenser and Sub-cooler at NLNG Plant

1.2 PROBLEM STATEMENT

To improve the thermal performance of cooling towers by the conditioning of air in order to overcome the performance limitation due to the wet-bulb temperature of the ambient air. Currently, the wet-bulb temperature of the ambient air is theoretically the coldest temperature the cooling tower can reduce the water temperature.

1.3 OBJECTIVE OF RESEARCH

The principal objective of this research is to successfully develop a new concept to pre-condition the supply air to cooling towers. The aim is to improve the thermal performance of cooling towers by ‘conditioning’ the supply air wet-bulb temperature via control of the air moisture content.

The Nigerian Liquefied Natural Gas (NLNG) plant cooling towers were used as a case study.

1.4 SCOPE OF RESEARCH

The scope of the research is limited to the thermal analysis of the cooling towers. The mechanical induced draft-type cooling tower was used for this analysis. Available data (specifications) on the fluid properties and plant/equipment layout installation at the Nigeria Liquefied Natural Gas Plant were used as the basis for this study. Furthermore, this research focused mainly on the technical feasibility of the design; an overview of the economic feasibility was, however, carried out as part of this study.

1.5 COOLING TOWER IN A LIQUEFIED NATURAL GAS PLANT

Gas liquefaction is performed with the aid of three cooling circuits: the cooling-water circuit, the propane refrigerant circuit and the mixed refrigerant circuit. The mixed refrigerant (which is a mixture of Nitrogen, Methane, Ethane and Propane gases) liquefies the natural gas by refrigerating it down to a temperature of -160°C . The propane refrigerant circuit pre-cools the natural gas and partially condenses the mixed refrigerant. The cooling water circuit condenses and sub-cools the propane. The entire liquefaction circuit is a complete refrigeration cycle consisting of condensers, compressors, expanders and Joule-Thompson valves, in addition to a cryogenic heat exchanger, which acts as the evaporator where the actual liquefaction of the natural gas occurs. A process flow diagram of the plant is given in Figure 1.3. The cooling tower provides cooling water to both process and equipment heat exchangers using water for cooling or condensing within the Liquefied Natural Gas (LNG) Plant. Figure 1.4 is a line diagram of the cooling tower at NLNG plant. Each LNG Plant Production Train is equipped with an independent open re-circulation water-cooling system.

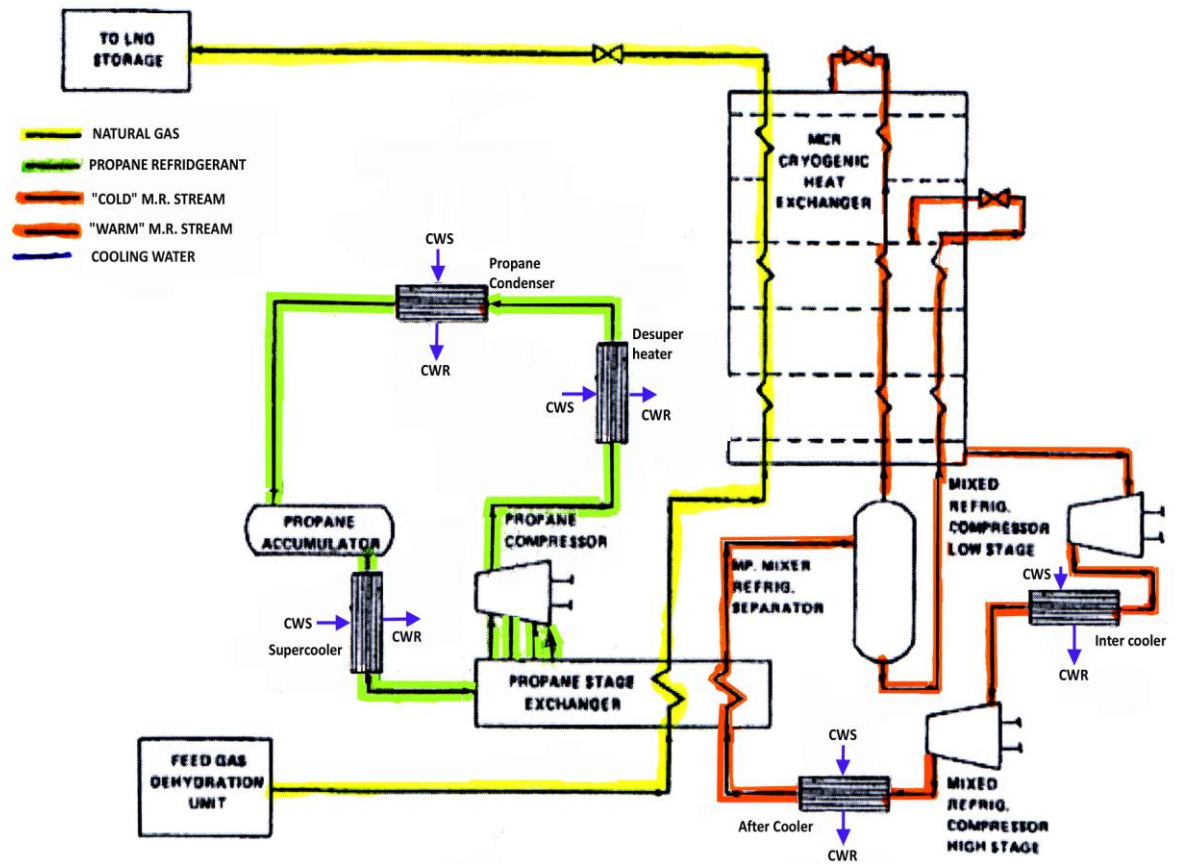


Figure 1.3: Process Flow Diagram of an LNG Plant
(Courtesy Thimm Engineering Inc.)

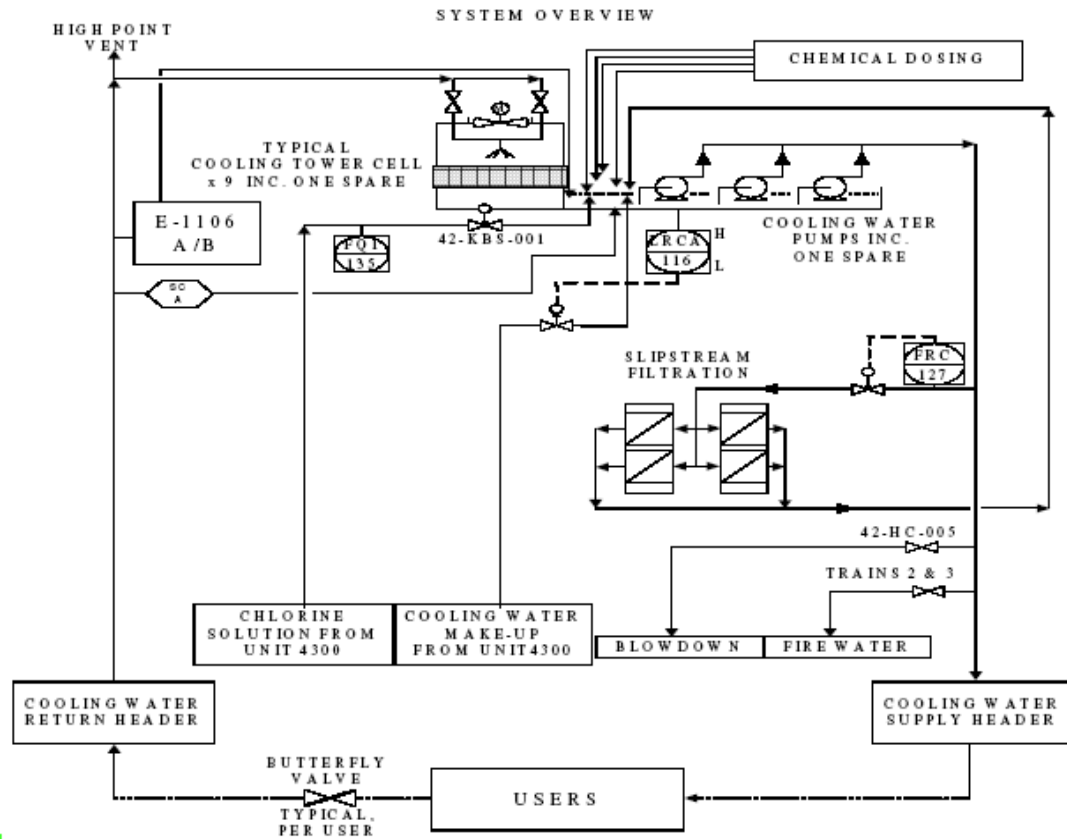


Figure 1.4: Line Diagram of NLNG Cooling Tower
(Courtesy of NLNG Operating Manual)

Optimisation is often a serious area of concern in the design of such process plants. During the optimisation stage, the primary question that needs to be answered is how to achieve maximum efficiency and use less energy. Since gas liquefaction is an energy intensive process, any liquefaction design will aim at minimising the energy consumption. A factor influencing the energy (power) consumption is the temperature at which the heat from the refrigerant cycle is rejected. Heat removal at low temperatures requires more energy than at high temperatures. Therefore, the heat transfer from a fluid to be cooled (the natural gas) to the refrigerant incurs an efficiency loss (coefficient of performance decrease), which increases with the temperature difference. The cooling capacity per unit of refrigerant is given by the enthalpy difference between refrigerant entering the compressor and that leaving the

water-cooled condenser (Silbert, 1999). This is shown in Figures 1.5 and 1.6 (i.e., $H_3 - H_1$). Since any heat produced by the process cycles or withdrawn from the natural gas finally ends up in the cooling water, the plant performance in terms of energy consumption and production thus depends on the efficiency of the cooling water circuit. The cooling water circuit is critical to the efficiency of the whole process. Hence the thermal optimisation of this circuit will invariably lead to improved performance of the process plant. For instance, the lower the cooling water temperature for condensing and cooling the propane refrigerant, the lower the compressor discharge pressure required and hence the compressor power (Figure 1.7). The condensation temperature of the propane is limited by the fact that condensation of a pure component takes place at a constant temperature. For example, with cooling water of 27°C and assuming a cooling water temperature rise of 10K and an approach³ of 5K (for the condenser), the condenser outlet temperature (of propane) is limited to 42°C (lowest possible temperature of propane from condenser outlet). Once it is liquid, it can be cooled another 10K to 32°C with water in a counter current cooler exchanger. With the propane now colder it will produce less vapour upon the first flash in the evaporator, which saves power. The saving is 11% for a single-stage propane compressor and 9% for the high-pressure stage of a four-stage propane compressor (Birchall, 1989). The power savings on the propane refrigerant circuit will eventually lead to power saving in the mixed refrigerant circuit and hence a total power savings in terms of cooling duty for the plant. This thus outlines the importance of thermal optimisation of the cooling water circuit which is the intent of this research.

A more detailed look at the cooling tower of a typical Liquefied Natural Gas Plant is presented in Chapter 2 (Literature Review).

³ The “approach” is the difference between the temperature of the condensed propane exiting the condenser and the temperature of the cooling water (from the cooling tower) entering the condenser.

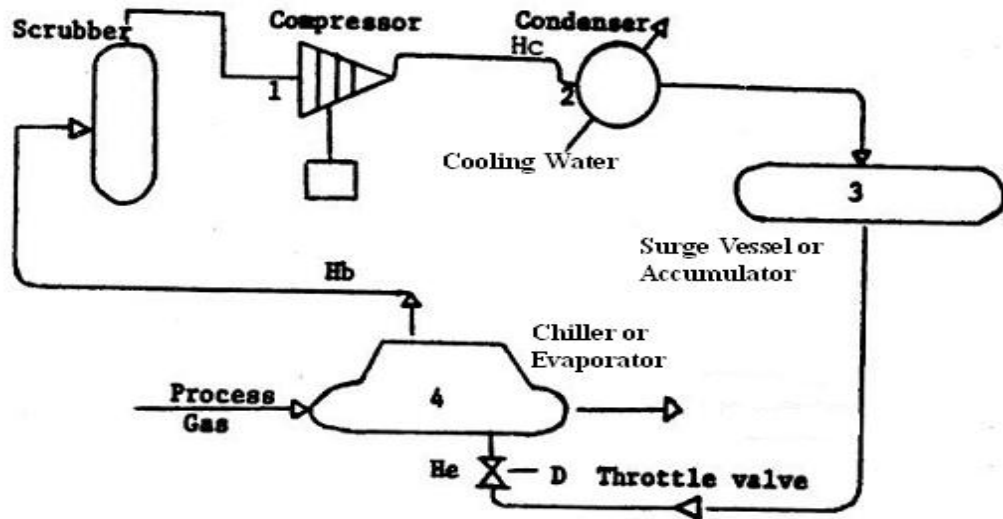


Figure 1.5: Propane Refrigeration Circuit for Natural Gas
(Courtesy Thimm Engineering Inc.)

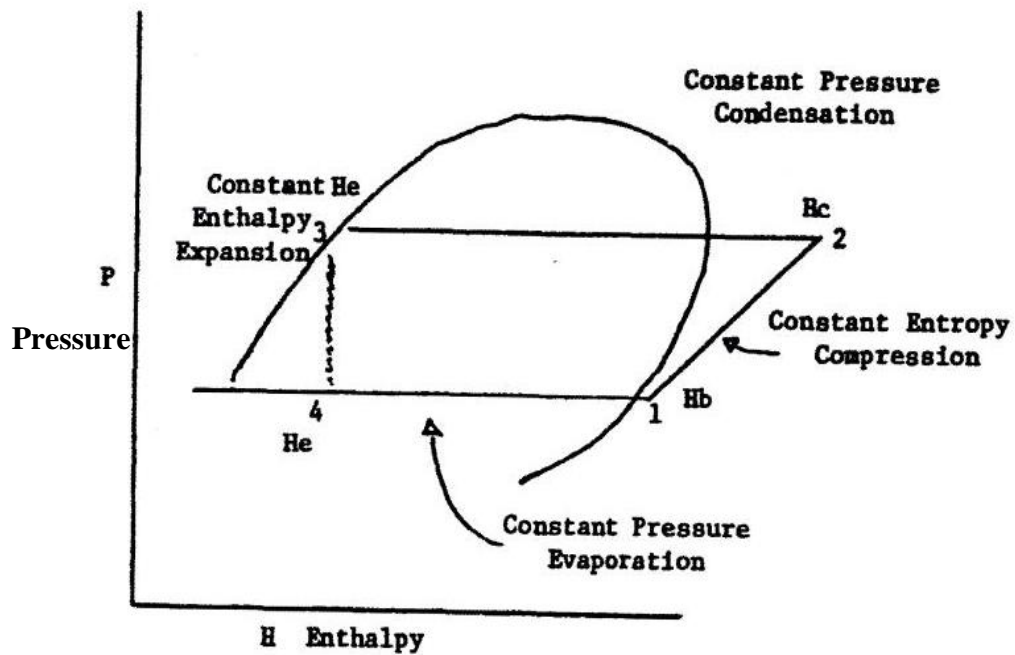
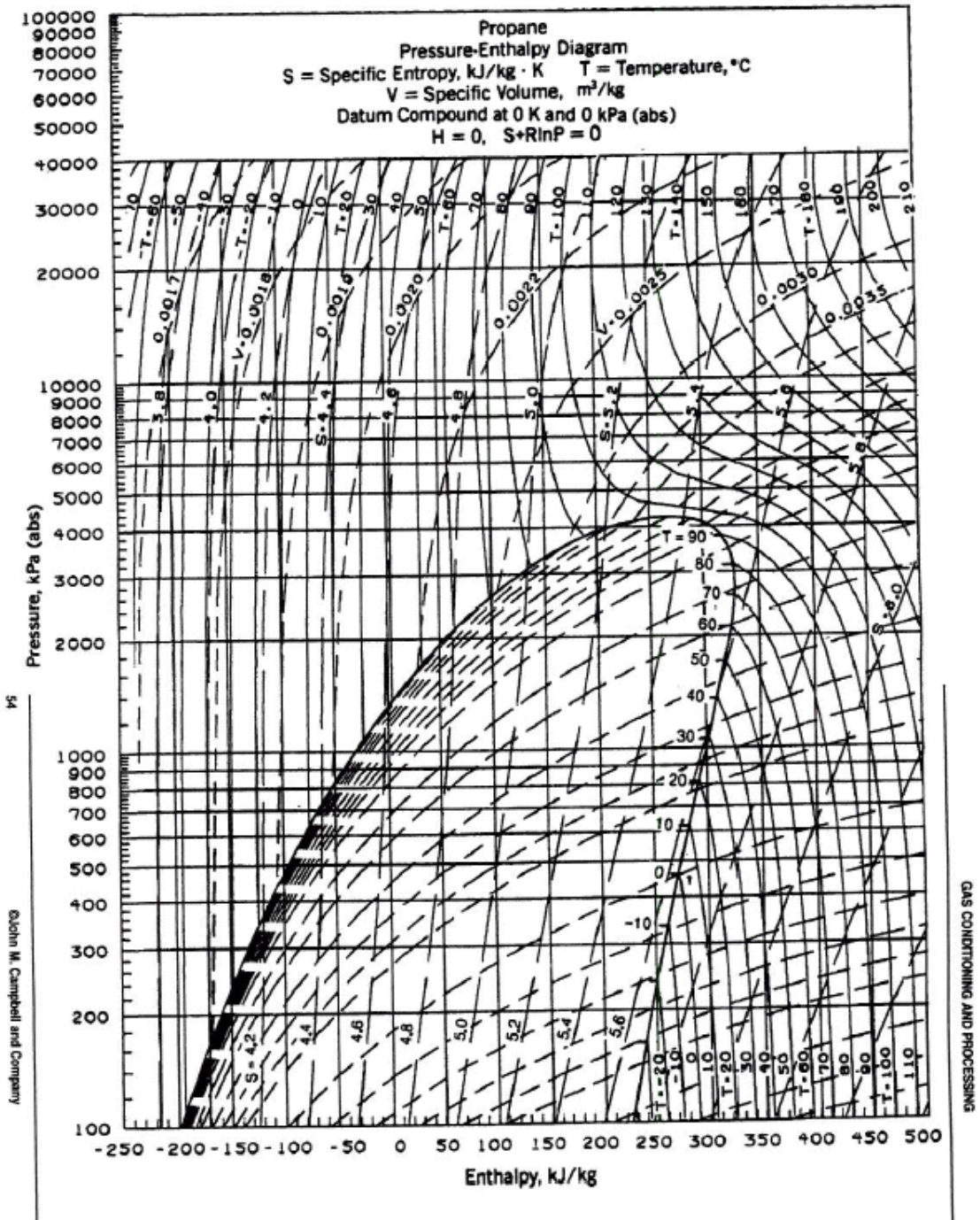


Figure 1.6: Pressure-Enthalpy Diagram
(Courtesy Thimm Engineering Inc.)



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Figure 1.7: Propane p-h Diagram
 (Courtesy Nigeria Liquefied Natural Gas Plant Operating Manual)

1.6 RESEARCH METHODOLOGY

The procedure that was followed in carrying out this research project is as given below:

- i) A technological / literature review of the design and operation of cooling towers has been provided to give a general overview of the subject. Also, work that has been done by other studies as well as work that has been done by this study to improve the performance of cooling towers, has been presented.
- ii) The applicable analytical theories are then presented.
- iii) Data to be used for the modelling and analysis were obtained from the Nigeria Liquefied Natural Gas (NLNG) plant. The following design data were obtained from the operating and engineering manuals of the NLNG plant.
 - The bulk water temperature.
 - The wet bulb temperature.
 - The mass flow rate of the cooling water.
 - The forced-draught fan speeds.
 - The air velocity through the tower cells.
 - The cooling tower height.

The following data were obtained from measurements at the NLNG plant:

- Cooling tower inlet (hot) water temperature.
- Cooling tower outlet (cold) water temperature.
- Relative humidity of the ambient air
- Volumetric flow rate of the cooling water supply
- Make-up water flow rate
- Dry-bulb temperature of ambient air

The following data were obtained from the relevant literatures / tables

- Enthalpy of air-water vapour mixture at wet bulb temperature.
- Enthalpy of inlet and outlet air from tower.
- Enthalpy of inlet and outlet water from tower.

- iv) The design of the proposed air conditioning system (circuit) of the cooling tower was then carried out. This involved the application of the various cooling tower and air conditioning theories based on specified design criteria / specifications.
- v) The modelling and theoretical analysis / evaluation of the cooling tower were carried out in order to determine the thermal performance with and without the air conditioning circuit. The modelling of the cooling tower, the air conditioning circuit and the gas liquefaction plant, were carried out using two simulation packages:
 - TRaNsient SYstem of Simulation (TRNSYS)
 - Engineering Equation Solver (EES)Using the above model results, the calculated results and data from the Nigeria Liquefied Natural Gas plant, the thermal performance the plant's cooling tower, as well the overall energy efficiency of the liquefied natural gas plant, was evaluated to determine the improvement or otherwise due to the conditioning of the inlet air to the cooling tower.
- vi) Finally, the conclusions drawn from this study and recommendations made are given.

1.7 DATA COLLECTION

Data for the modelling were obtained as follows:

- 1) The following design data were obtained from the Operating and Engineering manuals of the Nigeria Liquefied Natural Gas Plant. These data include:
 - The cooling tower inlet (supply) and outlet (exhaust) water temperature.
 - The cooling tower wet bulb temperature.
 - The mass flow rate of the cooling water.
 - The mass flow rate of air.
 - Ambient (dry-bulb) temperature.
 - The cooling tower forced-draught fan speeds.
 - The air velocity through individual tower cell.

- The cooling tower height.
 - The relative humidity
- 2) The following actual (site), real-time data were obtained from measurements at the NLNG plant:
- Cooling tower inlet water temperature.
 - Cooling tower outlet water temperature.
 - Average ambient air temperature.
 - Relative humidity of cooling tower inlet air.
- 3) The following data were obtained from the relevant literature or tables/charts:
- Enthalpy of air-water vapour mixture at wet-bulb temperature.
 - Enthalpy of inlet and outlet air from tower.
 - Enthalpy of inlet and outlet water from tower.

1.8 SUMMARY

In this chapter, the following were presented:

- The background, justification and benefits for this research study
- The problem statement, objectives and scope of the study
- The application of cooling tower in a natural gas liquefaction plant, which has been used as a case study for this research
- The research methodology that has been used.
- Data collection methodology used for this study.

The next chapter presents a review of cooling tower and gas dehydration technologies, previous studies on these areas and the novelty of this study.

CHAPTER 2

TECHNOLOGY / LITERATURE REVIEW

2.1 INTRODUCTION

In this chapter, a review of the general principles of cooling towers, the types of cooling tower and the thermal design principles of cooling towers, as well as liquid desiccant / glycol dehydration systems, are presented. Furthermore, previous studies that have been carried out on improving the performance of cooling towers, as well as the modifications and improvements that were made by this research, are presented. Also, a brief review of the cooling water circuit of the Nigerian Liquefied Natural Gas (NLNG) plant is presented in this chapter.

2.2 GENERAL PRINCIPLES OF COOLING TOWER

Although there are different types of cooling towers, all use the same principles and have similar components that provide cooling. A cooling tower and one or more heat exchangers comprise an open recirculation water-cooling system. In a continuous cycle, heated water is exchanged for cooled water. The cool water flows from the cooling tower into one or more heat exchangers. In most cases, the heat exchange process begins when cool water from the cooling tower flows through the tube side of a heat exchanger while at the same time, the hot process liquid flows through the shell side of the same exchanger. This cools the process liquid because its heat is transferred to the tube-side cooling water through the tube walls. After the cooling water is heated through this exchanger, it returns to the cooling tower, where it is cooled and then sent back to the heat exchanger to remove more heat from the process liquid (Rosaler, 1995) - Figure 2.1 shows the process flow of an open recirculation water-cooled system.

A cooling tower is usually composed of several tower cells that are in parallel and share a common sump.

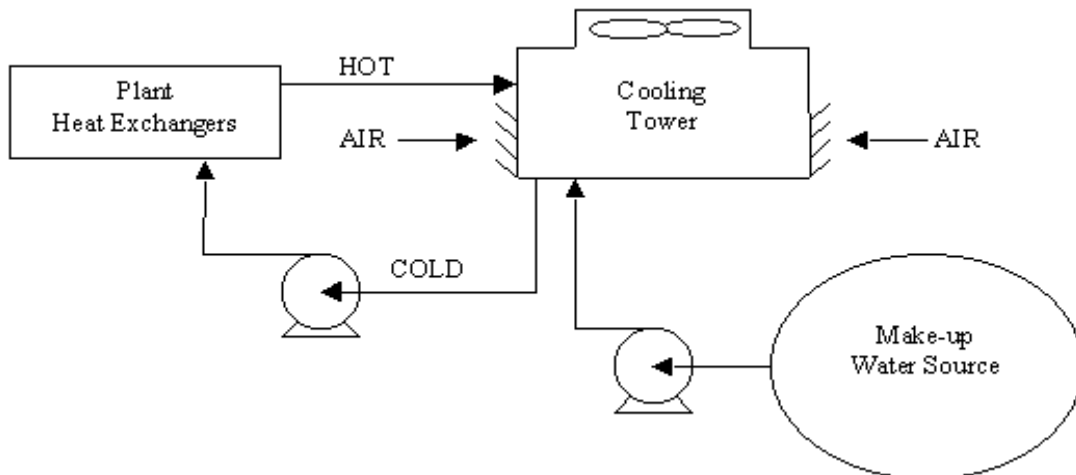


Figure 2.1: Closed Loop Cooling Tower System
(Courtesy www.cheresources.com)

A cooling tower cools warm water by exposing it to the air. This temperature reduction is the result of two processes: sensible heat transfer and evaporation (latent heat and mass transfer).

Sensible heat is the heat associated with a temperature difference. In a cooling tower, sensible heat is transferred from the warm water to the cooler air.

Evaporation is another form of heat transfer. During evaporation, when water changes from liquid to a vapour state, it takes latent heat energy with it. As a result, the temperature of the remaining liquid falls. But evaporation also causes a loss of the circulating water. For every cubic meter of water evaporated, a cubic meter of make-up water must be added to the system to maintain a constant volume of circulating water.

Sensible heat transfer accounts for 10 to 20% of the cooling effect in a cooling tower while evaporation accounts for 80 to 90%.

The rate of evaporation is the most important factor in cooling tower performance. Any condition interfering with evaporation, such as reduced air flow, can greatly reduce efficiency. However, the moisture content of the air, known in general terms as humidity, has the greatest effect on the rate of evaporation.

Humidity can be expressed in any of several measures such as absolute humidity, relative humidity, specific humidity and humidity or mixing ratio. Humidity is directly related to cooling tower performance. For example, relative humidity is the ratio, in percentage, of the moisture (water vapour) actually in the air to the moisture it would “hold” if it were saturated at the same temperature⁴. It is a useful index of “dryness” or “dampness” of air for determining evaporation, or absorption of moisture. If the relative humidity is 30%, the air has 30% of the total amount of water it can hold at its present temperature. A relative humidity of 100% means that the air can hold no more water at its current temperature. The lower the relative humidity of the air entering a cooling tower, the more the water evaporated and the colder the remaining bulk of water leaving the tower. More detailed discussion on humidity is presented in Section 2.12 of this chapter.

As the temperature of the air increases, so does its ability to hold or contain moisture. So the higher the temperature, the more water vapour the air can hold. Hence, temperature also affects cooling tower performance.

There are two ways to measure air temperature for cooling towers:

- i) The dry-bulb temperature (DBT) is the temperature as measured by a temperature sensor such as a dry-bulb thermometer.
- ii) The wet-bulb temperature (WBT) takes into consideration the moisture content of the air. The wet-bulb thermometer is similar to the dry-bulb thermometer, but its bulb is covered by a water-saturated wick. As the water evaporates it has a cooling effect. When the relative humidity of the air is 100% the wet-bulb temperature is the same as the dry-bulb

⁴ When one says that the air can “hold” a certain amount of water vapour, the fact that is being addressed is that a certain amount of water vapour can be resident in the air as a constituent of the air. The air does not “hold” water vapour in the sense of having some attractive force or capturing influence.

temperature, because the water cannot evaporate anymore. However, when the relative humidity is less than 100%, the wet-bulb temperature will be less than the dry-bulb temperature as the water will evaporate. So the wet-bulb temperature is lower than the dry-bulb temperature whenever the air is not saturated with water. The drier the air is, the greater the difference will be. The capability of a cooling tower is a measure of how close the tower can bring the water temperature to the wet-bulb temperature of the supply air. A detailed discussion on wet-bulb temperature is presented in Section 2.12 of this chapter.

2.3 TYPES OF COOLING TOWER

There are many types of cooling tower used in evaporative cooling but generally they tend to be divided into two groups depending upon the method used for moving air through the tower:

- a) Natural draft;
- b) Mechanical draft.

Either of these may be associated with counter-flow or cross-flow arrangements (Figures 2.2a and 2.2b).

A natural draft cooling tower depends on natural forces to move air through the cooling section of the tower. One type of natural draft cooling tower is called an atmospheric tower. It is a louvered enclosure containing a water spray system. The air enters through one louvered wall, passes horizontally over the packing, and exits through the opposite louvered wall. The rate of airflow depends on wind velocity.

Another type of natural draft cooling tower is the hyperbolic tower. In this case, the airflow is produced by density differences between the warm air inside the tower and the cooler air outside the tower. Designed for use primarily in power generating plants, hyperbolic towers are built for flow rates greater than about 32 m³/s. Hyperbolic towers effect cooling using either cross-flow or counter-flow. In a counter-flow tower, the water falls by gravity through the packing and is cooled as it descends. Air enters the bottom of the tower and flows upwards. In a cross-flow tower, the water also flows downwards through the packing, but the air flows

horizontally across the packing and the water. More air passes through the cross-flow tower, but evaporative heat removal is more efficient in the counter flow tower.

Mechanical draft towers use fans to help move the air. Fans speed the cooling process, and increase tower efficiency, by allowing a greater volume of air to flow over the falling water. The greater air volume obtained in a mechanical draft tower means more efficient cooling and more control over tower conditions. Multiple fans increase this effect.

The two types of mechanical draft towers are classified by their fan placement. First, there is the forced draft tower, where the fans are situated

at the bottom of the tower – at the air intake. The fans force the air through the tower in a counter flow pattern. While the air entering a forced draft tower moves at a higher velocity than a natural draft tower, the exiting air moves relatively slowly. This low exit velocity frequently results in the recirculation of warm air, which can reduce the efficiency of the tower up to 20% (CTI, 1975).

The second type of mechanical draft tower is the induced draft tower. In this case, the fans are on top of the tower. The air is pulled, or induced, through the fill and the water. Recirculation is of little concern since the top-mounted fans discharge the heated air well above the incoming air stream.

Mechanical draft towers, particularly the induced type, range from small, commercial air-conditioning systems to large, multi-cell units used for industrial plant cooling.

Mechanical draft towers offer control of cooling rates in their fan diameter and speed of operation. These towers often contain several areas (each with their own fan) called cells (Cheresources.com, 2002).

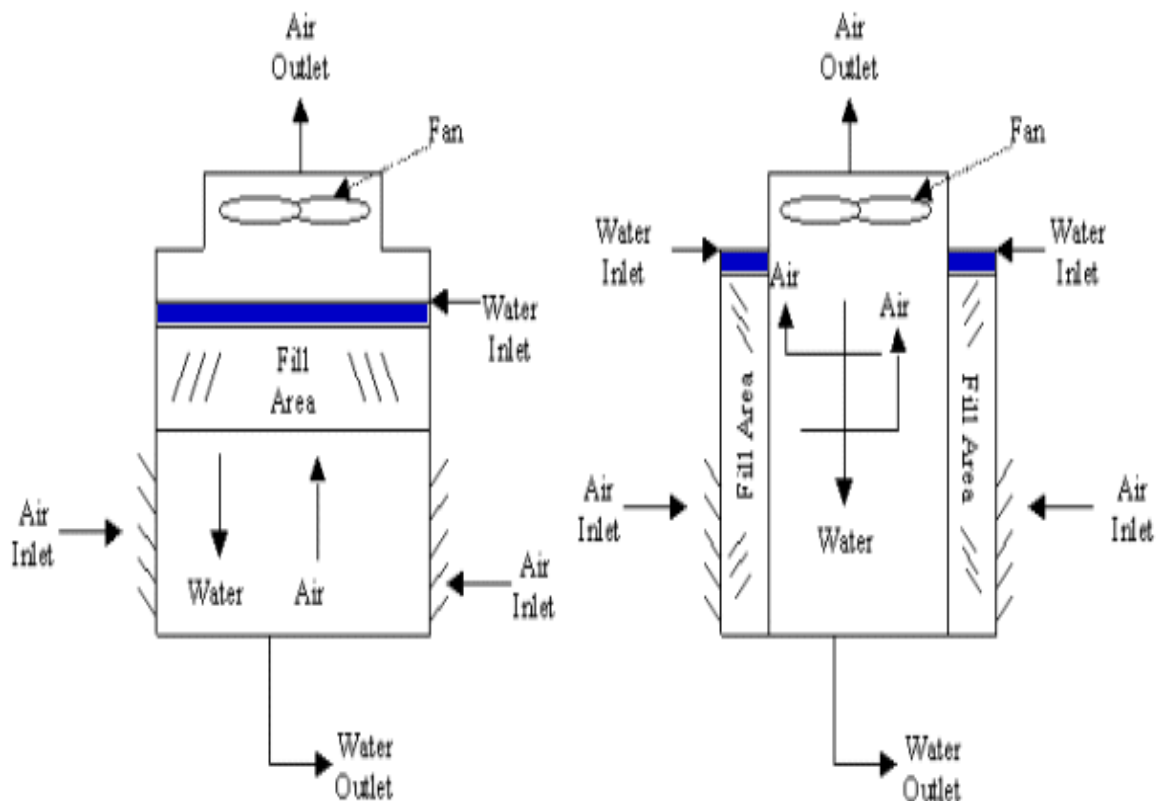


Figure 2.2(a): Mechanical Draft

Figure 2.2(b): Mechanical Draft

Counterflow Tower

Crossflow Tower

(Courtesy www.cheresources.com)
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2.4 COOLING TOWER COMPONENTS

Although there are different types of cooling towers, all have similar components that provide efficient cooling.

All cooling towers have the following: a water distribution system which spreads the hot water evenly over the largest possible surface area; packing⁵ or splash bars⁶

⁵ Packing consists of thin, closely spaced plastic surfaces over which the water spread, forming a thin film in contact with the air. These surfaces may be flat, corrugated or other patterns.

which aid evaporation by exposing as much water surface as possible to the air; air intake louvers to efficiently direct the air flow through the packing; and drift eliminators to remove moisture carried in the exit air leaving the tower (Figure 2.3).

Other important cooling tower equipment includes water circulation pumps, air fans, motors and/or turbines drivers (for the pumps and fans).

The materials used to construct a cooling tower must be able to withstand the effects of constant temperature fluctuations and destructive water-borne impurities.

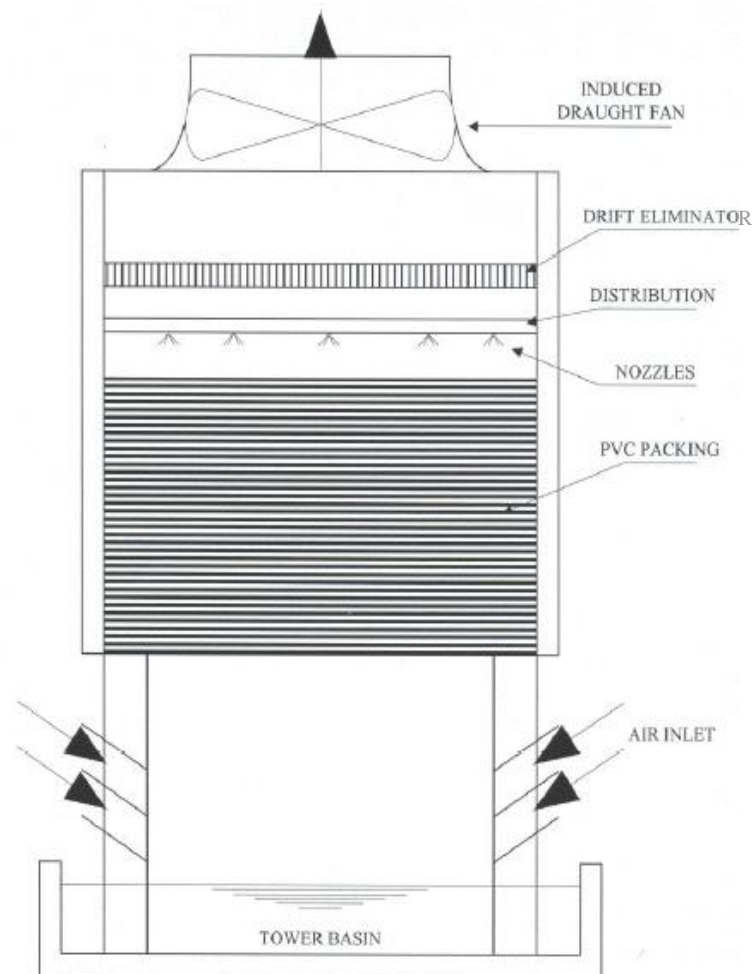


Figure 2.3: Cooling Tower Components
(Courtesy NLNG Operating Manual)

⁶ Water falls over successive layers of horizontal splash bars, continuously breaking into small droplets. The splash bars are made of plastics or woods. Packing and splash bars are generally referred to as “fill”.

2.5 COOLING TOWER THERMAL DESIGN PRINCIPLES

As stated earlier, a cooling tower is a heat exchanger in which warm water falls gravitationally through a cooler current of air. The transfer of heat is effected from the water through the boundary film of saturated air in contact with the water surface. This air is saturated at the water temperature. From this saturated air film, heat transfer occurs to the general mass of air flowing through the tower (Green, 1994).

In the interest of efficiency, it is essential that both the area of water surface in contact with the air and the time of contact be as great as possible. This may be achieved by forming a large number of water droplets as repetitive splash effects in one basic kind of tower packing, or by leading the water in a thin film over lengthy surfaces. The direction of air travel may be opposed to the direction of water flow giving counter-flow conditions, or may be at right angles to the flow of water giving cross-flow conditions. Although the method of analysis may be different for counter-flow and cross-flow conditions, the fundamental heat transfer process are the same in both cases.

Present cooling tower technology relies on the fact that, with acceptable error, the effects of evaporative and sensible heat transfer can be combined into one dependent on enthalpy difference. The difference concerned is that between the enthalpy of the boundary film of air surrounding the water surface (taken to be at water temperature) and the enthalpy of the general mass of air flowing through the tower. This enthalpy difference varies according to the point of measurement in the tower, but at all points it provides the enthalpy potential or driving force for the heat transfer (BS 4485, 1988). Figure 3.2 in Chapter 3 shows details of this process.

The combined heat transfer theory depends upon certain approximations (assumptions), which are reasonable at normal cooling water temperatures and particularly when the characteristics of the packing have been determined in accordance with the theory. These approximations are: (1) the flow rate of water from the cooling tower is assumed to be the same as the flow rate of water supply to the cooling tower – the effect of water loss due to evaporation is neglected and, (2)

the Lewis number⁷ is assumed to be unity. However, the approximations become progressively less valid with increasing water temperature and a more exact analysis is adopted in applications where the mean water temperature exceeds 35°C (BS 4485, 1988). A more detailed discussion of the cooling tower theory is presented in Chapter 3.

The air and air-water film condition, in passage through the tower, may be illustrated on a psychrometric chart (Figure 2.5). The cooling range of the tower corresponds to the difference in temperature of the air-water film between entry to and exit from the tower.

Air enters the tower having wet and dry bulb characteristics dependent on the ambient conditions. It is generally in an unsaturated state and achieves near-saturation in passing through the tower. It may be considered saturated at exit in all but very dry climates.

The enthalpy of the entering air is considered, with acceptable error, to be equivalent to the enthalpy of air saturated at the entry wet bulb temperature (since lines of enthalpy and wet bulb are almost coincident on a psychrometric chart). For the purposes of enthalpy differences in heat transfer, only the wet bulb temperature of the ambient air is therefore of significance. The dry bulb temperature has, however, to be considered for draught assessment purposes in those natural draught towers whose air flow relies on thermally created draught. The thermal draught is defined by the change of density of the air between entry to and exit from the cooling tower multiplied by the effective shell height, and the dry bulb temperature as well as the wet bulb temperature is of significance in determining the density of ambient air (BS 4485, 1988).

The theoretical limit to which the water may be cooled is that of the ambient wet bulb temperature. This could only be achieved with an infinitely large tower and, in practice, the exit (cold) water temperature has some approach⁸ to the wet bulb temperature. This approach may vary from about 3K in rigorous chemical plant

⁷ Lewis number is a dimensionless number defined as the ratio of thermal diffusivity to mass diffusivity. It is used to characterise fluid flows where there is simultaneous heat and mass transfer by convection.

⁸ The “approach” is the difference between the temperature of the re-cooled water exiting the cooling tower and the wet-bulb temperature of the air entering the cooling tower.

cooling to 10K or above in easier conditions and increases or decreases as ambient conditions vary from cold to warm, respectively, about the design point (CTI, 1975).

2.6 FACTORS INFLUENCING COOLING TOWER

PERFORMANCE

The thermal performance of cooling towers depends mainly upon the amount of evaporation of water that takes place. The amount of evaporation, in turn, depends upon the following factors:

- 1) The amount of water surface exposed to the air.
- 2) The length of the exposure time.
- 3) The water-to-air ratio and the velocity of air passing over the water droplets formed in the cooling tower.
- 4) The wet-bulb temperature of the atmospheric air.

The temperature and the moisture content of the air are increased as the air passes through the tower. The wet-bulb temperature of the atmospheric air entering the cooling tower is theoretically the lowest temperature to which the water can be cooled. For this reason, the wet-bulb temperature determines the minimum operating temperature level throughout a water-cooled plant, process or system. The cooling capacity of the cooling tower increases as the wet-bulb temperature of the inlet air decreases (Khurmi, 2004). Hence, the thermal performance of the cooling tower depends principally on the wet-bulb temperature of the inlet air. This is evident also from the cooling tower heat transfer equations given in Chapter 3 (Section 3.3). When the wet-bulb temperature of the air decreases, the air can hold more water vapour and therefore increase the amount of evaporation of water that takes place in the cooling tower. When operating at the design wet bulb temperature, the tower should produce water of the range and at the temperature specified. When the wet bulb temperature falls, however, it is the same as increasing the potential in the tower. If the heat load on the tower and the weight of water and air circulated through the tower are all to be kept constant, the water still undergoes the same range of cooling in it but the inlet and outlet temperatures will be colder. The cooling effect is therefore dependent on the initial (inlet) wet-bulb temperature of the cooling air; the lower the wet-bulb temperature, the higher the water-cooling capability of the

cooling tower.

The aim of this research is to improve the thermal performance of cooling towers by investigating the new concept of conditioning the supply air in order to reduce its wet-bulb temperature. This involves decreasing the moisture content of the inlet air to the cooling tower (control of humidity).

As the moisture content of the air decreases, the wet-bulb temperature of the air decreases and hence the air can hold more evaporated water as it flows through the cooling tower. The more the water evaporated by the air, the colder the remaining bulk water leaving the cooling tower.

Thus, the heat transfer of the cooling tower increases as the wet-bulb temperature of air decreases. Figure 2.4 is a simulation plot showing the variation of water temperature to wet-bulb temperature of air. There is approximately a 2 K reduction in temperature of water for every 5 K reduction in the wet-bulb temperature of air.

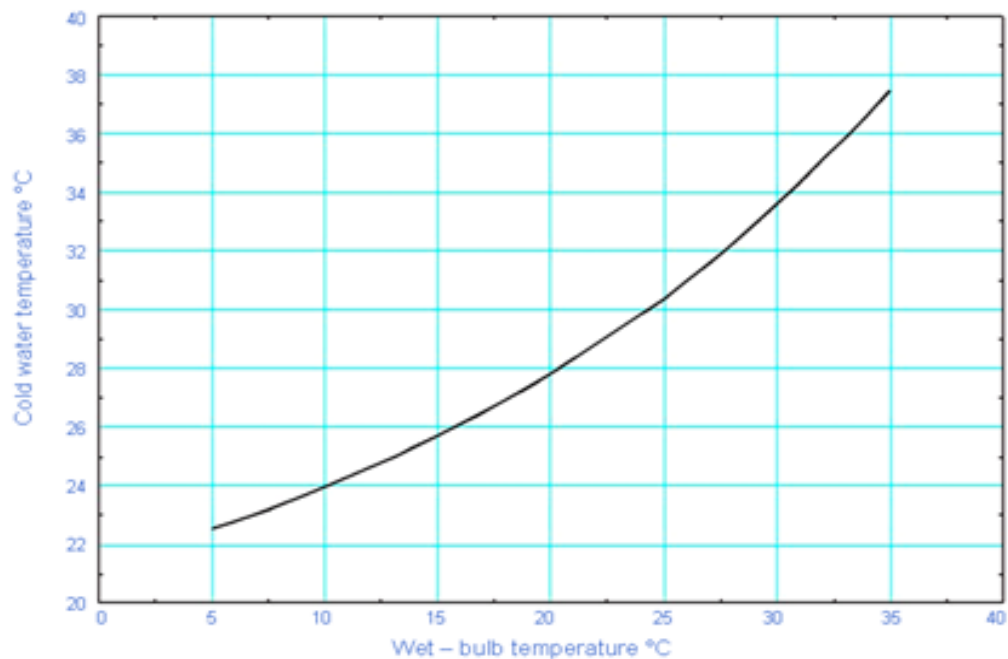


Figure 2.4: Variation of Water Temperature with Wet-bulb Temperature of Air

2.7 PREVIOUS STUDIES ON ANALYSIS AND PERFORMANCE

IMPROVMENT OF COOLING TOWERS

The development of cooling tower analysis began in 1922, with the first work due to Robinson (1923), who established the principle applicable to cooling tower design and derived equations for the use of the designer. Robinson was the first to use Lewis' work (1922), who developed a series of fundamental concepts about the mechanism involved in the transfer of heat between liquid and gases and in the vaporization of liquids.

The first practical theory of cooling tower operation was developed by Merkel (1925), who used the enthalpy potential as a driving force for air-water heat exchange and assumed a similarity between heat and mass convective transfers by assuming Lewis number of one⁹. Merkel's method has been the basis of most cooling tower analysis. His analysis and simplification have been used until today.

More recently, several authors have studied the convection phenomena occurring in cooling towers. Baker and Shryock (1961) developed a detailed explanation of the concept of cooling tower performance, clarifying the assumptions and approximations used by Merkel.

Sutherland (1983) showed that Merkel's theory leads to an underestimation of tower volume from 5% to 15%. In practice, however, the errors are not nearly as large, because the mass transfer coefficients utilised in the Merkel method are generally determined by matching results from models to small-scale tests.

Braun (1989) developed a unified method to model the performance of both cooling towers and dehumidifying coils. Effectiveness relationships were developed, taking into account a saturated air-specific heat used for sensible heat exchangers. The results of this method were compared with numerical solutions of detailed heat and mass transfer models and experimental results.

Shelton and Weber (1991) used mathematical models based on manufacturer's performance data in order to investigate the effect of water and air flow rates.

Bernier (1994) presented an analysis of basic heat and mass transfer processes

⁹ Assuming a Lewis number of one infers that the air enthalpy varies linearly with the water temperature in the cooling tower. Hence, the enthalpy change of the air can be equated to the (Footnote continued on next page)

occurring around a droplet in transient cooling. A unidimensional analysis of the idealized spray-type counterflow tower was presented, showing the influence of fill height, water retention time, and water-air flow ratio on the tower performance.

Lebrun et al. (1993) presented the fundamentals of a new simulation toolkit orientated towards simple solutions for primary heating, ventilation and air conditioning (HVAC) equipment. This toolkit uses, as much as possible, the literature models. Each model is described by reference to a conceptual scheme in which very classical engineering components are interconnected. The cooling tower proposed there is no more than a generalization of Braun's approach (1989), in which air enthalpies are replaced by wet-bulb temperatures as driving forces.

Several design concepts have been developed in previous studies with the aim of improving the performance of cooling towers. Most of these previous studies have been focused on improving the design of the cooling tower components, such as the packing, water distribution nozzles and air fans. Other studies have looked at improving the chemical treatment of the water as a way of increasing the air-water contact and hence performance of cooling towers. Lemouari et al (2006) investigated the thermal performance improvement of cooling towers using Vertical Grid Apparatus (VGA) type packing in their work titled "Thermal Performances Investigation of a Wet Cooling Tower". The VGA packing consists of vertical grids arranged in a zigzag form. This study investigated the effect of air and water flow rate on the cooling water range as well as the tower characteristics, for different inlet water temperatures (35 °C, 43 °C and 50 °C respectively). The results showed that the tower characteristic, KaV/L , decreases with an increase of the water-to-air mass flow ratio, L/G . It was also found that the cooling water range increases with an increase in air flow rates, while it decreases with an increase in water flow rates. The highest values of cooling range were reached for lower values of L/G .

Goshayshi and Messeden (1999) in their work "The Investigation of Cooling Tower Packing in Various Arrangements", investigated the effect of form with corrugated packing on mass transfer and pressure drop characteristics in atmospheric cooling towers. They conducted experiments to investigate the effect of spacing and surface

temperature change of the water. This simplifies the cooling tower combined heat and mass transfer analysis.

roughness on the mass transfer and pressure drop in polyvinyl chloride (PVC) packing. The experiments were carried out for comparative types of packing in a counterflow cooling tower. The results showed that the overall mass transfer coefficient, K , and pressure drops of ribbed corrugated packing increased considerably compared with smooth packing and are affected by the spacing of the packing and the distance between the ribs. The values of K for corrugated packing were 1.5 – 2.5 times higher than comparable smooth packing K values when the water-to-air ratio was 1.0. Also it was found that the shape and configuration of the roughness projections are as important as the height of those projections in determining their effect on the mass transfer coefficient. The study concluded that the volumetric overall mass transfer coefficient based on enthalpy difference changes not only with the mass flow rate of air and water and temperature conditions, but also with the dimensions, shape and arrangement of the cooling tower packing.

Gharagheizi et al (2006) did an experimental study on the performance of mechanical cooling towers with Vertical Corrugated Packing (VCP) and Horizontal Corrugated Packing (HCP) designs in their work titled “Experimental Study on the Performance of Mechanical Cooling Tower with Two Types of Film Packing”. In this work, an experimental and comparative study in terms of tower characteristic (KaV/L), water-to-air flow ratio (L/G) and efficiency for the VCP and HCP type packing were carried out for L/G ratio of 0.2 to 4. From the results of this work, it was concluded that the performance of the cooling tower is affected by the type and arrangement of the packing. The HCP caused lower water loss during the cooling process in the tower than the VCP. The results showed the tower with vertical corrugated packing (VCP) had higher efficiency than the one with horizontal corrugated packing (HCP). The obtained results also showed a decrease in the tower performance with an increase in L/G .

A system for improving the performance of cooling towers by optimising the control of the water temperature within a cooling tower has also been designed and patented¹⁰ by Kaya and Moss (1983). The temperature and the relative humidity of the air outside the cooling tower are measured and are selectively utilized as inputs to

¹⁰ United States Patent Number 4474027.

a plurality of modules containing function blocks in a logic arrangement. The function blocks produce an output signal indicative of the wet bulb temperature of the outside air, which is compared to the cooling tower water temperature, causing the speed of the cooling tower fans to be adjusted in response to the temperature difference there-between.

It is noted, however, that none of these studies have looked at the idea of *modifying* the physical properties of the air in order to improve the cooling tower thermal performance. This is the novelty of this present study, which proposes to improve the thermal performance of cooling towers by conditioning the supply air by way of reduction in its wet-bulb temperature.

2.8 CONCEPT OF THIS RESEARCH -

CONDITIONING OF COOLING TOWER INLET AIR

As stated earlier in Chapter 1, the aim of this research is to improve the thermal performance (cooling capability) of cooling towers by conditioning the wet-bulb temperature of the inlet air.

In essence, the research involves the design and development of an air-conditioning system, as an integral part of the cooling tower, for dehumidification (conditioning) of the inlet air to the cooling tower.

The design of the cooling tower air-conditioning system involves the application of air conditioning theory based on the use of liquid desiccant in a vapour-absorption cycle. Specifically, the control of the cooling tower inlet air moisture content (humidity ratio) and, hence, wet bulb temperature, is achieved by the use of glycol dehydration processes.

A theoretical study of the simultaneous heat and mass transfer between air and liquid desiccant was carried out by modelling the system. The modelling results (output) were compared with real-time, actual site data from the NLNG plant. In addition, the gas liquefaction plant was modelled in order to analyse its overall energy consumption with and without the proposed cooling tower air conditioning system.

A comprehensive survey of the performance characteristics of the cooling towers at

the Nigeria Liquefied Natural Gas Plant and other “actual” operating data¹¹ gathered during the research project formed an important source of information for the following purposes:

- Calibration of newly developed models.
- Validation of results obtained from model simulations.
- Prediction of both design and off-design conditions performance of cooling towers employing the calibrated / validated numerical models.

2.9 MODELLING AND SIMULATION OVERVIEW

A model is a representation of the construction and working of some system of interest. A model is similar to, but simpler than, the system it represents; it is a close approximation to the real system and incorporates most of its salient features. A good model is a judicious trade-off between realism and simplicity. Simulation practitioners recommend increasing the complexity of a model iteratively. On the other hand, it should not be so complex that it is impossible to understand and experiment with it. One purpose of a model is to enable the analyst to predict the effect of changes to the system.

Modelling is the process of producing a model. A key issue in modelling is model calibration and validation. Model calibration consists of changing values of model input parameters in an attempt to match field conditions within some acceptable criteria. This requires that field conditions at a site be properly characterised. Lack of proper characterisation may result in a model that is calibrated to a set of conditions that are not representative of actual field conditions. The calibration process typically involves calibrating steady-state and transient (dynamic) conditions. With steady-state simulations, there are no observed changes with time for the field conditions being modelled. Transient simulations involve the change of parameters with time (Michigan.gov, 2008). Model validation techniques include simulating the model under known input conditions and comparing model output with system output.

A simulation of a system is the operation of a model of the system. The model can be

¹¹ A 1-year actual real-time operating data of the NLNG plant, obtained from site for the period of (Footnote continued on next page)

reconfigured and experimented with; usually, this is impossible, too expensive or impracticable to do in the system it represents. The operation of the model can be studied, and hence, properties concerning the behaviour of the actual system or its sub-system can be inferred. In its broadest sense, simulation is a tool to evaluate the performance of a system, existing or proposed, under different configurations of interest and over long periods of real time. In a simulation study, human decision making is required at all stages, namely, model development, experimental design, output analysis, conclusion formulation and making decisions to alter the system under study. The only stage where human intervention is not required is the running of the simulations, which most simulation software packages perform efficiently. The steps involved in developing a simulation model, designing a simulation experiment and performing simulation analysis are (1) Identify the problem, (2) Formulate the problem, (3) Collect and process real system data, (4) Formulate and develop a model, (5) Calibration / Validate the model, (6) Document model for future use, (7) Select appropriate experimental design, (8) Establish experimental conditions for runs, (9) Perform simulation runs, (10) Interpret and present results and (11) Recommend further course of action. Although this is a logical ordering of steps in a simulation study, many iterations at various sub-stages may be required before the objectives of a simulation study are achieved. Not all steps may be possible and/or required. On the other hand, additional steps may have to be performed (Anu, 1997).

Generally, a model intended for a simulation study is a mathematical model developed with the help of simulation software. Mathematical models usually describe a system by a set of variables and a set of equations that establish relationship between the variables. The actual model is the set of the functions that describe the relations between the different variables. The values of the variables can be practically anything, e.g., real or interger numbers. These variables are usually abstractions of quantities of interest in the described systems, and they are acted upon by mathematical operators, which can be algebraic, functions, differential operators, etc. If all the operator in a mathematical model present linearity, the

resulting mathematical model is defined as linear. A model is considered non-linear otherwise (Amu, 1997).

Mathematical model classifications include deterministic (input and output variables are fixed values) or stochastic (at least one of the input or output variables is probabilistic); static (time is not taken into account) or dynamic (time-varying interactions among variables are taken into account). Dynamic models typically are represented with differential equations.

Details of the modelling and simulation aspects of this study are presented in Chapter 4.

2.10 LIQUID DESICCANT AIR CONDITIONING

The liquid desiccant (vapour-absorption) air-conditioning was selected for the design of the proposed cooling tower air-conditioning system because it is an energy efficient air system compared to the conventional vapour-compression system (Khurmi, 2004). Desiccants are materials that naturally attract moisture from gases and liquids (hygroscopic). The material becomes saturated as moisture is absorbed or collects on the surface; but when heated, the desiccant dries out (regenerates) and can be used again. Desiccants are unique in that they can dry air without first cooling the air below its dew point. Conventional solid desiccants include silica gel, activated alumina, lithium chloride salt, molecular sieves, titanium silicate and synthetic polymers. Liquid desiccants include lithium chloride, lithium bromide, calcium chloride and triethylene glycol solutions. Desiccant dehumidifiers remove moisture from the air. Liquid desiccant systems have a number of features that make them a preferred choice. It is best for services where latent loads are large relative to sensible loads and also for applications that place a high value on a liquid-desiccant system's unique ability to suppress microbial growth. In the context of waste-heat recycling, they can make effective use of low temperature waste streams that other technologies cannot, like the cooling water from engine generator sets, etc. Their air-conditioning and regeneration functions can be physically separated, so the humidity-absorbing solution can be regenerated at any onsite heat source and conveniently piped to the area where dehumidification is needed. Thus, liquid desiccant systems provide dehumidification and cooling at a lower cost than competing technologies

while using less energy (AIL Research, 2006).

Under humid outdoor conditions, conventional electric vapour compression cooling systems are not capable of removing the moisture without first cooling the air down below the dew point temperature and then reheating in the conditioned space. This method would result in excessive energy requirements and higher utility demand /cost.

Desiccant systems are growing in popularity because of their ability to independently control humidity levels (latent loads), thereby allowing conventional air-conditioning systems to primarily control temperature (sensible loads) (AIL Research, 2006).

Air-conditioning is usually carried out using vapour compression systems. However, increased global warming and the environmental impact of chlorofluorocarbon (CFC) and other similar refrigerants on the ozone layer has stimulated interest in developing “environmentally-friendly” air-conditioning systems. Liquid desiccants employ packed beds, packed towers, spray chambers or sprayed coil units. These systems have several advantages including lower pressure drop of air across the desiccant material, suitability for dust removal by filtration, lower regeneration temperature and ease of manipulation (AIL Research, 2006).

A liquid desiccant air-conditioner has three main components: the absorber (also called the contactor), the regenerator and the interchange exchanger (Figure 2.5).

In the absorber, air is dried as it passes up through packing sprayed with cooled, concentrated liquid desiccant. The air leaves the absorber drier and at a much lower wet-bulb temperature.

In the regenerator, heated desiccant is sprayed down over another packing. Heat is supplied in the regenerator by a gas-fired boiler, furnace, solar thermal collectors, recovered heat from an engine or fuel cell, or other energy source (this is the main energy driving the dehumidification process). As the temperature of the desiccant increases, water evaporates into an air stream that is then discharged outdoors.

The hot, concentrated desiccant that leaves the regenerator and the cool, diluted desiccant that flows to the regenerator exchange thermal energy in the interchange heat exchanger. This exchange increases the efficiency of the regenerator and decreases the cooling load on the absorber. The efficiency of the regenerator can also

be increased by adding an air-to-air heat exchanger to pre-heat the air that enters the regenerator using the warm, humid air that leaves it.

In both the absorber and the regenerator, desiccant dripping from the packing (or trays) is collected in a sump. Pumps are used to produce the sprays in the absorber and regenerator and drive the desiccant through the heat exchanger. The absorber and regenerator are usually fitted with mist eliminators on their outlet to conserve desiccant.

Although many liquids have desiccant properties, glycols and solutions of halide salts, particularly lithium chloride and calcium chloride, are the most viable liquid desiccants for HVAC applications. However, the high chloride concentrations in solutions of the halide salts eliminate even most stainless steels from service in contact with the desiccant and hence it is maintenance intensive. If maintenance is to be acceptable, all wetted surfaces of liquid desiccant air-conditioner (LDAC) using chloride solutions, should be a plastic with suitable properties (AIL Research 2006). Glycols, on the other hand, are much less corrosive than the chloride solutions. Furthermore, they are highly hygroscopic, do not solidify in a concentrated solution and are easily regenerated to a high concentration. However, all glycols have significantly high vapour pressures at regeneration temperatures. This means significant amounts will evaporate into the scavenging air as it carries away the water produced by the dehumidifier. Hence, a high glycol make-up cost is incurred. However, the loss of glycol can be reduced or eliminated by the use of high-efficiency mist eliminators. Another draw back of glycol desiccants is that they degrade faster in the presence of oxygen. However, this can be mitigated by efficient temperature control in the regenerator (AIL Research, 2006)

Considering the advantages and disadvantages of the various liquid desiccants, this research has opted to use glycol desiccant for the design of the cooling tower air-conditioner.

Several of the glycols possess the advantages outlined above. Diethylene (DEG), triethylene (TEG) and tetraethylene (TREG) glycols all possess suitable traits. However, almost 100% of the glycol dehydrators use TEG.

DEG is somewhat cheaper to buy and sometimes is used for this reason. But, compared to TEG, DEG has a larger carry-over loss, offers less dew point depression

and regeneration to high concentration is more difficult.

TREG is more viscous and more expensive than the others. The only real advantage is its lower vapour pressure which reduces absorber carry-over loss. It may be used in those relatively rare cases where glycol dehydration will be employed on a gas whose temperature exceeds about 50°C (Campbell, 2004).

TEG was used in this research study, even though some of the system characteristics also apply for all glycols. TEG is not considered to be a health hazard when handled under normal industrial conditions. It has few irritating effects and has a much lower acute and chronic oral toxicity than the lighter glycols. It is, however, harmful if swallowed. TEG is also safer at high temperatures as it is less flammable. Appendix 7 gives the physical properties of TEG and the other glycols.

The primary energy input to a liquid-desiccant air conditioner is heat for regeneration. The electrical demand for pumps and fans will be typically less than one-fifth that for a conventional compressor-driven electric air conditioner. As stated earlier, the heat needed to regenerate the desiccant can come from many sources (AIL Research, 2006).

2.11 THE BASIC GLYCOL DEHYDRATION UNIT

Figure 2.5 shows the basic glycol dehydration (dehumidifier) unit.

The wet gas (air), free of liquid water, enters the bottom of the absorber (contactor) and flows counter-current to the glycol. Glycol-gas contact occurs on trays or packing. The dried air leaves the top of the absorber.

The lean glycol (low water content) enters on the top tray or at the top of the packing and flows downward, absorbing water from the counter-current gas stream as it goes. It leaves rich in water. At the bottom, both the entering air and glycol leaving are rich in water; at the top end both are lean in water. The rich glycol (high in water content) leaves the bottom of the absorber and flows to a reflux condenser at the top of the still column (providing reflux cooling in the still column). The rich glycol then enters a flash tank where most of the volatile components (entrained and soluble) are vaporised. Flash tank pressures are typically 300-700kPa. Leaving the flash tank, the rich glycol flows through the glycol filters and rich-lean exchanger where it exchanges heat with the hot lean glycol. The lean glycol then enters the still column

where the water is removed by distillation.

The still column and the reboiler are often called the regenerator or reconcentrator. This is where the glycol concentration is increased to the lean glycol requirement (specification). Heat is provided at the bottom of the regenerator in order to evaporate water from the glycol. The reboiler may be directly fired or indirectly heated by electricity, hot oil or steam. Water and volatile species present are evaporated from the rich glycol, and reflux is provided to reduce glycol losses.

The regenerated hot lean glycol leaves the reboiler, flows into the surge tank from where it is pumped through the hot side of the rich-lean glycol exchanger and a lean glycol cooler into the absorber.

The regeneration unit is usually designed to operate at the prevailing atmospheric pressure. The initial thermal decomposition temperatures¹² of the glycols are given in Table 2.1 (Campbell, 2004)

Table 2.1: Decomposition temperatures of glycols

Glycol	Temperature
Ethylene (EG)	165°C
Diethylene (DEG)	164°C
Triethylene (TEG)	206°C
Tetraethylene (TREG)	238°C

¹² The *decomposition temperature* of a substance is the temperature at which the substance decomposes (break up) into smaller substances or into its constituent atoms when heated.

At the pressures involved, Raoult's Law¹³ applies (Campbell, 2000):

$$x_w = \left(\frac{P}{P_v} \right) y_w \quad 2.1$$

where: x_w = mol fraction of water in lean glycol

P = system pressure

P_v = water vapour pressure at reboiler temperature

y_w = mol fraction of water in the reboiler vapour (in equilibrium with x_w)

Once the highest allowable temperature has been used (to maximise vapour pressure, P_v), " P " and/or " y_w " must be lowered if lower water concentrations (in the lean glycol) are needed. A vacuum pump or ejector may be used to lower the system pressure, P .

The addition of some vapour (from other sources) to the reboiler will decrease the " y " for water; this may be accomplished with a stripping gas. Any inert gas is suitable (e.g., nitrogen); the quantity required is small. It is common to use a distribution pipe along the bottom of the reboiler for the supply of the stripping gas.

Once the components in the solution have reached chemical equilibrium, the total vapour pressure of the solution is:

$$P_{\text{solution}} = p_{1 \text{ pure}} x_1 + p_{2 \text{ pure}} x_2 + \dots \quad 2.1(a)$$

and the individual vapor pressure for each component is

$$P_i = p_{i \text{ pure}} x_i \quad 2.1(c)$$

where

$p_{i \text{ pure}}$ is the vapour pressure of the pure component

x_i is the mole fraction of the component in solution

¹³ Raoult's law states: the vapour pressure of an ideal solution is dependent on the vapour pressure of each chemical component and the mole fraction of the component present in the solution.

Consequently, as the number of components in a solution increases, the individual vapor pressures decrease, since the mole fraction of each component decreases with each additional component.

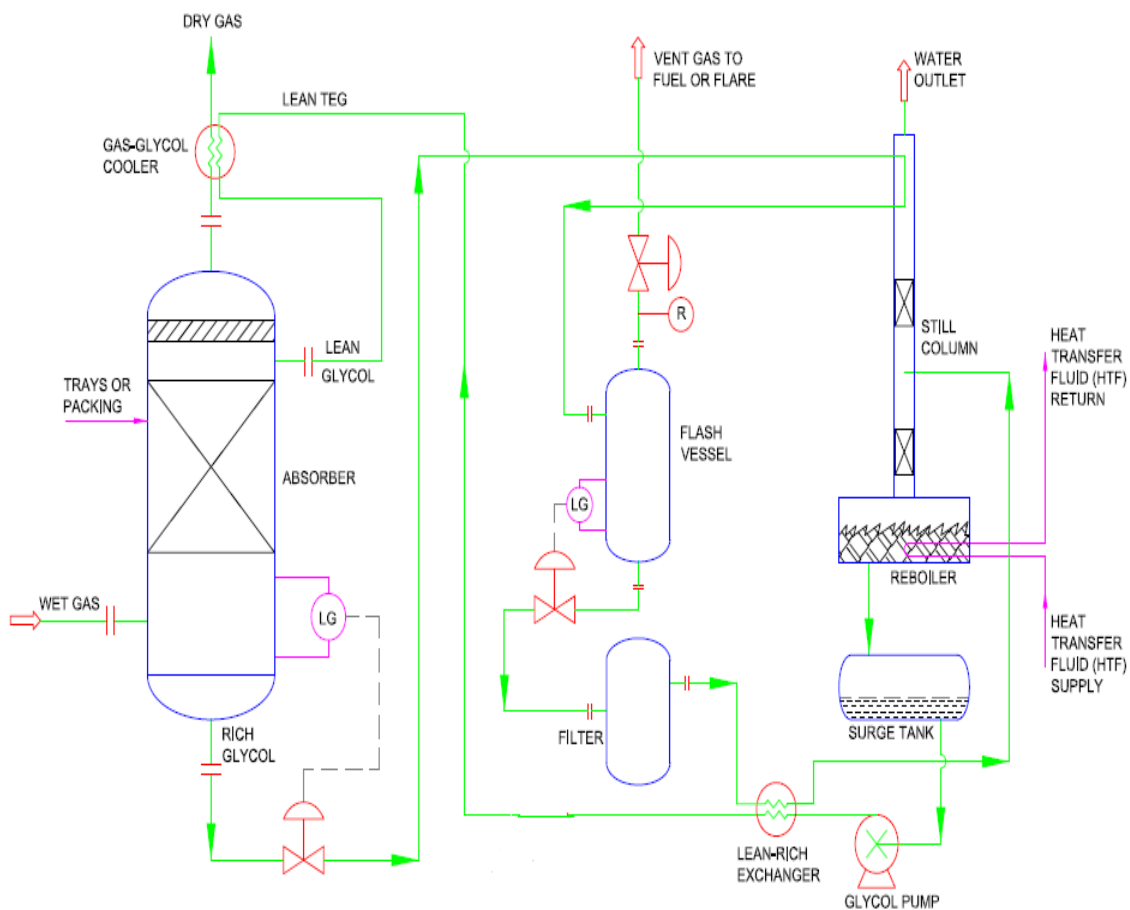


Figure 2.5: Basic Glycol Dehydration Unit
(Reproduced courtesy J. Campbell)

2.12 PSYCHROMETRY (HYGROMETRY)

This is the science of humid (moist) air and its properties. It deals with the physical and thermodynamic properties of air-water vapour mixtures. Moist air is treated as a mixture of independent perfect gases (comprising of dry air and water vapour), each obeying the perfect gas equation (Khurmi, 2004).

2.12.1 Properties of Moist Air

Several fundamental properties are associated with moist air. Two independent properties plus the barometric pressure are required to establish the thermodynamic state. Commonly used psychrometric properties are as follows (McLean, 2009):

- *Humidity Ratio* (also known as *Moisture Content*, or *Mixing Ratio*) – It is the ratio of the mass of water vapour to the mass of dry air in a given volume of the mixture (at the given conditions: dry-bulb temperature, wet-bulb temperature, dew point, relative humidity, etc) and it is denoted by the symbol ω . That is,

$$\omega = \frac{m_s}{m_a} = \frac{V_a}{V_s} \quad 2.2$$

Subscript “s” denotes water vapour while subscript “a” denotes dry air. v_a and v_s are the specific volumes of the dry air and water vapour respectively. Using the perfect gas equation, the humidity ratio may also be obtained as:

$$\omega = 0.622 x \frac{P_s}{P - P_s} \quad 2.3$$

where P is the total barometric pressure and P_s is the partial pressure of the water vapour.

Specific humidity is the ratio of mass of water vapour per unit mass of moist air in a given volume. It has nearly the same values as mixing ratio.

- *Relative Humidity* – It is defined as the ratio of the actual partial pressure exerted by the water vapour in any volume of moist air to the partial pressure

that would be exerted by the water vapour if the air was saturated at the same temperature. That is:

$$\phi = \frac{P_s}{P_g} \quad 2.4$$

where P_s is the actual partial pressure of the water vapour and P_g the partial pressure of the water vapour when the air is saturated (saturation vapour pressure) at the same temperature. The relative humidity is often expressed as a percentage.

- *Degree of Saturation* - Although numerically close to Relative Humidity, it is not exactly the same. The degree of saturation is define as the ratio of the actual mass of water vapour in a given volume of dry air to that mass of vapour which the air would have if it were saturated at the same temperature. Like Relative Humidity, the Degree of Saturation is expressed in percentage:

$$\psi = \frac{\omega}{\omega_g} \times 100 \quad 2.5$$

where ω is the actual moisture content and ω_g is the moisture content at saturation for the same air temperature and total pressure.

Note: $\omega_g = 0.622 \frac{P_g}{P - P_g}$ and hence ψ can be expressed as, $\psi = \phi \frac{(P - P_g)}{P - P_s}$

- *Absolute Humidity* –The absolute humidity of the air at any given condition is the mass of water vapour per unit volume of air at that condition and, as such, is actually an expression of vapour density. Absolute humidity is usually expressed in kg/m^3 . The actual mass of water vapour per unit volume of air is solely a function of the dew point temperature of the air. Because of this fixed relationship between the dew point temperature and the absolute humidity of air, when the value of one is known the value of the other can be readily computed. The characteristic gas equation can be used in conjunction with property tables to determine the absolute humidity (vapour density) of any sample of air when either the dew point temperature or the vapour

pressure is known. For example, if the dew point temperature of a given volume of air is known the absolute humidity can be calculated from the characteristic gas equation:

$$\rho = \frac{P}{RT} \quad 2.6$$

where p is the vapour pressure corresponding to the saturation (dew point) temperature, T and R is the gas constant for water vapour.

The absolute humidity can be determined also directly from property tables simply by taking the reciprocal of the vapour specific volume (at the given dew point temperature) as listed in the table.

- *Dew Point Temperature* – It is the temperature to which unsaturated air must be cooled at constant pressure for it to become saturated (or for condensation to begin). Obviously, then, the dew point temperature of the air is always the saturation temperature corresponding to the partial pressure exerted by the water vapour. Consequently, when the partial pressure exerted by the water vapour is known, the dew point temperature, θ_d , of the air can be determined from property tables or from the expression

$$\theta_d = 6.98 + 14.428 \ln(P_s) + 1.05[\ln(P_s)]^2 \quad 2.7$$

where P_s is the partial pressure of the water vapour in kPa.

Likewise, when the dew point temperature of the air is known, the partial pressure exerted by the water vapour can be determined from the property table or from the expression

$$P_{gd} = P_s = \frac{\omega P}{0.622 + \omega} \quad 2.8$$

Note: at dew point temperature, $P_s = P_{gd}$ and $\omega = \omega_s$, where P_{gd} is the saturation vapour pressure that corresponds to the dew point temperature.

Once the dew point temperature of an unsaturated air has been determined, its Relative Humidity becomes:

$$\phi = \frac{P_{gd}}{P_g} \quad 2.9$$

This implies that P_g is the saturation pressure that corresponds to the higher (initial) temperature when cooling started, while P_{gd} is the saturation pressure that corresponds to the dew point (lower or final) temperature when cooling is completed.

- *Maximum Water Vapour Content* – The maximum amount of vapour that can be mixed with any given volume of dry air depends only on the temperature of the air. Since the amount of water vapour in the air determines the partial pressure exerted by the water vapour, it is evident that the air will contain the maximum amount of water vapour when the water vapour in the air exerts the maximum possible pressure. Since the maximum possible pressure that can be exerted by any vapour is the saturation pressure corresponding to its temperature, the air will contain the maximum amount of water vapour when the pressure exerted by the water vapour is equal to the saturation vapour pressure corresponding to the temperature of the air. At this condition the temperature of the air and the dew point temperature will be one and the same, and the air is said to be saturated. The higher the temperature of the air, the higher the maximum possible vapour pressure and the greater the possible water vapour content.

- *Wet-bulb Temperature* – The wet-bulb temperature is a type of temperature that reflects the physical properties of a system with a mixture of a gas and vapour, usually air and water vapour. Wet-bulb temperature can have several technical meanings:
 - Thermodynamic (or isobaric) wet bulb temperature, θ^* – the temperature a volume of air would have if cooled adiabatically to saturation at constant pressure by evaporation of water into it, all latent heat being supplied by the volume of air. It is the thermodynamic wet-bulb temperature that is normally plotted on a psychrometric chart. Mathematically, the thermodynamic wet-bulb temperature is expressed as,

$$\theta^* = \theta - \frac{(h_g^* - h_f^*)}{Cp_{as}}(\omega_s^* - \omega) \quad 2.10$$

where $Cp_{as} = Cp_a + \omega Cp_s$ is 'humid' specific heat (kJ/kg K); θ is the dry-bulb temperature (K); h_g^* is the specific enthalpy of moist air at saturated vapour state (kJ/kg); h_f^* is the specific enthalpy of moist air at saturated liquid state (kJ/kg); ω is the moisture content of the moist air at the dry-bulb temperature (kg/kg) and ω_s^* is the saturated moisture content of the air at wet-bulb temperature (kg/kg).

For the fixed pressure P , the thermodynamic wet-bulb temperature is a function of the dry-bulb temperature θ and the moisture content of the moist air ω and is uniquely determined for any given moist air state.

The value indicated by a simple wet-bulb thermometer often provides an adequate approximation of the thermodynamic wet-bulb temperature.

- The temperature read from a wet-bulb temperature device. One of the most convenient devices available is the psychrometer, which consists essentially of a dry-bulb and a wet-bulb thermometer.

The wet-bulb thermometer has its bulb wrapped in cloth—called a *sock*—that is kept wet with water via wicking action.

The equilibrium temperature assumed by a wet-bulb thermometer in a given airstream is the wet-bulb temperature θ' . Unlike θ^* , θ' is not a thermodynamic property.

Over the years various relations have been devised with which observed temperature readings can be used to determine the partial pressure of water vapour. The empirical equation most often used is:

$$P_s = P_g' - BP(\theta - \theta') \quad 2.11$$

where B is called the psychrometric constant.

For the commonly used 'sling' psychrometer, B is generally taken as $6.66 \times 10^{-4} K^{-1}$ or alternatively as: $B = 0.0006455 (1 + 1.876 g_s')$.

- Adiabatic wet-bulb temperature - the temperature a volume of air would have

if cooled adiabatically to saturation by reduction of pressure and then compressed adiabatically to the original pressure in a moist-adiabatic process. Such cooling may occur as air pressure reduces with altitude.

The thermodynamic wet-bulb temperature is the minimum temperature which may be achieved by purely evaporative cooling of a water-wetted, ventilated surface.

For a given parcel of air at a known pressure and dry-bulb temperature, the thermodynamic wet-bulb temperature corresponds to unique values of relative humidity, dew point temperature, and other properties. The relationships between these values are illustrated in a psychrometric chart.

For air that is less than saturated ($< 100\%$ relative humidity), the wet-bulb temperature is lower than the dry-bulb temperature; and the dew point temperature is less than the wet-bulb temperature.

The wet-bulb temperature, as measured by a wet-bulb thermometer, and the adiabatic saturation temperature are approximately equal for air-water vapor mixtures at atmospheric temperature and pressure. This is not necessarily true at temperatures and pressures that deviate significantly from ordinary atmospheric conditions.

The wet-bulb thermometer reports the thermodynamic wet-bulb temperature if:

- The sock is shielded from radiant heat exchange with its surroundings.
- Air flows past the sock quickly enough to prevent evaporated moisture from affecting evaporation from the sock.
- The water supplied to the sock is at the same temperature as the thermodynamic wet-bulb temperature of the air.

In usual practice, the value reported by a wet-bulb thermometer differs slightly from the thermodynamic wet-bulb temperature because:

- The sock is not shielded so well from radiant heat exchange.

- Air flow rate past the sock may be less than optimum.
- The temperature of the water supplied to the sock is not controlled.

At relative humidity below 100%, water evaporates from the bulb which cools the bulb below ambient temperature. To determine relative humidity, ambient temperature is measured using an ordinary thermometer, better known in this context as a dry-bulb thermometer. At any given ambient temperature, less relative humidity results in a greater difference between the dry-bulb and wet-bulb temperatures; the wet bulb is colder. The ‘wet-bulb depression’ is the difference between the dry-bulb temperature and the wet-bulb temperature.

The precise relative humidity is determined by finding wet-bulb and dry-bulb temperatures on a psychrometric chart (or via complex calculation).

The relationship between the thermodynamic and adiabatic processes is quite obscure. Comparisons indicate, however, that the two temperatures are rarely different by more than a few tenths of a degree Celsius, and the adiabatic version is always the smaller of the two for unsaturated air. Since the difference is so small, it is usually neglected in practice.

The wet-bulb temperature is used to determine the efficiency of the tower. The tower cools circulating water from a definite inlet to a definite exit temperature. The difference between the temperature of the water leaving the tower and the wet-bulb temperature of the incoming air indicates just how efficient the tower is. This measurement is called the approach of the tower (Silbert, 1999).

- *Enthalpy* – The enthalpy of a mixture of perfect gases is equal to the sum of the individual partial enthalpies of the components and a function only of temperature. With a datum of 0°C for dry air and water, the enthalpy of moist air (based on unit mass of dry air) may be expressed (in kJ/kg) as,

$$h = C_{p_a}\theta + \omega h_g \quad 2.12$$

where C_{p_a} is specific heat of dry air (kJ/kg K); $C_{p_a}\theta$ (or h_a) is

specific enthalpy of dry air (kJ/kg); h_g is specific enthalpy of saturated water vapour at the temperature of the mixture (kJ/kg) which can be approximated by the equation,

$$h_g = 2501 + 1.82\theta \quad 2.13$$

where $2501 = h_{fg} = h_g$ at $0^\circ C$ and $1.82 = Cp_s$ (specific heat of water vapour).

Taking $Cp_a = 1$ kJ/kg K, the moist air enthalpy becomes,

$$h = \theta + g(2501 + 1.82\theta) \quad 2.14$$

- *Specific Volume* – The specific volume of moist air is the volume per unit mass of dry air. Mathematically, this can be expressed (in m³/kg) as,

$$v = \frac{V}{m_a} = \frac{R_a T}{P_a} = \frac{\omega R_s T}{P_s} = \frac{(1 + \omega) R_{as} T}{P} \quad 2.15$$

where P_a is partial pressure for dry air (Pa); P_s is partial pressure of water vapour (Pa); V is volume of moist air (m³); T is absolute temperature (K); m_a is mass of dry air (kg); R_a is gas constant for dry air (J/kg K); R_s is gas constant for water vapour (J/kg K); R_{as} is gas constant for the mixture or moist air (J/kg K).

2.12.2 Saturation Vapour Pressure

The saturation vapour pressure, P_g , of an air-water vapour mixture is the maximum vapour pressure which can exist at any given temperature.

Under this condition, the water vapour in the mixture can coexist in neutral equilibrium with a surface of water at the same temperature and the moist air is said to be ‘saturated with vapour’.

Saturation vapour pressure is a function only of temperature and is independent of the dry air which simultaneously occupies the same space. The value of P_g given in property tables are supplemented by various expressions such as that given Osborne, Stimson and Ginning in their work. That is (McLean, 2009),

$$\log_{10} P_g = 30.59051 - 8.2 \log_{10} T + (2.480 \times 10^{-3})T - 3142.31/T \quad 2.16$$

where P_g is in kPa.

2.12.3 Cooling and Dehumidification

Simultaneous cooling and dehumidification of the air will take place any time the air is passed across a cooling surface whose temperature is below the initial dew point temperature of the air (Khurmi, 2004).

2.12.4 Psychrometric charts

These charts are graphical presentations of the psychrometric properties of air. The use of such charts permits the graphic analysis of psychrometric data and processes and thereby facilitates the solution of many practical problems dealing with air that otherwise would require more tedious mathematical solutions. A typical psychrometric chart is illustrated in Figure 2.6.

As previously stated, the thermodynamic state of moist air is uniquely fixed if the barometric pressure and two independent properties are known. A psychrometric chart may be constructed for some single value of barometric pressure.

The vertical lines on the chart are lines of constant dry bulb temperature, whereas the horizontal lines are lines of constant dew point temperature and humidity ratio (moisture content). The closely spaced diagonal lines are lines of constant wet bulb temperature and the more widely spaced diagonal lines are lines of constant specific volume.

The curved lines extending from lower left to upper right on the chart are lines of constant relative humidity. The curved line bounding the chart on the left side is the line of 100% relative humidity and is known as the saturation curve. Air at any condition such that its state can be plotted as a point falling anywhere along the saturation curve is saturated air. The other curve lines on the chart are enthalpy deviation lines.

The enthalpy deviation is the difference between the actual or true specific enthalpy of the air at any given condition and the specific enthalpy of saturated air at the same wet-bulb temperature.

Values for the dry-bulb and wet-bulb temperature lines are given at their intersection with the saturation curve. Values for dry-bulb temperature are given also at the base of the chart, as are the values for the specific volume lines.

Values for the dry-bulb temperature lines are read at the saturation curve and the

corresponding humidity ratio is read on the same line from the scale on the right side of the chart.

The specific enthalpy of saturated air at any given wet-bulb temperature is found by following the wet-bulb line to the enthalpy scale above the saturated curve. Enthalpy deviations are determined by interpolation between the lines of enthalpy deviation. The true enthalpy of air at any condition is determined by adding (algebraically) the enthalpy deviation to the specific enthalpy at saturation (McLean, 2009).

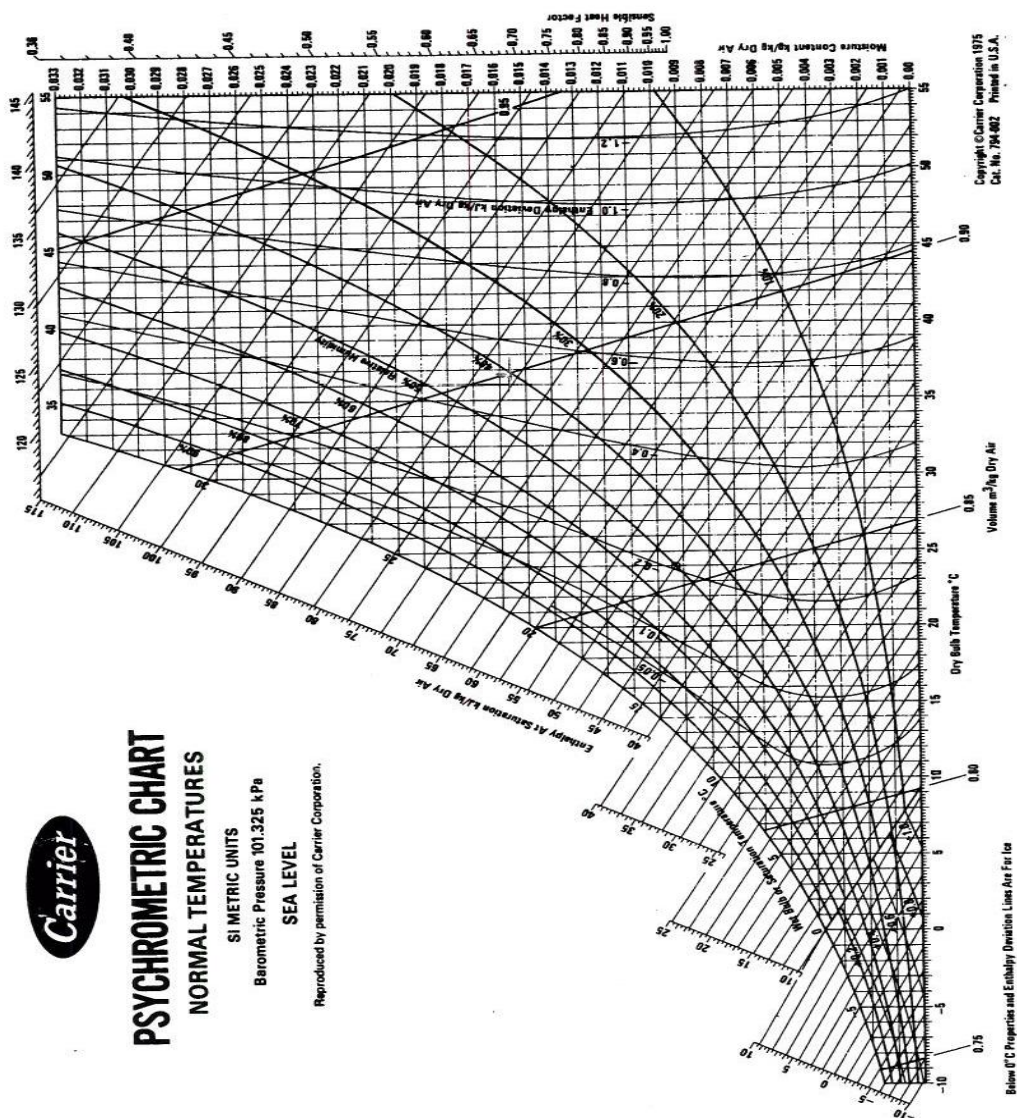


Figure 2.6: Psychrometric chart (Courtesy of Carrier)

2.13 DESCRIPTION OF THE COOLING-WATER CIRCUIT OF NLNG PLANT

Like crude oil, natural gas has to be transported from where it is produced to where it is needed for use, and sometimes both places are thousands of miles apart. One way of accomplishing this is to liquefy the natural gas from the reservoirs so that it can be loaded into ships and transported to the places where it is to be used. This is essentially the reason for setting up natural gas liquefaction plants.

The function of the cooling-water circuit is to provide cooling water to both process and equipment heat exchange within the Liquefied Natural Gas (LNG) plant liquefaction process train. Each LNG train is equipped with an independent open re-circulation water-cooling system. The cold-water temperature is 32°C but can vary between 30°C (cold cooling water temperature) and 33.5°C (hot cooling water temperature).

These temperatures are based on the corresponding wet bulb temperature of 28°C, 25°C and 30°C respectively. The cooling tower design is based on the most stringent of these operating points (that is, cold water temperature of 30°C and a wet bulb temperature of 28°C).

Cooling is achieved by pumping treated fresh water to process exchangers and other equipment, then back to the induced draught film type counter-flow cooling towers. In the Cooling Towers the air and water streams come into intimate contact as the water falls by gravity into the three cooling tower basins that feed the cooling water pumps (Figure 2.3).

The system is divided into the following parts:

Water Circulation, Distribution and Return.

Two out of three circulating water pumps are in operation at any given time (with one pump on stand-by). These pumps take cold water from the cooling tower basin and deliver it to an underground header for distribution around the gas liquefaction process train. The total flow rate and flow rates to individual exchangers are set by the following:-

Pump performance curve - fixed.

System hydraulic resistance - fixed, excluding balancing valves and fouling.

Fouling - variable, slowly increasing with time.

Position of balancing valves - fixed during commissioning.

The collection pipe-work and headers combine the water streams exiting the various exchangers and directs them to the induced draught cooling cell bank of the cooling tower, where the water is cooled by direct contact with a counter current air stream.

The flow through individual exchangers is fixed by the balancing valve on each exchanger cooling water outlet. The system high point is the inlet to the cooling tower cell bank, which ensures that the system does not partially empty on shut down. This is also the location of an 0.46 m (18 inch) atmospheric vent (elevated 25m above sea level), which prevents over pressurization of the system in the event of an exchanger tube failure and also provides a safe point for the venting of the resultant vapours. The distribution system is effectively open to atmosphere at all times.

Cooling Towers.

Each process Train is equipped with a bank of 9 induced draught cooling water cells which reduce the water temperature by evaporation and convection. This occurs as the upward airflow comes into intimate contact with the film of downward flowing cooling water passing through the corrugated PVC packing.

The water to be cooled is fed into the top of the cooling tower cells via separate connections to the cooling water return header. Each of the connections has a balancing valve. The balancing valves are operated from the cooling tower fan deck. Within the cells the water is distributed to the spray nozzles that spray the water evenly onto the packing. The water falls through the packing counter current to a stream of air which is drawn upwards by the induced draft fans (one per cell) through the air inlets situated at the base of the tower, through the tower's louvers and then through the fill (packing) system. The cooled water is collected in a basin at the bottom of the tower.

The cooling tower fans can be operated at 2 speeds to give some regulation of the cooling water temperature as ambient conditions change. However, the amount of cooling capacity can be a limiting factor for LNG production. Hence the fans are run at high speed most of the time.

The cooling tower basin level is controlled by adding make-up water from a Water Treatment Unit under automatic control using a level controller. The make-up water source is used to replenish water lost to evaporation. This make-up water contains approximately 1 ppm of free residual chlorine to inhibit biological growth.

Figure 2.7 is a picture of NLNG plant cooling tower while, Figure 2.8 is a picture showing the internal parts of the tower.



Figure 2.7: Picture of NLNG Plant Cooling Tower (9 cells)

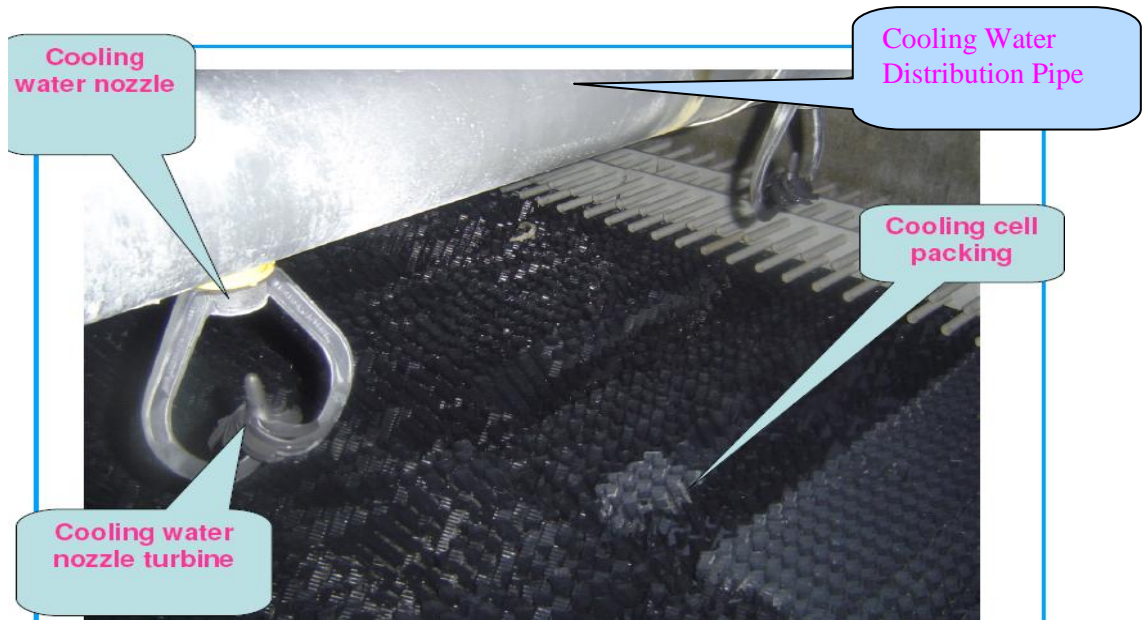


Figure 2.8: Picture of Internals of NLNG cooling Tower

2.14 SUMMARY

In this chapter, the following were presented:

- A review of the general principles of cooling towers and liquid dehydration systems.
- The factors influencing cooling tower performance.
- A review of previous studies that has been carried out by others to improve the thermal performance of cooling towers.
- The new concepts / modifications proposed by this research study to further improve the thermal performance of cooling towers and, consequently, the overall energy efficiency and production of water-cooled plants serviced by cooling towers.
- A review of the science of humid air (psychrometry) and air conditioning.
- A detail description of the cooling water circuit of the Nigeria Liquefied Natural Gas (NLNG) plant, which is used as a case study for this research work.

The next chapter presents the analytical techniques / theories that were used in this study.

CHAPTER 3

ANALYTICAL THEORIES

3.1 INTRODUCTION

In this chapter, the theories used for the analysis, modelling, design (specifications and sizing) and evaluation of the proposed air conditioning system (circuit) of the cooling tower are presented. In addition, the data collection methodology used for the study is also presented.

3.2 COOLING TOWER ANALYTICAL THEORY

Whenever a liquid is converted into vapour by evaporation as a result of exposure to a volume of gas, heat is absorbed from the liquid or any object in contact with it. This heat is the latent heat used in causing the transformation from liquid to vapour. The absorption of this latent heat from the liquid brings about a fall in its temperature. The faster the evaporation, the greater is the fall in temperature.

As stated in Chapter 2, in a cooling tower, the transfer of sensible and latent heat (evaporation) transfers heat from water droplets to the surrounding air. The transfer of heat is effected from the water through the boundary film of saturated air in contact with the water surface. This air is saturated at the water temperature. From this saturated air film, heat transfer takes place to the general mass of air flowing through the tower. Vaporisation, and hence cooling, can take place as long as the partial pressure of the water vapour at the air-water interface (boundary film) exceeds the partial pressure of the water vapour in the bulk air stream. When the vapour pressure of the air stream equals that of the air-water interface, the air is saturated and vaporisation ceases

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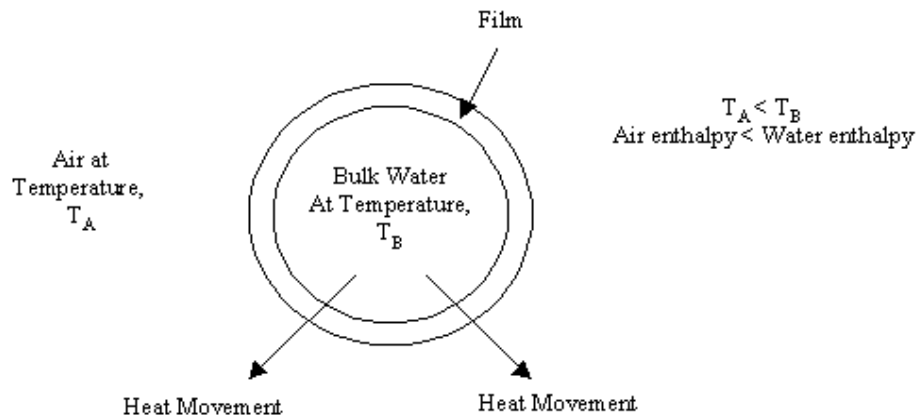


Figure 3.1: Schematic showing heat movement around a water droplet in a cooling tower (Courtesy www.cheresources.com)

This movement of heat can be modelled with a relation known as the Merkel Equation, which is (BS 4485, 1988):

$$\frac{KaV}{LC_w} = \int_{T_2}^{T_1} \frac{dT}{h_s - h_a} \quad 3.1$$

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where:

KaV/L = tower characteristic

K = mass transfer coefficient [$kg/(m^2 \cdot s \cdot (kg/kg))$]

a = contact area/tower volume (m^2/m^3)
(area of effective transfer surface per unit of tower packing volume)

V = active cooling volume/plan area (m^3/m^2)
(effective packing volume per unit area of packing)

L = water rate [$kg/(m^2 \cdot s)$]
(mass water flow per unit plan area of packing)

C_w = specific heat capacity of water (kJ/kg K)

T_1 = hot water temperature ($^{\circ}C$)
 T_2 = cold water temperature ($^{\circ}C$)

T = bulk water temperature ($^{\circ}\text{C}$)

h_s = enthalpy of air-water vapour mixture (saturated boundary air film) at bulk water temperature (kJ/kg dry air)

h_a = enthalpy of air-water vapour mixture (bulk air stream) at wet bulb temperature (kJ/kg dry air).

An energy balance also dictates that the heat removed from the water

must be equal to the heat absorbed by the surrounding air (Cheresources.com, 2003):
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$$L(T_1 - T_2)C_w = G(h_2 - h_1) \quad 3.2$$

where:

C_w = specific heat capacity of water, kJ/(kg.K)

h_2 = enthalpy of air-water vapour mixture (bulk air) at exhaust wet-bulb temperature (kJ/kg)

h_1 = enthalpy of air-water vapour mixture (bulk air) at inlet wet-bulb temperature (kJ/kg)

G = Air rate [kg/($\text{m}^2 \cdot \text{s}$)]

(mass air flow per unit plan area of packing){PRIVATE "TYPE=PICT;ALT=ctowers6.gif (3725 bytes)"}

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The tower characteristic value can be calculated by solving the Merkel Equation with the Tchebycheff numerical method¹⁴ (CTI, 1975).

Note: The Tchebycheff method for numerically evaluating the integral $\int_a^b y dx$ uses values of y at predetermined values of x within the interval “ a ” to “ b ”, so selected that the sum of these values of y multiplied by a constant times the interval ($b-a$) gives the desired values of the integral. In its four-point form, the approximation formula becomes;

$$\int_a^b y dx = \frac{b-a}{4} (y_1 + y_2 + y_3 + y_4) \quad 3.3$$

where:

y_1 are values of y at $x = a + 0.1 (b-a)$

y_2 are values of y at $x = a + 0.4 (b-a)$

y_3 are values of y at $x = b - 0.4 (b-a)$

y_4 are values of y at $x = b - 0.1 (b-a)$

Hence:

$$\frac{KaV}{L} = \int_{T_2}^{T_1} \frac{dT}{h_s - h_a} = \frac{T_1 - T_2}{4} \left(\frac{1}{\Delta h_1} + \frac{1}{\Delta h_2} + \frac{1}{\Delta h_3} + \frac{1}{h_4} \right)$$

where:

Δh_1 = value of $h_s - h_a$ at $T_2 + 0.1(T_1 - T_2)$

Δh_2 = value of $h_s - h_a$ at $T_2 + 0.4(T_1 - T_2)$

Δh_3 = value of $h_s - h_a$ at $T_1 - 0.4(T_1 - T_2)$

Δh_4 = value of $h_s - h_a$ at $T_1 - 0.1(T_1 - T_2)$

Details of the derivation of the Merkel cooling tower equation is given in Appendix 8

Figure 3.2 is a cooling tower enthalpy-temperature diagram showing the graphical integration of the $dT/(h_w - h_a)$. An understanding of this diagram is important in visualising the cooling tower process (Cheresources.com, 2003).

¹⁴ Other methods of suitable accuracy are (1) Numerical integration using Simpson's rule and (2) Steven's diagram (using the Carey and Williamson chart).

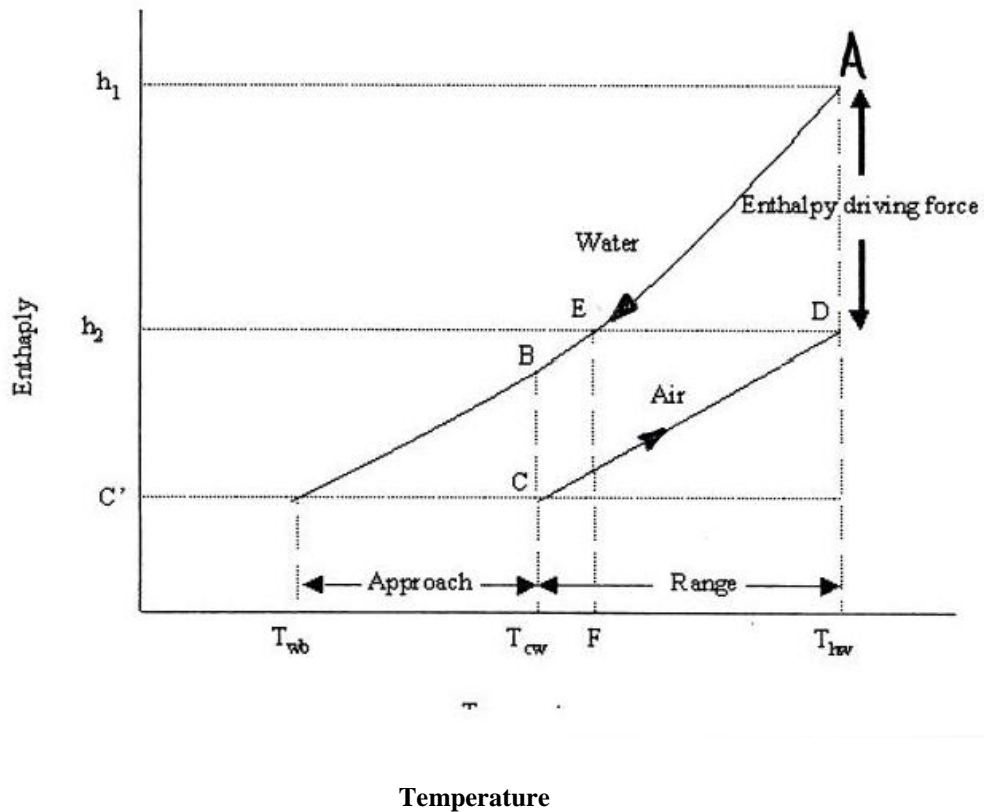


Figure 3.2: Graphical Representation of the Cooling Tower Characteristics
(Courtesy www.cheresources.com)

The following represents a key to Figure 3.2:

C' = Entering air enthalpy at wet-bulb temperature, T_{wb}

BC = Initial enthalpy driving force

CD = Air operating line with slope L/G

DEF = Projecting the exiting air point onto the water operating line and then onto the temperature axis shows the outlet air wet-bulb temperature.

Note: The water flows downwards from A to B, while the air flows upwards from C to D.

As stated in Section 2.5, present cooling tower technology relies on the fact that, with acceptable error, the effects of evaporative and sensible heat transfer can be combined into one dependent on enthalpy difference. The difference concerned is that between the enthalpy of the film of air surrounding the water surface (taken to be at water temperature) and enthalpy of the general mass of air flowing through the tower. This enthalpy difference varies according to the point of measurement in the tower, but at all points it provides the enthalpy potential or driving force for the heat transfer. In Figure 3.2, the water operating line is AB and is fixed by the inlet and outlet water temperatures. The air operating line begins at C, vertically below B and at a point having an enthalpy corresponding to that of the entry wet bulb temperature. Line BC represents the initial driving potential, $h_w - h_a$, for transfer of heat from the water to the air. The water-to-air ratio, L/G , is the slope of the air operating line (Equation 3.2). The air leaving the tower is represented by point D. The cooling range is the projected length of line CD on the temperature scale. The cooling tower approach is shown on the diagram as the difference between the cold water temperature leaving the tower and the ambient wet-bulb temperature.

The coordinates refer directly to the temperature and enthalpy of any point on the water operating line but refer directly only to the enthalpy of a point on the air operating line. The corresponding wet-bulb temperature of any point on the CD line is found by projecting the point horizontally to the saturated (water) curve, then vertically to the temperature coordinate. The area ABCD in the diagram represents the integral of the Merkel equation. This value gives the number of diffusion units¹⁵ (also called the number of transfer units – NTU) and is known as the tower characteristic, KaV/L , varying with the water-to-air ratio, L/G . The smaller the area, the smaller the total potential for heat transfer and, hence, the less the diffusion units. If the water loading on the tower is increased without increasing the air quantity it will result in an increase of the slope of the air operating line, L/G and hence, a decrease in the tower characteristic (potential) due to the reduction of area ABCD. This is the simple observation that the more water circulated per kg of air the less the extent of cooling.

In order to predict tower performance, it is necessary to know the required tower characteristic for fixed ambient and water conditions.

As shown by the Merkel Equation, by finding the area between ABCD in Figure 3.2,

¹⁵ Diffusion of water vapour from saturated boundary film to the bulk air stream.

the tower characteristic can be found.

When carrying out a performance test on a cooling tower, the test value of L/G is calculated from the average water flowrate and fan horsepower during the test using the following equation (CTI, 1975):

$$\left(\frac{L}{G}\right)_t = \left(\frac{\text{Test water flowrate}}{\text{Design water flowrate}}\right) * \left(\frac{\text{Design fan horsepower}}{\text{Test fan horsepower}}\right)^{\frac{1}{3}} * \left(\frac{L}{G}\right)_d \quad 3.4$$

where subscript “t” denotes test value and subscript “d” denotes design value.

3.3 COOLING TOWER HEAT TRANSFER (ENERGY) EQUATIONS

The basic heat transfer equation (balance) for a cooling tower is (BS 4485, 1988):

Heat transferred = heat loss by water = heat gained by air

$$Q = LC_w dT = Gdh = K (h_s - h_a) dV \quad 3.5$$

where:

$$Gdh = G (h_2 - h_1) = G (C_a dt + \lambda dw)$$

and,

$$h_2 = h_1 + C_w dT (L/G)$$

Equation 3.5 gives the heat transfer capacity of a cooling tower per unit area of packing and per unit time.

Note: The air loading G remains constant throughout the tower. The water loading L is not quite constant owing to the evaporation of water into the air (the water evaporated is $dL = Gdw$). However, the saturation loss from the water to air amounts to less than 2% of the water circulated over the tower; hence L may be considered constant without introducing serious error.

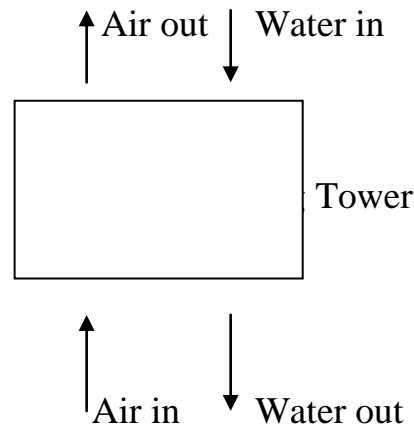


Figure 3.3(a): Cooling tower schematic

Alternatively, the heat balance can be written as:

$$Q = LC_w dT = GC_a dt + G \lambda dw \quad 3.6$$

where:

$GC_a dt$ = Sensible heat gained by air flowing out of tower

$G \lambda dw$ = latent heat gained by air flowing out of tower

The right-hand side (RHS) of Equation 3.6 gives the heat that is transferred to the air from the water in the tower (by evaporation and sensible heat transfer). For a heat balance, this heat should equal the heat energy of the water flowing into the tower. However, there may be an imbalance, where the right-hand side is less than the left-hand side ($\text{RHS} < \text{LHS}$) of the equation (e.g., due to heat loss through the tower walls). The difference gives an indication of the performance of the cooling tower in terms of energy balance.

Equation 3.5 is the key equation for the design and analysis of performance in cooling towers. ‘ K ’ is the overall transfer-rate term analogous to overall heat transfer coefficient, U , in tubular exchangers. However, in tubular exchangers the heat-transfer surface is usually known or can be readily computed. In cooling towers, the value of ‘ a ’ cannot be determined directly; since it is composed of a random disposition of droplet and film surface. (The film surface is nearly independent of the thickness of the film; while the droplet surface is dependent upon both the portion of

the liquid forming the droplet and the average size of the droplet formed). The inability to calculate 'a' is overcome by experimentally determining the product 'Ka' as a whole for a particular type of tower fill and at specific rates of flow of water and air.

Rearranging Equation 3.5 gives:

$$\frac{KaV}{Lc} = \int_{T_2}^{T_1} \frac{dT}{h_s - h_a} \quad 3.7$$

where KaV/L is the tower characteristic. In cooling tower practice, the integrated value of Equation 3.7 is commonly referred to as the number of transfer units (NTU) or number of diffusion units. This value gives the number of times the average enthalpy potential $(h_s - h_a)_{avg}$ goes into the temperature change of the water $(T_1 - T_2)$ and is a measure of the difficulty of the task. Thus, one transfer unit has the definition of $c_p \Delta T / (h_s - h_a)_{avg} = 1$ (ASHRAE, 2008). If a quantity of water of a given inlet temperature and a quantity of air of a given inlet wet-bulb temperature come in contact in the cooling tower, it will require a certain number of transfer units as determined by the integration of Equation 3.7 to reduce the water to any desired temperature. Hence, the number of transfer units is determined only by the process conditions imposed upon the tower. Integration of Equation 3.7 gives the NTU for a given set of conditions (ASHRAE, 2008).

The term cooling efficiency (tower effectiveness) is used to define the ratio of actual cooling in the tower to the maximum possible (or ideal) cooling (BEE, 2004).

$$\eta = \frac{T_1 - T_2}{T_1 - T_{wb}} \quad 3.8$$

In other words, the cooling tower effectiveness is: *Range / Range + Approach*.

The heat load on the cooling tower is given by (BEE, 2004):

$$Q = \dot{m}_w C_w (T_1 - T_2) \quad 3.9$$

It is a measure of the tower loading. Most often, cooling towers are designed to dissipate a maximum of 110 to 120% of the design process heat load in order to fulfil its design conditions, e.g., cold water temperature, range and approach (BS 4485, 1988).

3.3.1 Energy Analysis

To make an energy analysis of a cooling tower, a control volume is drawn around the entire tower as shown in the Figure 3.3(b). The process is assumed to be adiabatic, the fan work is neglected, and changes in kinetic energy and potential energy are negligible (Mc Conkey, 1978).

Hence, the basic steady state energy equation

$$Q + W = \sum_{out} \left(h + \frac{V^2}{2} + gz \right) m - \sum_{in} \left(h + \frac{V^2}{2} + gz \right) m$$

reduces to

$$\sum_{out} (mh) - \sum_{in} (mh) = 0 \quad 3.10$$

That is,

$$m_{a1}(h_{a1} + w_1 h_{v1}) + m_{w3} h_{w3} = m_{a2}(h_{a2} + w_2 h_{v2}) + m_{w4} h_{w4} \quad 3.11$$

where:

m_{a1} = mass flow rate of inlet atmospheric air at point 1

m_{a2} = mass flow rate of saturated outlet air at point 2

m_{w3} = mass flow rate of inlet water at point 3

m_{w4} = mass flow rate of outlet water at point 4

h_{a1} = enthalpy per unit mass of inlet atmospheric air at point 1

h_{a2} = enthalpy per unit mass of saturated outlet air at point 2

h_{w3} = enthalpy per unit mass of inlet water at point 3

h_{w4} = enthalpy per unit mass of outlet water at point 4

h_{v1} = enthalpy per unit mass of water vapour at point 1

h_{v2} = enthalpy per unit mass of water vapour at point 2

w = humidity ratio (moisture content) of air

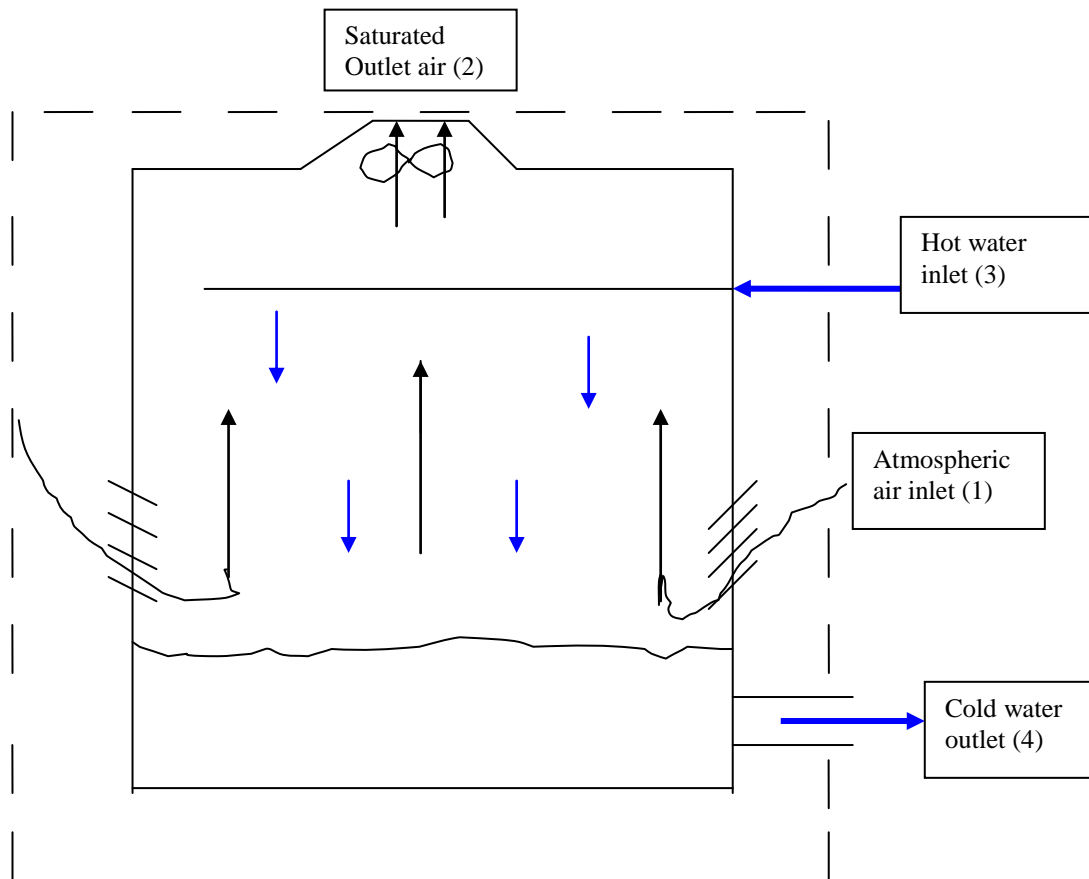


Figure 3.3(b): Control volume of cooling tower

For atmospheric air, the enthalpy of the water vapour may be approximated by h_g for a saturated vapour at the given air temperature; and for the water stream, enthalpy may be evaluated by the enthalpy of saturated liquid water, h_f , at the given temperature.

Also,

$$h_a = C_{pa}T \quad \text{and} \quad m_{a1} = m_{a2} = m_a$$

Hence, Equation 3.11 becomes

$$m_a \{C_{pa} (T_1 - T_2) + w_1 h_{g1} - w_2 h_{g2}\} = m_{w4} h_{f4} - m_{w3} h_{f3} \quad 3.12$$

In many uses of cooling towers the value of m_{w3} is known, as well as the temperature drop needed for the water stream. Hence it is convenient to replace m_{w4} in terms of m_{w3} . This can be done by making a mass balance on the water passing through the control volume. That is,

$$m_{w4} = m_{w3} - m_a (w_2 - w_1) \quad 3.13$$

Substituting Equation 3.13 into Equation 3.12 yields

$$m_a [C_{pa}(T_1 - T_2) + w_1 h_{g1} - w_2 h_{g2} + (w_2 - w_1)h_{f4}] = m_{w3} (h_{f4} - h_{f3}) \quad 3.14$$

Equation 3.14 can be used to calculate the mass flowrate of air required.

3.3.2 Determination of Tower Capability

The manufacturer's tower characteristic curve, which is a plot of tower characteristic (KaV/L) versus water to air flow ratio (L/G), is used for the determination of the capability of a cooling tower. The plot should fit an equation of the form (CTI, 1975):

$$KaV/L = c*(L/G)^n \quad 3.15$$

where c and n are mass transfer constants for a particular design of packing.

On the same plot, a curve representing design approach, cooling range and wet bulb temperature is shown and the intersection of the two curves indicate the design value of L/G for the cooling tower. The plot covers a range of L/G values from 20% above and 20% below design value (CTI Code, 1975).

To determine the tower performance, the point representing values of L/G and KaV/L , calculated from test data, is located on the manufacturer's plot. Through this

point a curve is drawn parallel to the tower characteristic curve. The intersection of the line so drawn with the design approach curve determines the value of L/G at which the tower will produce a design approach at design range and wet-bulb temperature.

The ratio of this value of L/G to the design value of L/G multiplied by 100 represents the percentage of the design circulating water flow which can be cooled at design range, approach and inlet air conditions. That is,

$$\% \text{ Capability} = \frac{(L/G)}{(L/G)_d} * 100 \quad 3.16$$

3.3.3 Evaluation of Thermal Performance

The measure of the thermal performance of a cooling tower is its ability to fulfil guaranteed conditions in terms of fan power, water flow, range, approach and inlet air conditions. This can be achieved by a direct comparison between the test results and the manufacturer's performance curves or by reference of the test results to design conditions (BS 4485, 1988).

The effect of measurement errors are usually taken into account. The reproducibility of the evaluation is $\pm 5\%$. Therefore, the tower is deemed acceptable if the evaluated result from the test equals or exceeds 95% of the design capability (BS 4485, 1988).

Figure 3.4 is an example of a cooling tower characteristic curve. This is the actual curve obtained from the performance test for the NLNG cooling tower. The design characteristic and approach for this cooling tower are 1.69 and 8.1 °C, respectively. This curve was used to determine the tower capability as explained in Sub-section 3.3.2 of this chapter.

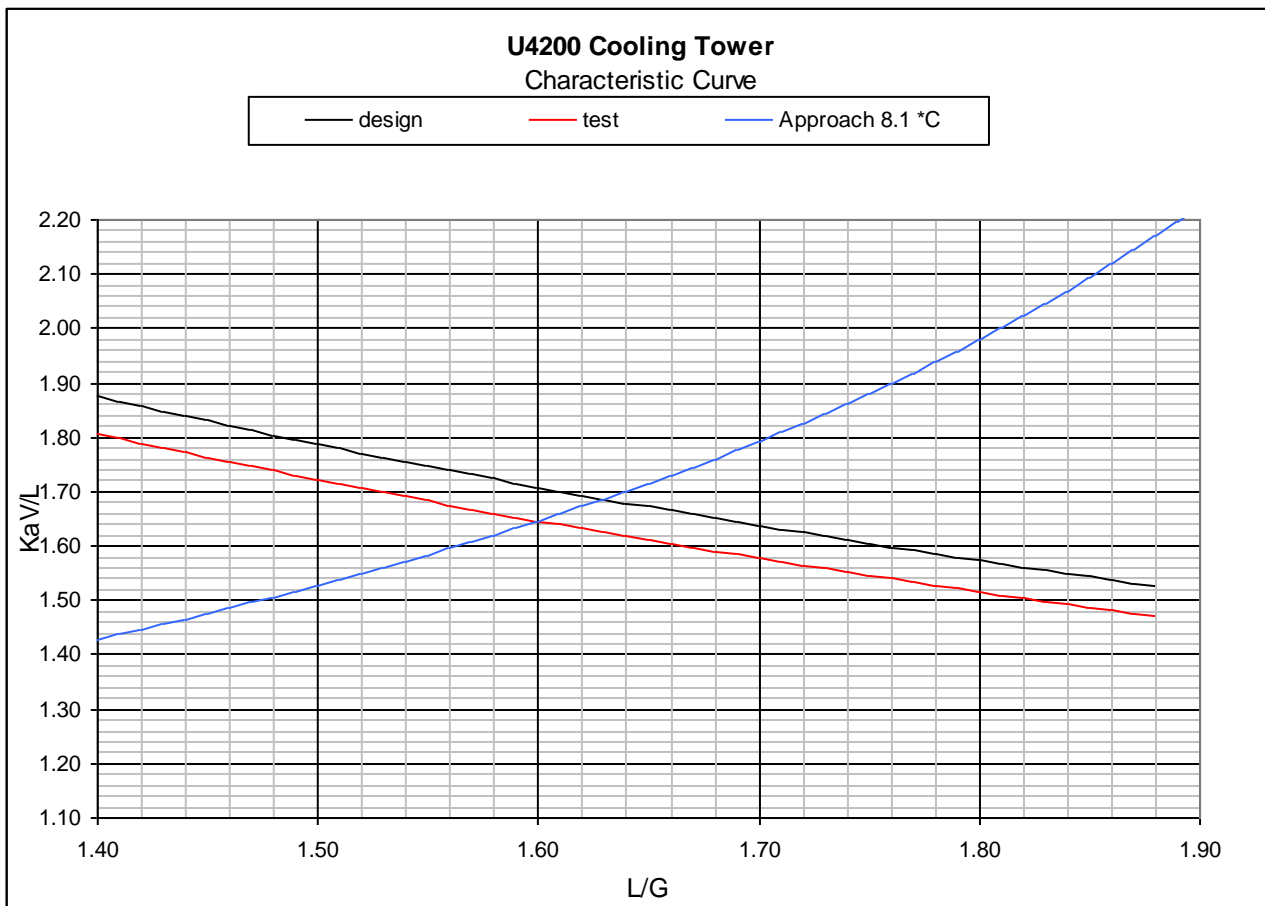


Figure 3.4: Cooling Tower Characteristic Curve
(Courtesy NLNG Operating Manual)

3.4 BASIC PROCESS DESIGN FACTORS AND ANALYSIS FOR GLYCOL DEHYDRATION

All factors controlling the behaviour of absorption systems also apply for glycol (TEG) dehydration.

In order to properly design the unit there is a need to know the maximum air flow rate, maximum temperature and pressure and the required dew point or moisture content of the outlet air. From these, the following can be calculated:

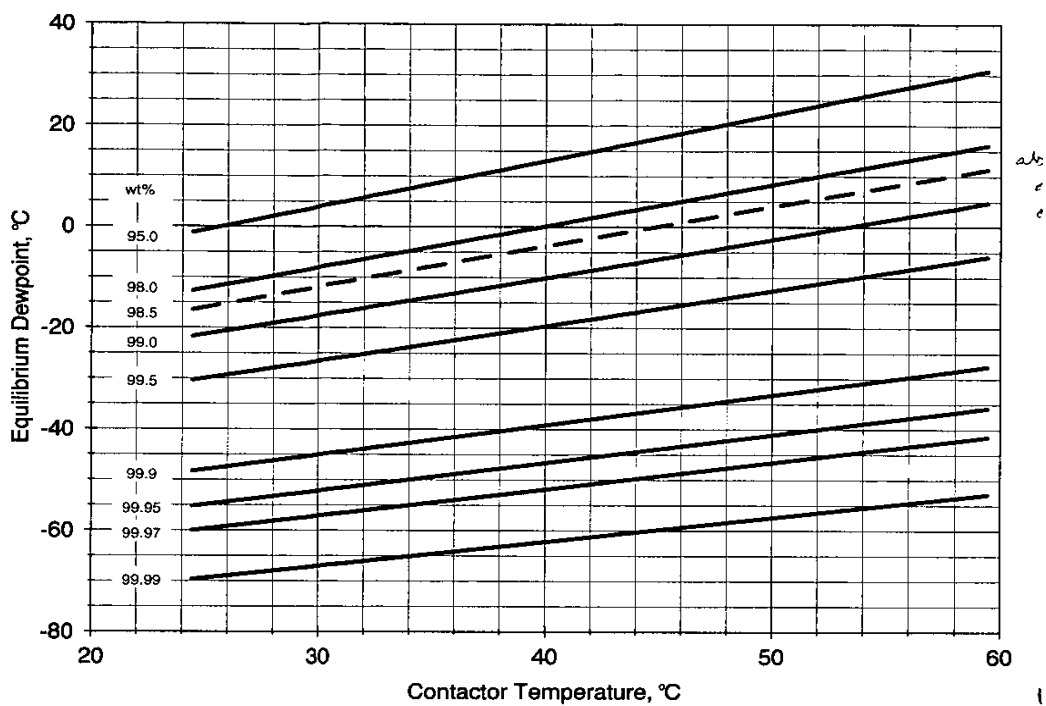
1. The minimum concentration of TEG (absorbent) in the lean solution entering the absorber required to meet outlet air specification (moisture content).
2. The lean TEG circulation rate required to meet the outlet air moisture content specification (by absorption of water from the air).
3. The amount of absorber contact required to produce the necessary approach to equilibrium required in (1) above at the chosen circulation rate.

To obtain these answers it is necessary to have a vapour-liquid equilibrium correlation for a TEG-water system. From this input, the sizing of equipment and the development of mechanical specifications can be performed (Campbell, 2004).

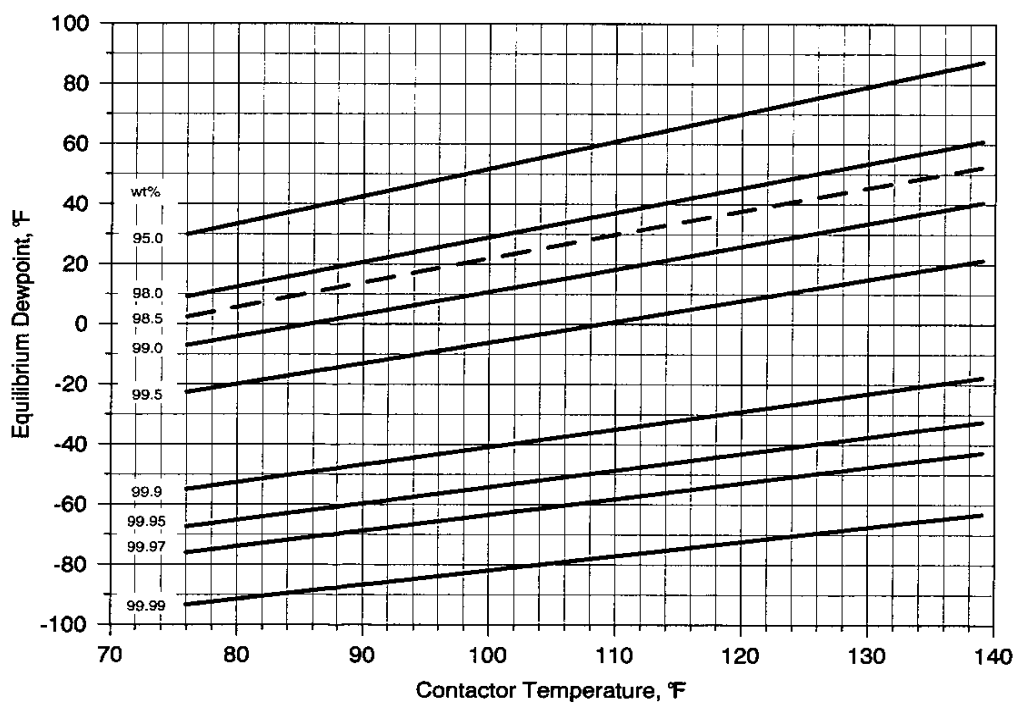
3.4.1 Minimum Lean TEG Concentration

If water-saturated gas (e.g., air) is placed in a static cell with a given concentration of TEG-water solution at a fixed pressure and temperature, equilibrium would be attained in time. Assuming the liquid (TEG-water solution) had a sufficiently lower water concentration, water would transfer to this liquid from the gas. At equilibrium, the mol fraction of water in the gas divided by its mol fraction in the liquid equals the equilibrium constant (K-value) for this system (Campbell, 2004).

Figures 3.5(a) and 3.5(b) are based on equilibrium data published by Parrish, et. al. (1986). The Parrish equilibrium correlation has been shown because equilibrium water concentrations in the vapour phase were determined at infinite dilution (essentially 100% TEG). The effect of pressure on TEG-water equilibrium is small up to about 13800kPa (Campbell, 2004).



(a)



(b)

Figure 3.5: Equilibrium H₂O Dew point vs. Temperature at Various TEG Concentrations
(Courtesy J. Campbell)

A TEG absorber is essentially isothermal. The heat of solution (energy released by TEG per kg of water absorbed) is about 21kJ/kg of water absorbed in addition to the latent heat. However, the mass of water absorbed plus the mass of TEG circulated is trivial compared to the mass of air. The temperature rise due to the heat of absorption seldom exceeds 1-2 K, except when dehydrating at pressures below about 1000kPa. In low pressure service, some temperature adjustment may be desirable (Campbell, 2004).

The diagonal lines in Figure 3.5 represent weight % TEG in a TEG-water mixture entering the top of the absorber. Equilibrium dew point could be attained in a test cell but not in a real absorber because the gas and TEG are not in contact for a long enough time to reach equilibrium. Numerous tests show that a well designed, properly operated unit will have an actual water dew point 5.5 to 8.5 K higher than the equilibrium dew point (Campbell, 2004). This approach to equilibrium can be used to specify the minimum lean glycol concentration. The procedure is as follows:

- 1) Establish the desired outlet air water dew point needed (from specification or minimum system temperature). The dew point refers to the temperature at which the moisture content in the air will saturate the air (at absorber or contactor temperature).
- 2) Subtract the “approach” from step (1) to find the corresponding equilibrium water dew point.

Note: Equilibrium dew point + Approach = Actual dew point

- 3) Enter the value in step (2) to the ordinate of Figure 3.5 and draw a horizontal line.
- 4) Draw a vertical line from the inlet gas temperature on the abscissa.
- 5) The intersection of the lines in steps (3) and (4) establishes minimum lean TEG concentration required to obtain the water dew point in step (1).

Note: If water content is specified or calculated in mass per unit gas volume, a water content, pressure, dew point temperature correlation is required. Also, the

equilibrium water dew points on the ordinate of Figure 3.5 are based on the assumption that the condensed water phase is a metastable¹⁶ liquid.

The dashed line in Figure 3.5 at about 98.5 weight percent (wt%) represents the concentration of lean TEG that can be produced routinely in a regenerator operating at standard atmospheric pressure and 204°C. This is a safe value for design and specification purposes (Campbell, 2004).

It is necessary to fix a lean TEG concentration for subsequent calculations. For the first consideration, the results obtained from Figure 3.5 will be used. If, however, the concentration obtained is less than 98.5 wt%, then 98.5 wt% will be used for calculation purposes (as 98.5 wt% is the minimum recommended for design purpose).

The minimum lean TEG concentration may not be the one used. A higher concentration than this may be specified to minimize circulation rate and optimize cost. It is usually a trade-off between TEG concentration and circulation rate (Campbell, 2004).

3.4.2 TEG Regeneration

The required lean TEG concentration is produced in the regenerator. The regenerator consists of a reboiler and a still column. A given lean TEG concentration is produced in the regenerator by control (adjustment) of the regenerator (reboiler) temperature, pressure and possible use of a stripping gas¹⁷.

The concentration of the rich TEG leaving the absorber is found by a water material balance around that absorber. By definition,

$$wt\% RichTEG = \frac{mass\ leanTEG}{mass\ leanTEG + mass\ water\ absorbed + mass\ water\ in\ leanTEG} \times 100 \quad 3.17$$

¹⁶ Metastability is a general scientific concept which describes states of delicate equilibrium. A system is in a **metastable state** when it is in equilibrium (not changing with time) but is susceptible to fall into lower-energy states with only slight interaction.

¹⁷ An inert gas used to enhance the removal of water from the glycol in the regenerator by reducing the partial pressure of the water.

The mass quantities in this equation may be found per unit time or per unit of gas (air) or glycol flow. In any case, the value depends on circulation rate. The actual TEG rate used depends on dew point requirements, lean TEG concentration, number of absorber contacts and economics. This rate usually will be 16-40 liters of lean TEG solution per kg of water absorbed from the gas. The minimum rate is governed by the rate required for effective air-liquid contact in the absorber; the maximum is limited by economics (Campbell, 2004).

Because regeneration takes place at a low pressure (near atmospheric pressure), under essentially ideal gas conditions, the calculation is very routine. Figure 3.6 is used to predict regenerator performance.

The minimum wt% lean TEG on the top of the abscissa (in Figure 3.6) is found from Figure 3.5. The wt% of rich TEG on the bottom of the abscissa is found from Equation 3.17. The rich TEG concentration may be calculated from,

$$Rich\ TEG = \frac{(\rho)(Lean\ TEG)}{\rho + (1/CR)} \quad 3.18$$

where:

ρ = lean TEG density (1.12kg/liter)

CR = circulation ratio (liters TEG/kg H₂O)

Rich TEG = wt% TEG in rich TEG solution

Lean TEG = wt% TEG in lean TEG solution

The diagonal lines in the lower left portion of Figure 3.6 represent various amounts of stripping gas for units where the stripping gas is sparged into the reboiler (it is seldom that more than 0.08m³ stripping gas per liter of TEG is used).

Three temperature lines are shown. Where high concentrations are desired, the specification of 204°C is normal unless the gas being dehydrated contains oxygen. This is close to the thermal decomposition temperature of TEG (in air).

The diagonal lines at the upper right in Figure 3.6 represent the effect of regeneration pressure in mmHg,

$$760\ \text{mmHg} = 14.7\ \text{psia} = 101.325\ \text{kPa}$$

$$100\ \text{kPa} = 1\ \text{bar} = 750\ \text{mmHg}$$

Unless a vacuum is being used, it is customary to use the 760 mm Hg line for design calculations (Campbell, 2004).

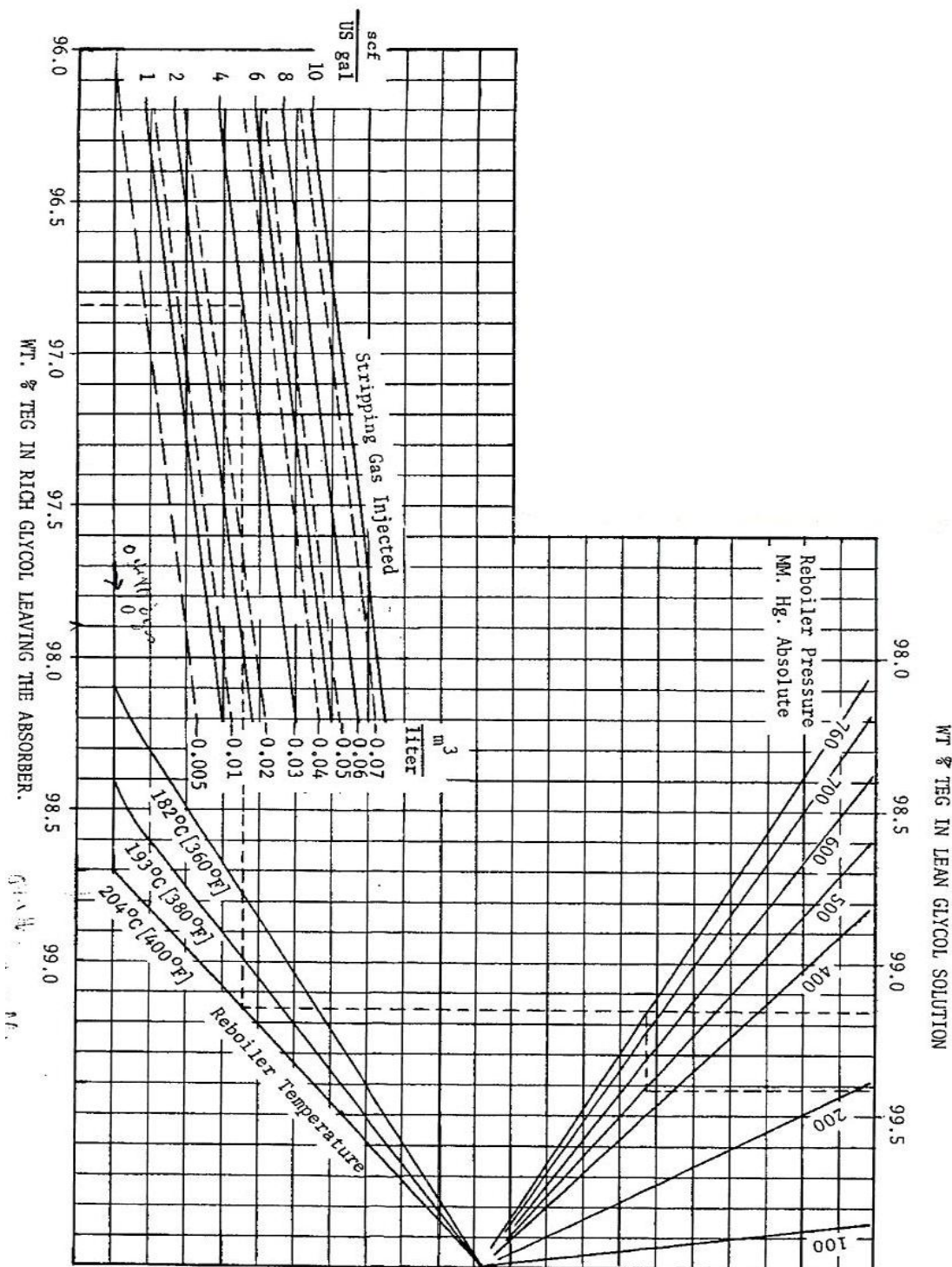


Figure 3.6: Nomograph for Estimating Regenerator Performance as a Function of Pressure, Temperature and Stripping Gas (Courtesy J. Campbell)

The general procedure for using Figure 3.6 is as follows (Campbell, 2004):

1. Atmospheric Pressure, No Stripping Gas – wt% rich glycol is not a variable. Proceed vertically from 0 stripping gas and temperature line intersection. This will read 98.7 wt% TEG at 204°C; 98.4 wt% at 193°C.
2. Atmospheric Pressure, Stripping Gas –
 - a. Proceed vertically from the desired wt% TEG in lean glycol solution (top abscissa) to the temperature line and then horizontally to the stripping gas curves.
 - b. Proceed vertically from the calculated wt% TEG in rich glycol leaving the absorber (bottom abscissas) to the stripping gas curves.
 - c. Intersection of the two lines (from steps a and b) fixes the amount of stripping gas.
3. Vacuum, No Stripping Gas –
 - a. Proceed vertically from intersection of zero stripping gas line and temperature line to atmospheric line (760 mmHg).
 - b. Proceed horizontally from the atmospheric pressure point in step (a) to pressure line necessary to fix value of the wt% TEG in lean glycol solution.

In the rare case where both stripping gas and vacuum are used, procedures (2) and (3) are combined.

As a general rule, vacuum is avoided unless necessary to simplify the unit operation. Air (oxygen) ingress accelerates glycol degradation.

Figure 3.6 is based on one equilibrium stage in the regenerator. Most regenerators will contain more than one equilibrium stage, particularly if a stripping column is installed between the reboiler and surge tank. Figure 3.7 shows lean TEG concentration vs. stripping gas rate for units employing a stripping column. In Figure 3.7, “N” refers to the number of theoretical stages in the stripping column.

Stripping gas rates seldom exceed $75 \text{ m}^3 \text{ (std)/m}^3 \text{ TEG}$ unless lean TEG concentration in excess of 99.99 wt% are required.

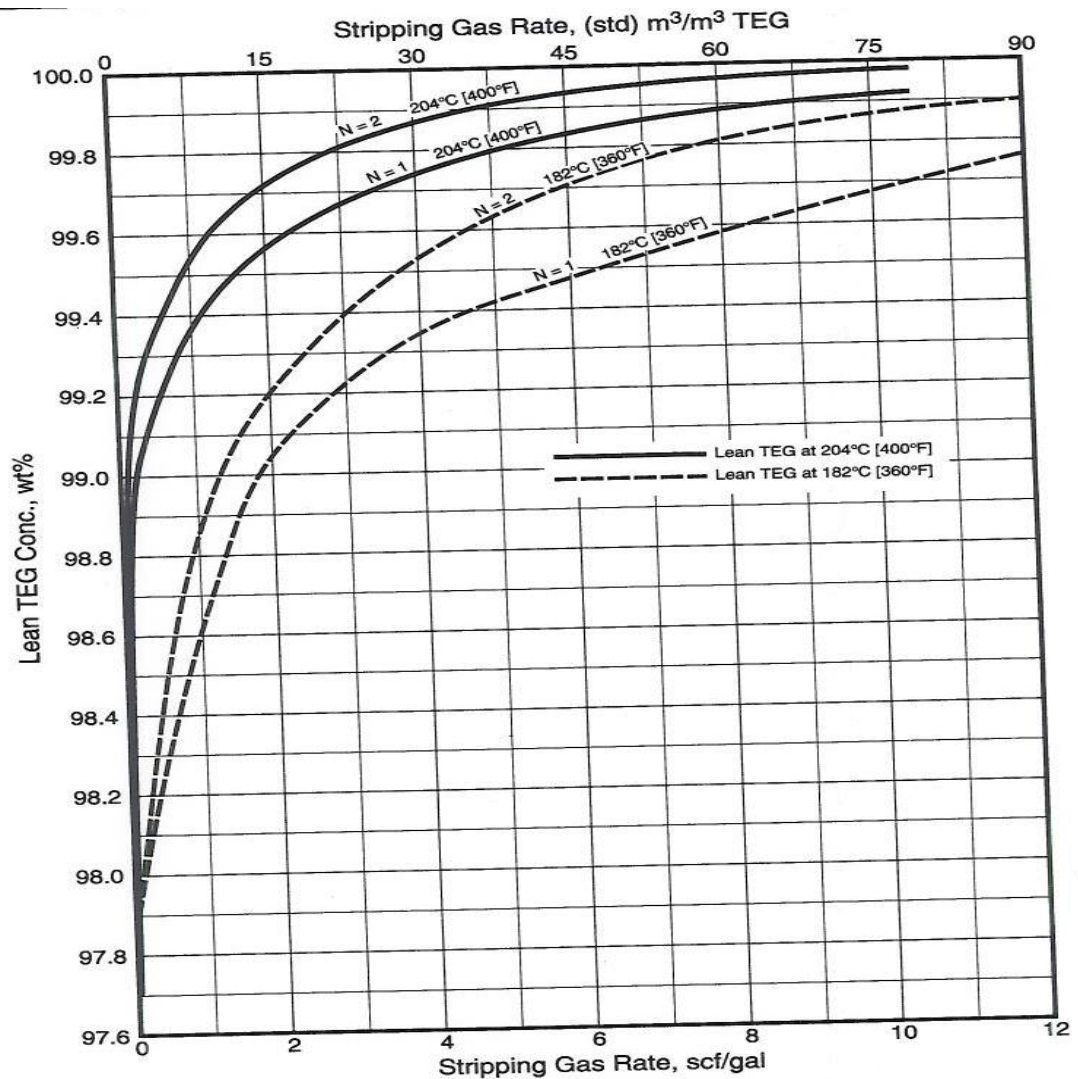


Figure 3.7: Effect of Stripping Gas on Lean TEG Concentration for Regenerators with Stripping Columns (Courtesy J. Campbell)

3.4.2.1 Reboiler

Heat input to the regenerator is provided in the reboiler. The heat source is usually direct fired with the fire-tubes immersed in a glycol bath. Other heat sources include hot oil (or other heat transfer fluid), steam or electric resistance heating.

Whichever heat source is used, the temperature of the glycol in the reboiler must not exceed 204°C due to the degradation of TEG at higher temperatures. However, in order to maintain the glycol bath at a temperature at or slightly above 204°C it is necessary to maintain the heat transfer surface of the fire-tubes above this value. This can lead to some degradation of the glycol in contact with the heat transfer surface.

For this reason the following maximum flux rates should not be exceeded. These flux rates, in turn, will set the heat transfer area (Campbell, 2004).

- Direct Fired – 19 kW/m^2
- Steam – 24 kW/m^2
- Hot Oil – 24 kW/m^2
- Electric – 12.5 kW/m^2

Most glycol reboilers maintain the bath temperature near 204°C . Lower temperatures may reduce degradation but result in lower lean TEG concentrations which, in turn, necessitate higher circulation rates or higher stripping rates.

The reboiler duty depends on the TEG circulation ratio (liters TEG/kg H_2O absorbed), the efficiency of the rich-lean exchanger, the reflux ratio, stripping gas rates and effectiveness of the insulation. The typical heat balance will indicate a reboiler load (duty) of 250-300 kJ/liter of TEG circulated. The usual practice is to size the reboiler to deliver 130-140% of the expected duty to provide for start-up, insulation losses, etc. A design value of 400 kJ/liter will typically provide sufficient heat input flexibility to meet any expected operating condition. The reboiler duty is always sized based on circulation pump capacity, not the expected circulation rates (Campbell, 2004).

3.4.2.2 Still Column

The still column is the “fractionator” portion of the regenerator. The column may be packed or trayed. Packed columns are more common and the packing is typically a random packing such as steel slotted rings. Packing sizes range from 16 mm to 51 mm depending on the still column size with the larger packing used in the larger diameter columns. Structured packing can also be used, and in very large units (diameters $> 1 \text{ m}$), trays are installed.

The still column is sized based on standard packed tower sizing correlations. Since the vapour loading is often tied to the glycol circulation rate, many correlations have been developed which estimate the still column diameter as a function of TEG circulation rate. Since the vapour load is low, the sizing of this column is not critical. It should, however, conform generally to the diameter approximated in Equation 3.19 (Campbell, 2004).

$$d = (A)(q_{TEG})^{0.5} \quad 3.19$$

where:

d = diameter of packed column / tower (mm)

q_{TEG} = glycol circulation rate (m³/h)

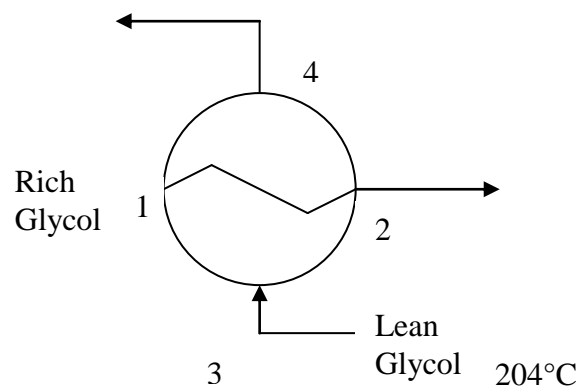
A = empirical constant = 210

The reflux ratio (liquid/distillate) employed in TEG systems is very small. The values typically range from 0.1 to 0.2 mol/mol. This is equivalent to condensing 10-20% of the total overhead vapour stream. The reflux rate should be the minimum required to maintain the still overhead temperature at the boiling point of water for the partial pressure of water at the top of the regenerator. When stripping gas is used, the water partial pressure will be less than 1 atm; hence the boiling point of water will be lower as well.

Reflux is normally supplied by using the rich glycol stream circulating through a condensing coil inserted in the top of the still column. Heat transfer coefficients in the reflux coil typically range from 110-230 W/m²K (Campbell, 2004).

3.4.3 Glycol-Glycol (Lean-Rich) Heat Exchanger

This is a basic heat exchanger and its efficiency has a direct effect on reboiler heat load.



The rich glycol from the absorber enters at about the temperature of the inlet gas (point 1). The lean glycol from the regenerator (point 3) enters usually at about 204°C. The temperature of the lean glycol at point 4 should not be greater than 60-65°C.

In most cases a 15-20 K approach in the heat exchanger is desirable. If it is too high the reboiler and glycol cooler duties will increase.

In this exchanger,

$$Q_{3-4} = Q_{1-2} = m_1 (h_2 - h_1) = m_3 (h_3 - h_4) \quad 3.20$$

where:

h = enthalpy (kJ/kg)

Q = heat transfer (kW)

m = mass flow (kg/s)

The easiest way to make this enthalpy calculation is to look up the average specific heat of the glycol and multiply it by ΔT across the exchanger to find Δh for the lean mixture ($\Delta h = C_p \Delta T$).

If an efficient glycol-glycol exchanger is used there may be no need for an additional cooler on the lean glycol stream. The most common types of glycol-glycol exchangers are pipe-in-pipe and plate. The former are used on small units, while the latter find extensive use in larger units and offshore. Plate exchangers provide effective heat exchange but are very susceptible to fouling in dirty service. Clean filtered glycol is imperative if plate exchangers are used (Campbell 2004).

3.4.4 Lean Glycol Cooler

Often, a final glycol cooler is required so that the lean glycol entering the top of the contactor (absorber column) is cooled to within 5-10 K of the gas temperature entering the top tray. The lean glycol is often cooled with a lean glycol-gas heat exchanger. Gas-glycol exchangers are inexpensive and compact, but the lean TEG temperature can increase to unacceptable levels at low gas rates. Air or water-cooled glycol coolers are also common since control of glycol temperature is then independent of the gas flow (Campbell, 2004).

3.4.5 Filters

Good filtration is critical. The full-flow type is preferred – two filters in parallel, with no by-pass lines, so that full filtration is assured.

A cloth fabric element that is capable of reducing solids to about 100 ppm by weight is preferred. Paper and fiberglass elements generally have proven unsatisfactory. Filter size in a properly operated glycol system should be 5-10 μm . The larger sizes (25-50 μm) may be required during start-up and in dirty service.

It is impossible to judge the effectiveness of filtration by colour alone. Even well filtered glycol will often be black. However, removal of most of the solids will reduce corrosion, plugging and deposits in the reboiler, and may reduce foaming - a major source of glycol loss (Campbell 2004).

3.4.6 Surge Drum

The surge drum is sized to provide the following:

- a retention time not less than 20 minutes between low and normal levels, based on design circulation rate;
- hold-up capacity between normal and high level;
- a reasonable length of time between glycol additions;
- a sufficient volume to accept the glycol drained from the reboiler to allow repair or inspection of the fire-tube or heating coil.

It is often vented to the reboiler. If it is not located directly under the reboiler, it is provided with a separate purge gas supply, e.g., nitrogen or dry fuel purge gas (Campbell, 2004).

3.4.7 Glycol Circulation pumps

Glycol circulation pumps may be electric driven or gas powered. The pumps are usually sized to provide a minimum of 25 percent excess capacity.

Pumps utilizing conventional electric drives are normally reciprocating type. A conservative, slow piston speed (0.6 m/s) pump is preferred since the lubricating properties of glycol are poor. Due to the expected turn-down of glycol systems, variable speed drives are often used. They can provide the flexibility to increase the

glycol circulation rate if needed to meet dew point requirements and reduce the glycol rate to operate more economically.

3.4.8 Circulation Rate – Absorber Contacts

The absorber (contactor) is where the water is removed from the gas by the process of physical absorption. The amount of water removed depends on three factors:

- glycol concentration
- glycol circulation
- number of contacts (e.g., trays) in the absorber

The effect of glycol concentration on water removal is illustrated in Figure 3.5. As the glycol (TEG) concentration increases, the equilibrium water dew point decreases. In an absorber this would mean a lower water concentration in the gas at the absorber outlet, hence higher water removal.

In a real absorber the gas leaving the absorber does not reach equilibrium with the incoming lean TEG. This is the reason for the “approach” discussed in Sub-section 3.4.1. Approach depends on two parameters – circulation rate and number of contacts. The theoretical outlet water dew point is set by the lean glycol purity (concentration). The actual water dew point depends on the circulation rate and the number of trays or height of packing. This is illustrated in Figure 3.8.

As seen in Figure 3.8, adding contacts (trays or packing) or increasing the circulation rate increases water removal. However, the curves in Figure 3.8 reach an asymptotic value of about 0.94. This water removal represents the equilibrium dew point condition at the top of the contactor. If we have an infinite number of contacts or an infinite circulation rate, the water removal percentage would still be 94% and the approach would be zero. At this point, the water removed depends exclusively on the lean TEG concentration.

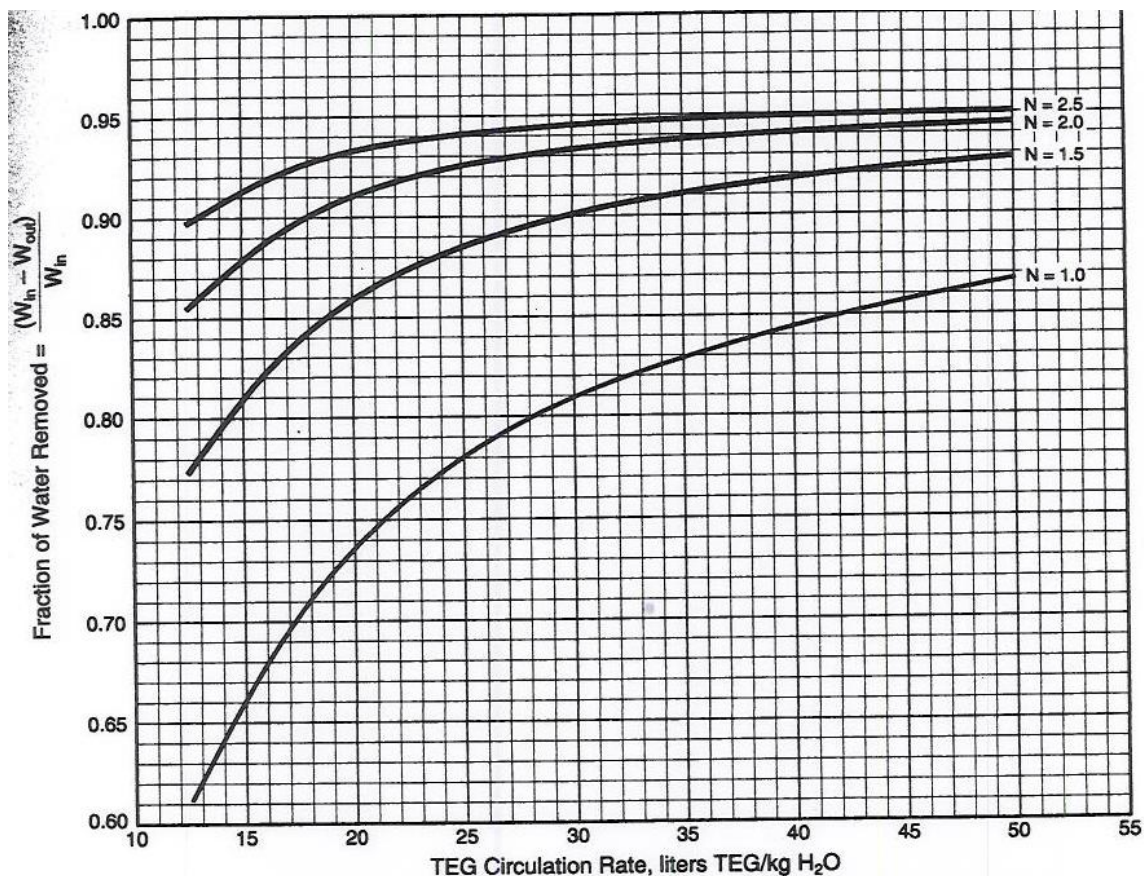


Figure 3.8: Water Removal vs. TEG Circulation Rate for Several Contactor (Absorber) Theoretical Stages (Courtesy J. Campbell)

Although a number of circulation rates are possible, the minimum feasible one will be used. The cost of purchasing and operating any absorption unit is a function of the circulation rate. As circulation increases, so does operating cost. The minimum feasible rate is fixed by the absorber characteristics (necessary to meet absorption specifications) and cost. The goal is to design a glycol system which is economically viable. There exist an infinite number of choices to meet the outlet dew point specification (e.g., we can select a contactor with several contacts and a low circulation rate or one with few contacts and a high circulation rate).

Most designs use a circulation rate of 15 – 40 liters TEG/kg H₂O absorbed. This is near the economic optimum. Higher circulation rates result in a larger regeneration system and higher energy consumption. Lower rates require a taller contactor and may result in poor tray / packing hydraulics.

For a given circulation rate a given number of absorber contacts are needed. The relationship between rate and amount of contact is described by the Kremser-Brown absorption equation (Campbell, 2004):

$$E_a = \frac{y_{N+1} - y_1}{y_{N+1} - y_0} = \frac{A^{N+1} - A}{A^{N+1} - 1} \quad 3.21$$

where:

E_a = efficiency of absorption

y_{N+1} = mol fraction water in entering wet gas

y_1 = actual mol fraction water in dried gas leaving

y_0 = water content of dried gas if it is in equilibrium with the entering lean glycol (value is less than y_1)

A = absorption factor, $A = \frac{l}{k \dot{v}}$

l = glycol circulation rate, moles/unit time

\dot{v} = gas flow rate, moles/unit time

k = equilibrium constant for water between water in gas and water in a TEG solution, $y = kx$

N = number of theoretical plates (trays) in the absorber

The mol fraction water, y_w , is related to the mass of water per standard volume flow rate of gas, w , by a fixed conversion factor, B ,

$w = B y_w$ where: $w = \text{kg}/10^6 \text{ std m}^3$

$y_w = w/B = 1.33 \text{ (E-06)} w$ $B = 761 \text{ 000}$

Thus, Equation 3.21 may be written in terms of w . Thus,

$$E_a = \frac{y_{N+1} - y_1}{y_{N+1} - y_0} = \frac{w_{N+1} - w_1}{w_{N+1} - w_0} \quad 3.22$$

Or,

$$E_a = \frac{w_{N+1} - w_1}{w_{N+1} - w_0} = \frac{A^{N+1} - A}{A^{N+1} - 1} \quad 3.23$$

where: w_{N+1} = water content of entering (wet) gas – $\text{kg}/10^6 \text{ std m}^3$

w_1 = water content of leaving (dry) gas - $\text{kg}/10^6 \text{ std m}^3$

$$w_0 = \text{water content of gas in equilibrium with incoming lean TEG - kg/10}^6 \text{ std m}^3$$

w_0 is a function of the lean TEG concentration and contactor temperature (T) and pressure (P).

The subscripts on “ w ” have the same significance as on “ y ”.

For a given calculation the values of inlet water content, outlet water content, gas flow rate, and absorber pressure and temperature are fixed. Using a correlation for determining the equilibrium k value of water in a TEG-water system, values for y_0 (or w_0) and k are available. The only variables left in Equation 3.21 are l (the TEG circulation rate) and N (the number of theoretical trays). In theory, there are an infinite number of combinations of l and N that satisfy Equation 3.21. In practice the choices are limited by economics and absorber performance.

The cost of purchasing and operating any absorber unit is a function of circulation rate. It is thus good practice to operate at, or near, the minimum rate necessary to meet absorption specifications.

Figure 3.9 is a plot of Equation 3.23 that is convenient for manual calculations. This uses what could be called an “overall absorption factor”. l_0 is the rate of lean TEG entering the top tray and v_{N+1} is the gas rate entering the bottom tray. Even though the ratio $\frac{l}{v}$ varies slightly throughout the absorber the effect of these changes are effectively cancelled by changes in the “ k ” value, so use of $(l_0/v_{N+1}k)$ as the average absorption factor has little effect on the accuracy of the method. The variation of $\frac{l}{v}$ will have a calculable, but usually non-significant, effect on unit design.

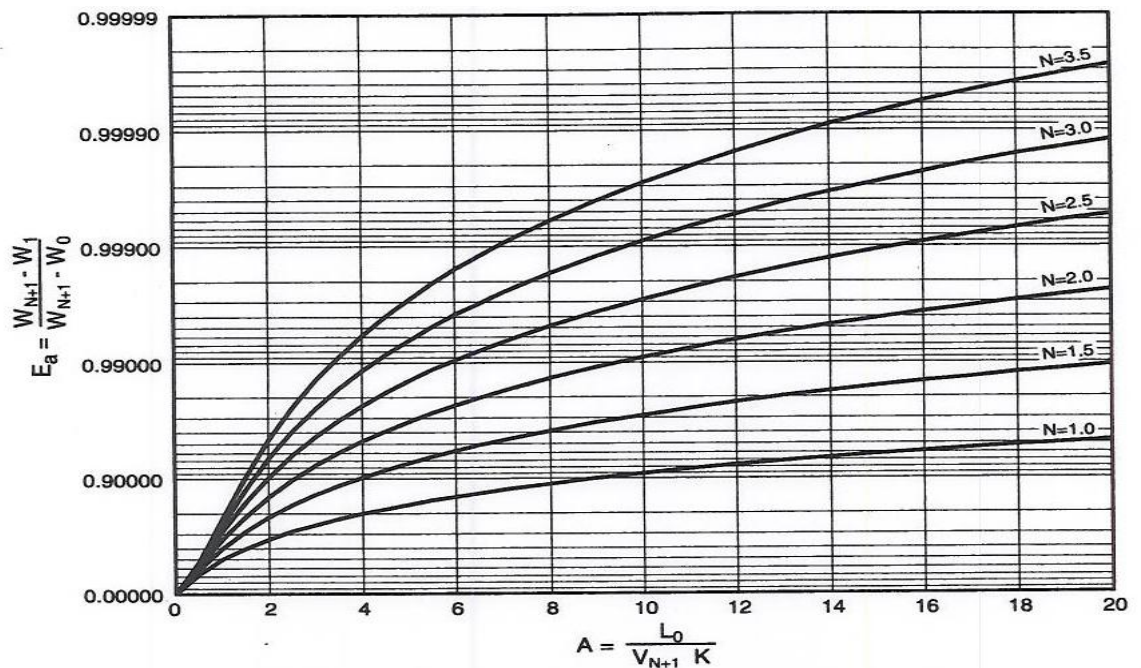


Figure 3.9: Plot of Kremser-Brown Equation (Courtesy J. Campbell)

The left-hand ordinate of Figure 3.9 is called “absorption efficiency” of absorption, E_a . It is the actual amount of water removed, divided by the maximum amount theoretically removable. The values of N encompass the range of theoretical trays usually employed.

In order to solve for l it must be recognized that it is different at each point in the absorption tower. The conservative approach is to assume the gas volume is constant and solve for l_o from A . This will yield a circulation rate sufficiently close to a more rigorous calculation.

It is usual to repeat the calculation for lean TEG rate or number of theoretical trays, N (depending on which is unknown) to obtain three lean glycol rate / absorber contact values that satisfy the required absorption efficiency. The final choice is economic. This usually involves selection of standard modules to make up the unit.

The calculation is usually made at the lowest pressure and highest temperature anticipated for the entering wet gas, to obtain the maximum water loading. Historically, the tendency has been to choose a design temperature lower than that actually obtained.

The overall tray efficiency in a well-designed TEG unit will vary from 25 – 40%. It

is usual to use 25% for most applications. This provides an affordable safety factor to help compensate for inherent errors in the design specifications.

3.4.8.1 Equilibrium Relationships

Various studies have been made of the equilibrium behaviour of water in the TEG-water system. The use of an activity coefficient (γ) is a convenient and reliable method of calculating k (Campbell, 2004). Using this relationship:

$$k = (y_w)(\gamma) = \frac{(w)(\gamma)}{B} \quad 3.24$$

where:

k = equilibrium constant for water in a TEG-water system

y_w = mol fraction water in the gas at saturation

γ = activity coefficient for water in the TEG-water system as found from

Figure 3.10

w = water content on a mass per volume basis, at saturation, as found from a regular water correlation

$B = 761\,000$ when $w = \text{kg}/10^6 \text{ std m}^3$ (measured at 15°C and 100 kPa)

γ , and thus k , vary with TEG concentration, which in turn varies (changes) throughout the absorber. Consequently, a mean or average k cannot be determined until the circulation rate is fixed. This involves a trial and error calculation. The inlet TEG concentration can be assumed as a first attempt. In most cases, both l and k are determined at top tray conditions, i.e., using the lean TEG concentration and lean TEG circulation rate. For most dehydration applications the increase in k from the lean to rich TEG is roughly proportional to the increase in $\frac{l}{v}$, so the absorption

factor ($A = l/kv$) remains relatively unchanged throughout the absorber.

In the absorption efficiency term,

$$y_o = k x_o \quad \text{and} \quad w_o = (w)(\gamma)(x_o) \quad 3.25$$

where:

x_o = mol fraction water in lean TEG entering the absorber

x_o may be calculated from X_{gl} , the weight percent TEG in the lean solution entering the absorber. This must be not less than the minimum value required from Figure 3.5.

$$x_o = \frac{(100 - X_{gl})/18}{[(100 - X_{gl})/18] + (X_{gl}/150)} \quad 3.26$$

Equation 3.26 is shown graphically in Figure 3.11

The above procedure using the Kremser-Brown approach is as good as a tray-to-tray balance around the absorber (it gives results comparable to computer simulation). This is true because there is little gas shrinkage and A is essentially constant throughout the tower.

The use of other equilibrium k values will have little effect on contactor (absorber) design. The required lean glycol concentrations may differ but the difference is normally less than the random error in process specifications and is statistically insignificant (Campbell, 2004).

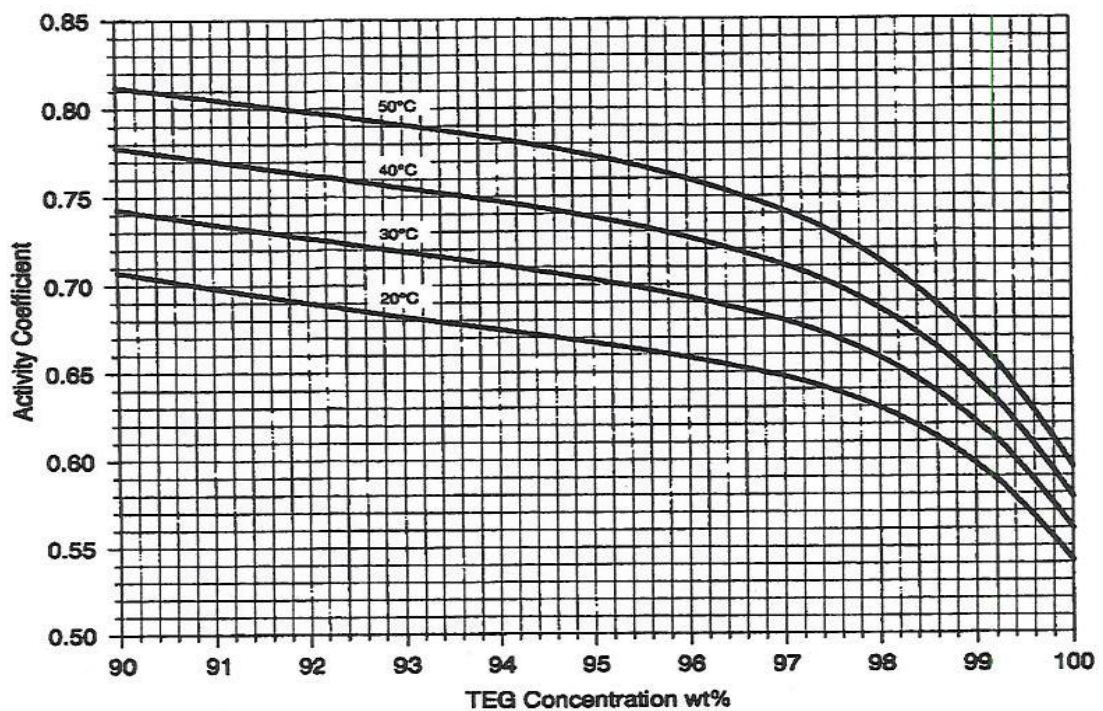


Figure 3.10: Activity Coefficient for H₂O Concentration at Various Temperatures (Courtesy J. Campbell)

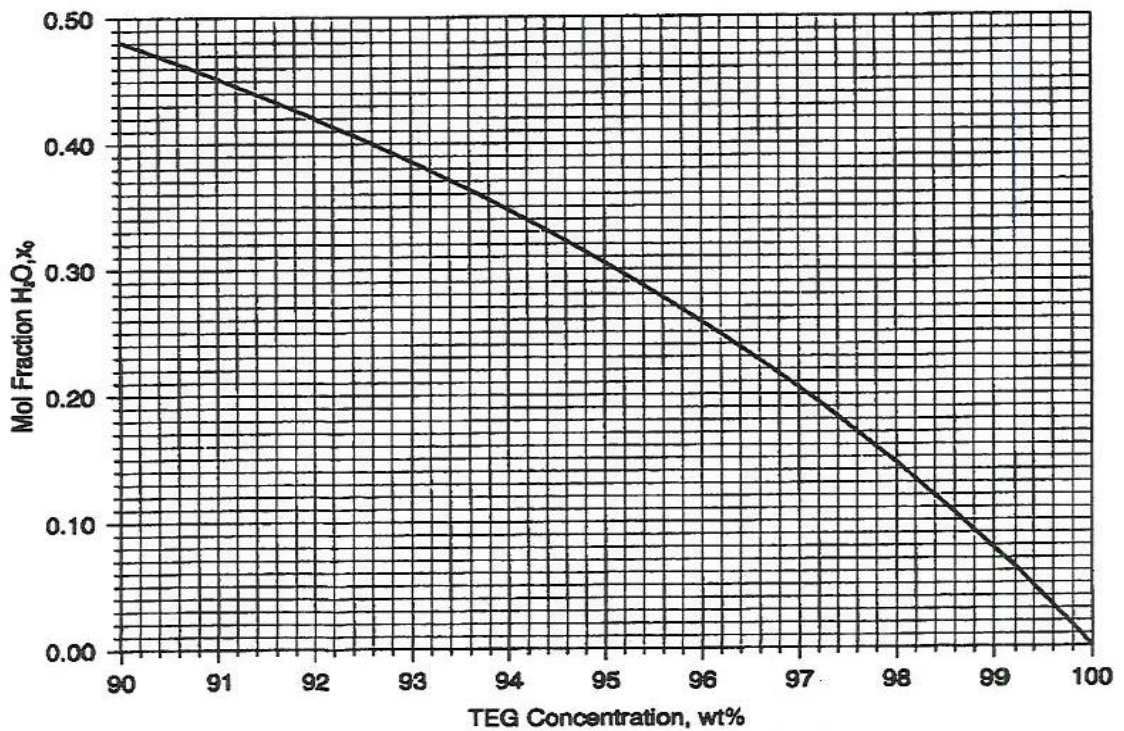


Figure 3.11: Mol Fraction H₂O, x_0 vs. TEG Concentration, X_{g1} (Courtesy J. Campbell)

When using the Kremser-Brown method, the terms v and l must be expressed in molar units. For the glycol solution this requires estimation of the molecular weight (MW). The molecular weight of a TEG-water solution may be calculated as follows (Campbell, 2004):

$$MW = 18x_o + 150(1 - x_o) \quad 3.27$$

3.4.9 Absorber Design

The design of the absorber (contactor) is based on two parameters:

1. gas rate which determines the contactor diameter, and
2. number of contacts, which determine the contactor height.

The contactor diameter depends almost exclusively on the gas rate and is virtually independent of the glycol rate. This is due to the low liquid loadings employed in glycol contactors. Two types of contactor internals are used in glycol systems – trays and packing.

When tray contactors are used, bubblecaps are preferred over other types of trays (valve and sieve) due to their higher turndown ratio and generally better efficiency at low liquid rates. Likewise, when a packed contactor is employed, structured packing is preferred over random packing because of its higher capacity, better turndown and superior performance at low liquid rates. Figure 3.12 shows a picture of a bubblecap tray, a structured packing and a pallring random packing.

Regardless of the type of tray, a minimum spacing of 24 inches (610 mm) is recommended. It is essential that stable foam does not fill the gas space between the trays to prevent excess glycol loss. This spacing also allows a suitable liquid level in the downcomers.

Bubblecap trays have been historically used in TEG systems. However, in recent times, many contactors have been installed with structured packing. This is primarily due to the superior gas handling capacity of structured packing relative to trays (Campbell, 2004).

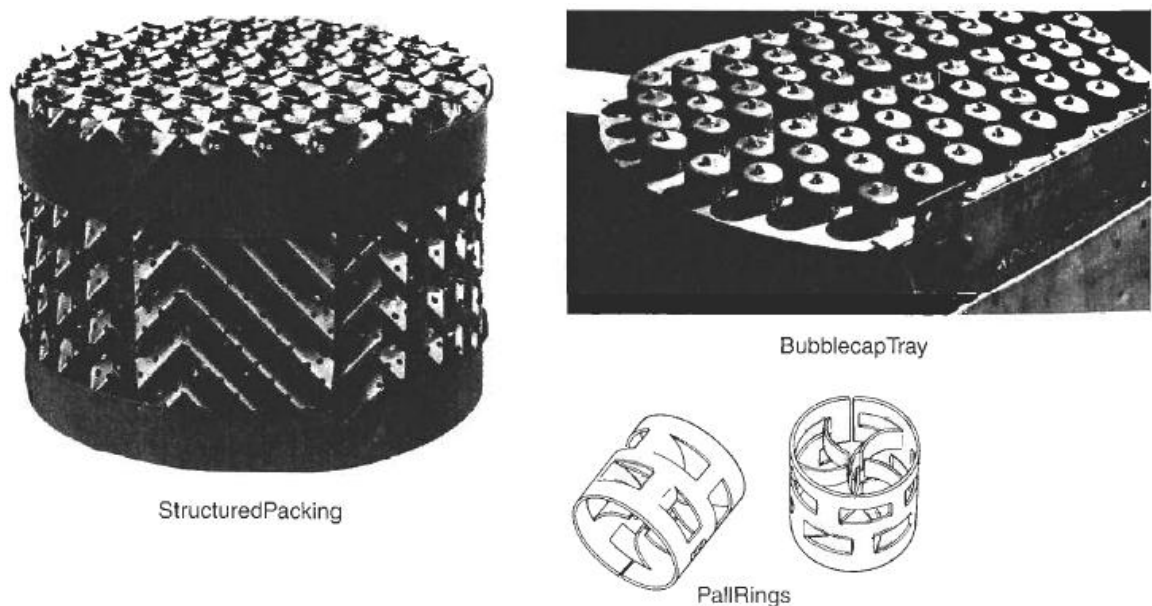


Figure 3.12: Bubblecap Tray, Random and Structured Packing
(Courtesy J. Campbell)

3.4.9.1 Contactor Diameter

As earlier stated, the diameter of a glycol contactor is determined, almost exclusively by the gas rate. If a contactor is operating near flood, changes in the glycol rate can have a noticeable effect on the glycol carryover, but for design purposes the liquid rate is typically not a factor in absorber sizing.

The calculation of diameter can proceed in two ways. The first method employs the Souders Brown equation below, frequently used to size separators (Campbell, 2004). That is,

$$v = K_s \left(\frac{\rho_L - \rho_g}{\rho_g} \right)^{0.5} \quad 3.28$$

where:

v = allowable gas velocity (m/s)

K_s = sizing parameter (0.055 m/s for bubblecap trays and 0.09-0.105 m/s for structured packing)

ρ_g = gas density (kg/m³)

ρ_L = liquid density (kg/m³) = 1120 kg/m³ for TEG systems

The calculation of the diameter follows:

$$d = \sqrt{\frac{4q_a}{\pi v}} \quad 3.29$$

where:

d = contactor diameter (m)

q_a = actual gas flow rate (m³/s)

The actual gas flow rate can be calculated from either the mass flow rate or the standard volumetric rate. That is,

$$q_a = \frac{m}{\rho_g} \quad 3.30$$

where:

m = mass flowrate

ρ_g = gas density (kg/m³)

Or,

$$q_a = \frac{q_{std}}{86400} \left(\frac{P_{std}}{P_a} \right) \left(\frac{T_a}{T_{std}} \right) Z \quad 3.31$$

where:

q_{std} = gas flowrate in standard volumes (std m³/d)

P_{std} = standard pressure (kPa)

P_a = actual flowing pressure of gas (kPa)

T_a = actual flowing temperature of gas (K)

T_{std} = standard temperature (K)

Z = gas compressibility factor at flowing conditions (dimensionless)

A second sizing equation uses F_s value which is related to the kinetic energy (per unit volume) of the gas (ρv^2). This is the more popular method of sizing packed towers.

$$v = \frac{F_s}{\sqrt{\rho_g}} \quad 3.32$$

where:

v = allowable gas superficial velocity (m/s)

F_s = sizing parameter = $P_a^{0.5}$

The sizing parameter, F_s , depends on the type of packing density, but for most structured packing, $F_s = 3.0$ in SI units.

In an actual design it is sound engineering practice to size the contactor for a gas rate 20 – 30% higher than expected rate. This contingency provides contactor capacity for changes in flow rate and pressure (Campbell, 2004).

3.4.9.2 Contactor Height

The contactor height is determined by the number of equilibrium contacts required and efficiency of the mass transfer. For trayed contactors the conversion from equilibrium (theoretical) stages to actual trays is accomplished by using tray efficiency. The tray efficiency is a measure of the approach to equilibrium and can be calculated from either vapour phase or liquid phase compositions.

The overall tray efficiency is defined as:

$$E_{overall} = \frac{\text{No. of Equilibrium Stages}}{\text{No. of Actual Trays}} \quad 3.33$$

For glycol contactors using bubblecap trays E_{overall} typically ranges from 25 – 30%. For most engineering calculations an overall efficiency of 25% will yield satisfactory results (Campbell, 2004).

A minimum spacing of 610 mm is recommended. As stated previously, it is essential that a stable foam does not fill the gas space between trays to prevent excessive glycol loss. This spacing also allows a suitable liquid level in the downcomers, as well as allows space for vapour disengagement above the top tray, inlet gas distribution below the bottom tray and rich glycol surge volume at the bottom of the column. Therefore, the total height of the absorber column will be based on the number of trays (or packing) required plus additional 1.8 to 3 m typically (GPSA, 1998).

Tray hydraulics design is critical because of the low circulation rate. Liquid can bypass caps or valves in some areas of the tray, ineffective gas-liquid contact can occur with low gas rates and tray liquid levels can be unstable. In situations like this, absorber performance can vary markedly with gas and liquid rate. A higher than calculated liquid rate may be necessary to provide the tray efficiency required.

For packed towers, equilibrium stages are converted to packing heights using an *HTU* (Height of Transfer Unit) or *HETP* (Height Equivalent to a Theoretical Plate / Stage).

HETP and *HTU* are related concepts and depend primarily on the gas and liquid properties, gas and liquid rates as well as the surface characteristics and density of the packing.

A *HETP* value of about 1.5 m is usually adequate for preliminary design or planning purposes. However, when the gas density exceeds about 100 kg/m^3 (generally at very high pressures), this conversion may not provide sufficient packing height (Campbell, 2004).

The mass transfer in a packed tower is continuous, and does not take place in discrete steps as in trayed towers. It is for this reason that many packing manufacturers prefer to work in *HTUs*. The *HTU* is multiplied by the Number of Transfer Units (*NTUs*) to arrive at the packing height. For overall mass transfer controlled by resistance on the gas side, the number of transfer units may be calculated from the following equation

(Campbell, 2004):

$$NTU = \int_{y_t}^{y_b} \frac{dy}{y - y^*} \quad 3.34$$

where:

NTU = Number of Transfer Units

y = concentration of water in the inlet gas (air)

y_b = concentration of water in the bottom of the contactor

y_t = concentration of water in the top of the contactor

y^* = equilibrium water concentration

A typical HTU for structured packed towers depends on gas density and packing type but for packing with specific area of $250 \text{ m}^2/\text{m}^3$ an HTU of 0.8 m gives good results for preliminary calculations (Campbell, 2004).

3.5 SUMMARY

In this chapter, the following were presented:

- Cooling tower analytical theories relevant to this study.
- Cooling tower heat transfer and energy analysis.
- Cooling tower performance criteria and procedure
- Glycol dehydration system analytical theories relevant to this study.

The next chapter presents the details of the modelling and sizing of the proposed cooling tower air conditioning system.

CHAPTER 4

MODELLING

4.1 INTRODUCTION

This chapter presents the details of the integrated modelling that was carried out in this research for the NLNG plant cooling tower, the proposed cooling tower air conditioning system based on the use of glycol (TEG) liquid desiccant and the gas liquefaction plant. This involves the modelling of the individual equipment / sub-systems and the synthesizing of these into the complete integrated system.

Modelling was used in this study to help understand the performance of the proposed cooling tower air conditioning system vis-à-vis its impact on the thermal performance of the cooling tower, the energy consumption of the gas liquefaction process and the liquefied natural gas production at the Nigeria Liquefied Natural Gas plant. The results from the simulations using the model are discussed in Chapter 5.

TRNSYS and EES modelling tools were used for the modelling and simulation, as explained in Section 4.6 of this chapter.

4.2 PROCESS FLOW DESCRIPTION

Figure 4.1 shows a schematic of the cooling tower with the proposed air-conditioning system that was modelled and simulated.

Moist (un-dehydrated) air is drawn from the atmosphere by an air blower and enters the absorber near the bottom. The air flows upwards through the internal trays of the absorber. Lean TEG enters the absorber near the top and cascades down through the absorber internals, making contact with the up-flowing air stream. The counter current flow path of the glycol (TEG) and the high contact surface area absorbs water into the glycol from the air stream.

Dehydrated air flows out of the top of the absorber into the cooling tower inlet, mixing with un-dehydrated air drawn directly from the atmosphere into the cooling tower by the induced draft fan at the top of the tower. Rich glycol flows out of the bottom of the absorber to the glycol Regeneration Package.

The TEG regeneration process involves passing the rich glycol through the Still Column to gain some heat before entering the Flash Drum. The glycol is then passed through a filter to remove particulates. The rich glycol is heated in a Lean-Rich Glycol (Interchange) Exchanger to preheat the feed to the Still Column where glycol present in the water vapour leaving the Reboiler is recovered.

The lean glycol concentration is increased by heating in the Reboiler using heat transfer fluid (HTF). Stripping gas can be used to further increase the lean glycol concentration. Lean TEG leaving the bottom of the Reboiler flows into the Surge Drum from where it is pumped back to the top of the absorber column, through the Lean-Rich Glycol Exchanger and the Lean Glycol Cooler, to repeat the process.

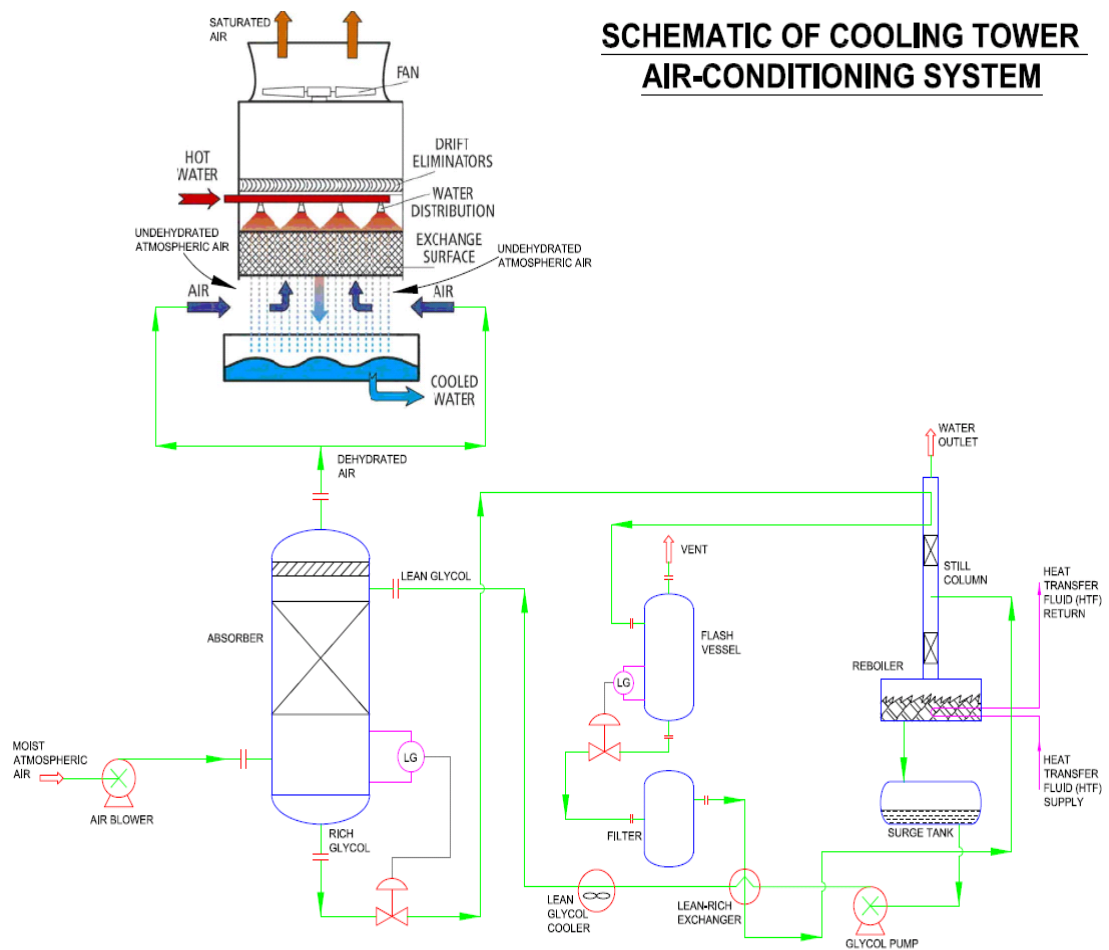


Figure 4.1: Schematic of cooling tower air-conditioning system

4.3 DESIGN CRITERIA

The design data of the cooling tower of the Nigeria Liquefied Natural Gas (NLNG) plant and the climatic conditions of Bonny Island (Nigeria) have been used as the basis for the modelling and component sizing of the cooling tower air conditioning system and to calibrate and validate the TRNSYS and EES models described later in this chapter.

Table 4.1 gives the design conditions of the NLNG plant cooling tower.

Table 4.1: Manufacturer's Design Data for NLNG Cooling Tower

DATA	DESIGN VALUE
Dry Bulb Temp.	30.0 °C
Wet Bulb Temp.	28.0 °C
Cold Water Temp.	32.0 °C
Hot Water Temp.	40.1 °C
Circulating Water Flow rate	28000 T/h (7777.778 kg/s)
Ambient pressure	1.006 bar (100.6 kPa)
Fan Drive Power Input	170 kW (per cell)
Number of cells	9
Number of cells in service at a time	8
Air density	1.057 kg/m ³
Air flow	535 m ³ or 566 kg/s (per cell)
Water to Air Ratio (L/G)	1.53
Cooling Tower Characteristics (KaV/L)	1.69
Cooling Tower Constant (n)	-0.7
Relative Humidity	86 %
Cooling Tower Heat Load	266MW

Other design data:

- 1) System Elevation- The cooling tower elevation is based on the maximum heat exchanger elevation in the plant (i.e. the propane sub-cooler). The high point of the cooling tower is 23.72m above sea level. Positive pressure is ensured everywhere in the network under flowing conditions.
- 2) System volume- The cooling water volume in pipes and exchangers is approximately 1500m³. In addition, the cooling tower basin and pump pit holds a maximum water volume of 7200m³.

4.4 RESEARCH DESIGN SPECIFICATIONS

Table 4.2 gives the specified conditions for the modelling and sizing calculations of the cooling tower air conditioning system.

In addition to the specifications given in Table 4.2, the following were applied in the modelling and sizing exercise:

- Only 20 % of the design total air flow rate (for the 9-cell NLG cooling tower) is pre-conditioned (dehydrated) in the cooling tower air-conditioning unit. This flow limit has been set due to physical and thermodynamic (energy) limitations on equipment that exist in the current NLNG plant set-up. This slip stream of dehydrated air was mixed with the 80% un-dehydrated ambient air at the cooling tower inlet to obtain a resulting **mixed wet-bulb temperature** for the mixture of dehydrated and un-dehydrated air flows. Section 4.5 provides more details on the 20 % flow limit used for the cooling tower inlet air that was pre-conditioned.
- Minimum glycol losses (due to foaming and/or carry-over)
- Minimum power consumption
- Optimum plant efficiency and design integrity
- Compliance to health, safety and environmental (HSE) requirements

Table 4.2: Specifications for the Cooling Tower Air-Conditioning System

DATA	SPECIFICATION	DATA SOURCE / JUSTIFICATION
Air (Inlet Dry Bulb) Temp.	30.1 °C	Site real-time measurement
Hot Water Temp.	40.9 °C	ditto
Wet Bulb Temp of Un-dehydrated Inlet Air	28.12 °C	ditto
Circulating Cooling Water Flow rate	29300 T/h (8138.889 kg/s)	ditto
Ambient pressure	1.006 bar (100.6 kPa)	Site average
Fan Drive Power Input	170 kW per cell	Equipment data sheet
Air Density	1.11 kg/m ³	Calculated site average
Number of cells in service	9	Actual site condition
Air flow through Absorber	1018.8 kg/s	Asymtotic limit
Relative Humidity of Inlet Air to Absorber	86 %	Site yearly average
Relative Humidity of Outlet Air from Absorber	10 %	Specified
Reboiler Heat Transfer Fluid (HTF) Flow Rate	373 kg/s	Actual site condition
Reboiler HTF Inlet Temp.	235.0 °C	ditto
Glycol (TEG) inlet Temp. to Absorber	184.0 °C	Based on recommended approach (Campbell, 2004)
Glycol (TEG) Outlet Temp. from Reboiler	204.0 °C	GPSA, 1998

4.5 JUSTIFICATION FOR LIMIT ON PRE-CONDITIONED AIR

As stated in Section 4.4, only 20 % (1018.8 kg/s) of the total design air flow rate (5094 kg/s) of the NLNG cooling tower, has been pre-conditioned in this study. The limit of 20 % has been set as a result of the following:

1. Increasing the air flow rate through the absorber of the air conditioning unit results in an increase in the glycol (TEG) circulation rate in order to meet the absorber exit air specification (moisture content). On the other hand, an increase in the glycol circulation rate results in further reduction of the exit temperature of the heat transfer

fluid (HTF) used in supplying the heat for regeneration of the glycol in the reboiler. The HTF inlet temperature used for calculations and modelling is 235 °C. This is the HTF temperature at the NLNG plant which has been used as a case study for this research work. Similarly, the HTF flow rate of 373 kg/s which has been used for calculations is the same as that at the NLNG plant. The specified exit temperature of the glycol in the reboiler is 204 °C. Glycol will degrade above this temperature. In order for the HTF supplied to the reboiler (at the specified inlet temperature and flow rate) to raise the temperature of the glycol (in the reboiler) to 204 °C, the inlet temperature of the glycol to the reboiler (which is also the exit temperature of the glycol from the rich-lean glycol exchanger) is fixed at 184 °C. These HTF and glycol specifications impose a physical limit on the amount of air flow through the air conditioning unit.

2. Above 20 % of the air flow, the calculated exit temperature of the heat transfer fluid (HTF) from the reboiler of the air conditioning unit falls below the inlet temperature of the glycol (TEG) to the reboiler, at the given specifications in point 1 above. This is thermodynamically impossible (heat balance).
3. As a result of point 2, it is mathematically impossible to calculate the logarithm mean temperature difference for the reboiler due to negative temperature difference between the HTF exit temperature and the TEG inlet temperature. The threshold exit temperature of the HTF (asymptotic value) is 185 °C.
4. As a result of points 2 and 3, it is impossible to calculate the overall heat transfer coefficient of the reboiler.

Therefore, based on the specifications for the reboiler, it is not theoretically feasible to pre-condition more than 20 % of the design air flow rate of the NLNG cooling tower.

It is noted, however, that this air flow limit is not a general situation as it applies specifically to the NLNG Bonny Island plant which has been used as a case study for

this study. It is indeed possible to dehydrate 100 % of the air flow into a cooling tower if there are no constraining boundary conditions.

4.6 MODELLING

Various approaches to cooling tower modelling have been used by different authors. Some of these modelling approaches include the work of Lebrun and Aparecida (2002) titled “Cooling Towers – Model and Experimental Validation”. The study presented a simplified method for analysing the combined heat and mass transfer phenomena in a cooling towers using Merkel’s theory as the theoretical basis for the model. The model assumed the cooling tower to be a classical heat exchanger working in the wet regime. This exchanger is characterised by an overall heat transfer coefficient – area product, UA. A correlation of UA was generated from experimental analysis as a function of water and air flow rates entering the cooling tower. Braun et al (1989) used the Effectiveness approach in their work titled, “Effectiveness Models for Cooling Tower and Cooling Coils”. The effectiveness relationships were developed through the introduction of an air saturation specific heat. Lastly, in the “Toolkit for Primary HVAC System Energy Calculation”, prepared for ASHRAE by Lebrun et al (1999), the model of the counterflow cooling tower was based on the Merkel’s theory (the Lewis number, which represents the ratio of heat and mass transfer coefficients, is assumed to equal unity). However, it was also assumed that the moist air enthalpy can be defined by the wet-bulb temperature only and that the moist air can be treated as an equivalent ideal gas characterised by a mean specific heat, similar to Braun’s effectiveness approach.

The modelling of the cooling tower air conditioning system is based on the glycol dehydration theories as given in Chapter 3.

As previously stated, two simulation software tools have been used in the modelling of the integrated cooling tower, the cooling tower air-conditioning system as well as the gas liquefaction plant of NLNG. These are:

- 1) EES – Engineering Equation Solver (Academic Version)
- 2) TRNSYS – TraNsient SYstem of Simulation (version 16)

TRNSYS is a component based model while EES is an equation based model that may be used to write a program for simulation.

While TRNSYS was used for time-series¹⁸ modelling, the EES model developed in this study can only model averaged conditions. However, the EES model was used as focus of this study as TRNSYS does not have a component for modelling the liquid dehydration system required for the conditioning of the cooling tower supply air or the gas liquefaction plant. An integrated EES model was therefore developed for the cooling tower, the proposed air conditioning system and the gas liquefaction plant. TRNSYS was, however, required to calibrate and validate the EES model.

The EES model was used to assess the ability of the air conditioning system to improve the performance of the cooling tower, as discussed in Chapter 5.

TRNSYS was first used for the validation of the steady state design condition and time series modelling of the cooling tower based on manufacturer's design data and real-time dataset from the NLNG plant. TRNSYS was then used to calibrate / validate the main EES model.

The EES cooling tower model for this study was built based on Merkel's and Braun's approaches. However, the mass and heat transfer constants used are different (based on the experimental work of Lebrun J.) and the flow rate of water is not assumed to be constant; the effect of evaporation (although minimal) has been taken into account to obtain more accurate results. Braun's approach (Braun, 1989) is the basis of the TRNSYS model.

The EES models for the air conditioning system and the gas liquefaction plant were a novelty of this research work, as they were developed during the course of the study. This is described in more details in Sub-section 4.6.1. below.

The TRNSYS and EES models are presented in Appendix 2.

4.6.1 EES

The basic function provided by EES (Engineering Equation Solver) is the solution of a set of algebraic equations. The tool is also capable of solving differential equations, equations with complex variables, optimization, linear and non-linear regression,

¹⁸ A time series is a sequence of data points, measured typically at successive times spaced at uniform time intervals.

uncertainty analyses and generates plots. The tool also provides many built-in mathematical and thermophysical property functions (e.g., steam, refrigerant property tables etc.).

A device or process is modelled in EES by inputting the fundamental governing equations (e.g., energy balance, mass balance, etc.). The tool then automatically identifies and groups equations that must be solved simultaneously.

EES uses a variant of Newton's method to solve systems of non-linear algebraic equations. The Jacobian matrix needed in Newton's method is evaluated numerically at each iteration. Sparse matrix techniques are employed to improve calculation efficiency and permit rather large problems to be solved in the limited memory of a microcomputer. The efficiency and convergence properties of the solution method are further improved by the step-size alteration and implementation of the blocking algorithm which breaks the problems into a number of smaller problems which are easier to solve. Several algorithms are implemented to determine the minimum or maximum value of a specified variable (Klein, 2009).

In this study, EES was used to model:

- the cooling tower (design conditions and time-series point states);
- the liquid desiccant system (used for conditioning a “fraction” of the air);
- the mixing of dehydrated (conditioned) air (20%) and un-dehydrated air (80%) at the cooling tower inlet and;
- the gas liquefaction plant of NLNG (to assess the impact of the conditioning of air on overall plant energy consumption as well as liquefied natural gas production).

4.6.1.1 MATHEMATICAL DESCRIPTION OF THE COOLING

TOWER EES MODEL

As previously stated, the EES cooling tower model in this study was built based on Merkel's and Braun's analytical methods for cooling towers. The main difference between this and that described in the “Toolkit for Primary HVAC System Energy Calculation”, prepared for ASHRAE by Lebrun et al (1999), is that the effect of water loss due to evaporation was taken into account (similar to TRNSYS).

An EES “procedure” was used to write the model routine based on the relevant

equations.

The mathematical description of the cooling tower model is as follows:

- **The Full-Load Regime Cooling Tower Model**

The main outputs to be defined are:

- 1) The cooling tower useful output, i.e. its exit water temperature (to be compared with the set point / desired value) or its corresponding useful power (i.e. the rate of heat flow rejected to the environment).
- 2) The fan and pump electricity consumption.

The main inputs are:

- 1) The exit water set point temperature and the control mode.
- 2) The load characteristic i.e. the inlet water temperature, the wet bulb temperature¹⁹ of the surrounding air and the water and air flow rates (assumed to be fixed at different operating modes).

The parameters are the heat transfer coefficient-area products associated with the air flow rates.

The information flow diagram for the cooling tower is given in Figure 4.2.

The model of the counterflow cooling tower is based on Merkel's theory. Based on this theory, the total heat transfer (sensible plus latent) due to the direct contact between air and water under steady-state conditions is defined as (ASHRAE Toolkit, 1999),

$$dQ_{w-air} = \frac{U dA}{c_p} (h_s - h_a) \quad 4.1$$

where,

h_s is the enthalpy of saturated air (boundary film) at the wetted-surface temperature

h_a is the enthalpy of air in the free stream

c_p , the specific heat of moist air, is assumed to be constant and equal to $1025 \text{ J kg}^{-1} \text{ K}^{-1}$

¹⁹ The leaving wet-bulb temperature can be defined as a secondary output.

$U dA$ is the heat transfer coefficient-area product.

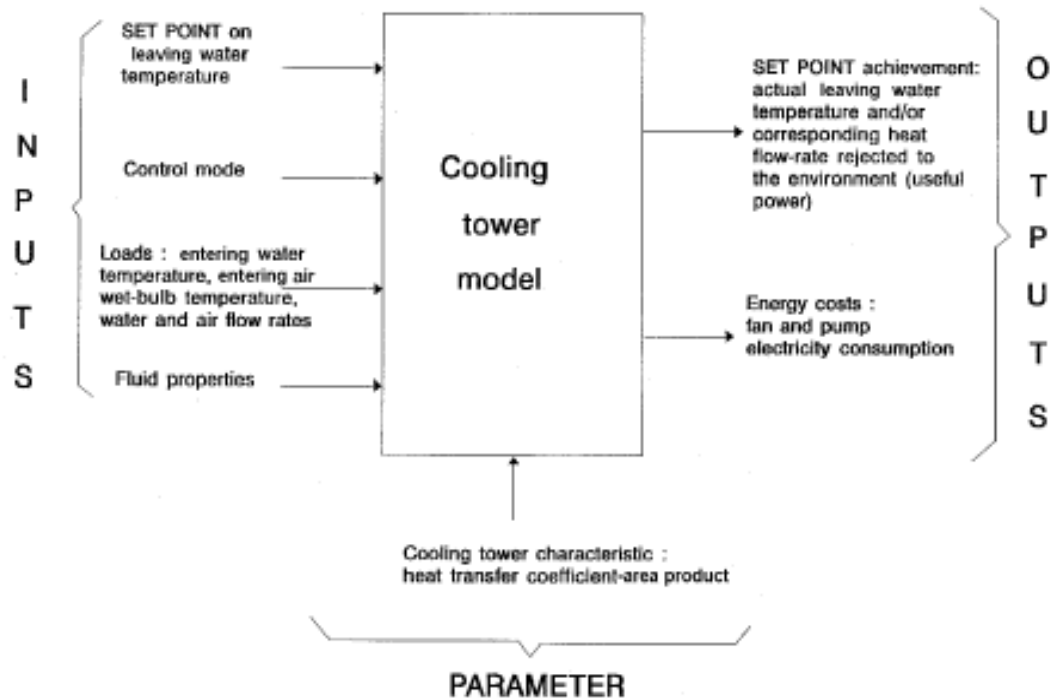


Figure 4.2: Information Flow Diagram of Cooling Tower Model
(Courtesy ASHRAE Toolkit)

Equation 4.1 is based on a series of assumptions:

- air and water vapour are assumed to behave as ideal gases;
- the inlet and exit water mass flow rates are assumed to be equal (i.e. the effect of water lost by evaporation is neglected);
- the air heating due to the fan is neglected;
- the interfacial air film is assumed to be saturated with water vapour;
- the Lewis number, which represents the ratio of heat and mass transfer coefficients, is equal to 1.

The first step in the modelling procedure is to assume that the moist air enthalpy can be defined by the wet-bulb temperature alone and that the moist air can be treated as

an equivalent ideal gas characterized by the following mean specific heat (ASHRAE Toolkit, 1999):

$$c_{pe} = \frac{\Delta h}{\Delta T_{wb}} \quad 4.2$$

The liquid side conductance is much greater than the gas side conductance. Therefore, the wetted-surface temperature is also assumed to be equal to the water temperature. On the basis of these assumptions, the expression of the total heat transfer becomes:

$$dQ_{w-air} = U_e dA (T_w - T_{wb}) \quad 4.3$$

where

$$U_e = \frac{U * c_{pe}}{c_p} \quad 4.3 (a)$$

An energy balance on the water and air sides gives the following:

$$dQ_{w-air} = m_w * c_{pw} * dT_w \quad 4.4$$

$$dQ_{w-air} = m_a * c_{pe} * dT_{wb} \quad 4.5$$

The water-side effectiveness of the cooling tower can be defined by analogy to the effectiveness of a simple heat exchanger:

$$\varepsilon = \frac{T_{w,in} - T_{w,out}}{T_{w,in} - T_{wb,out}} \quad 4.6$$

Integrating the combination of Equations (4.3), (4.4) and (4.5) and, combining the result of this integration with Equation (4.6), provides the following expression of the cooling tower effectiveness:

$$\varepsilon = \frac{1 - \exp \left[-NTU \left(C - C_r \right) \right]}{1 - C_r * \exp \left[-NTU \left(C - C_r \right) \right]} \quad 4.7$$

where,

$$C_r = \frac{C_w}{C_e} \quad 4.7(a)$$

$$C_w = \dot{m}_w * c_{pw} \quad 4.7(b)$$

$$C_e = \dot{m}_a * c_{pe} \quad 4.7(c)$$

$$NTU = \frac{UA_e}{C_w} \quad 4.7(d)$$

Equation 4.7 is entirely similar to the expression for the effectiveness of an indirect contact counterflow heat exchanger. The same conclusion is reached if the air-side effectiveness is considered; in this case,

$$C_r = \frac{\dot{m}_a * c_{pa}}{\dot{m}_w * c_{pw}} \quad 4.7(e)$$

and,

$$NTU = \frac{UA_e}{\dot{m}_a * c_{pe}} \quad 4.7(f)$$

Consequently, the cooling tower can be modelled, in the steady-state regime, by an equivalent classical counterflow heat exchanger as depicted in Figure 4.3 (ASHRAE Toolkit, 1999).

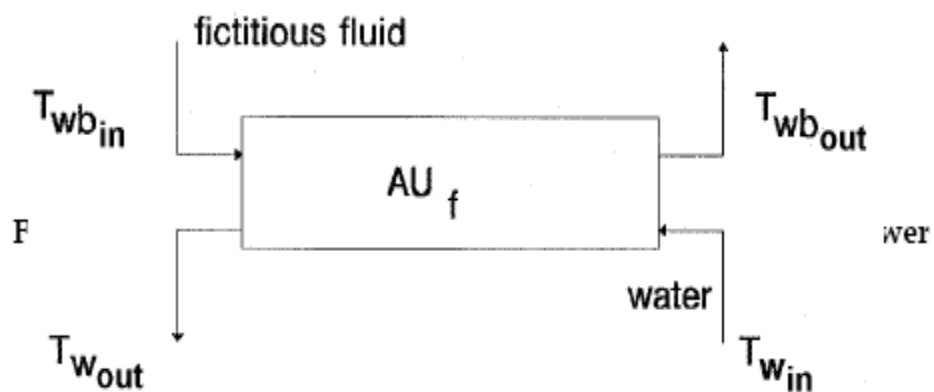


Figure 4.3: Conceptual Schematic of the Counterflow Cooling Tower
(Courtesy ASHRAE Toolkit)

The first fluid is water and the second fluid is an equivalent fluid²⁰ entering the heat exchanger at temperature $T_{wb,in}$ and characterized by the specific heat c_{pe} . The heat exchanger is characterized by a single parameter, its overall heat transfer coefficient-area product UA_e . The actual cooling tower heat transfer coefficient-area product is related to UA_e by the following expression (ASHRAE Toolkit, 1999):

$$UA = UA_e * \frac{c_p}{c_{pe}} \quad 4.8$$

This heat transfer coefficient-area product is assumed to be a function of the air mass flow rate only and is estimated from laboratory test results or manufacturers' catalog data.

The enthalpy variation of this surrogate gas is given by the relationship

$$dh = \left(c_p \right)^* dT_{wb} \quad 4.9$$

where,

$\left(c_p \right)^* =$ Surrogate specific heat

$dT_{wb} =$ Change in wet-bulb temperature

- **Simulation of a Cooling Tower in the Steady-State Regime**

The purpose of the simulation is to determine the leaving water temperature and the leaving air wet-bulb temperature when the cooling tower is operating in the steady-state regime. The information flow diagram is presented in Figure 4.4.

²⁰ In the modelling of heat and mass transfer on wet surfaces (evaporative condensers, direct expansion coil and cooling-towers), humid air is replaced by a surrogate ideal gas whose temperature is the wet bulb temperature.

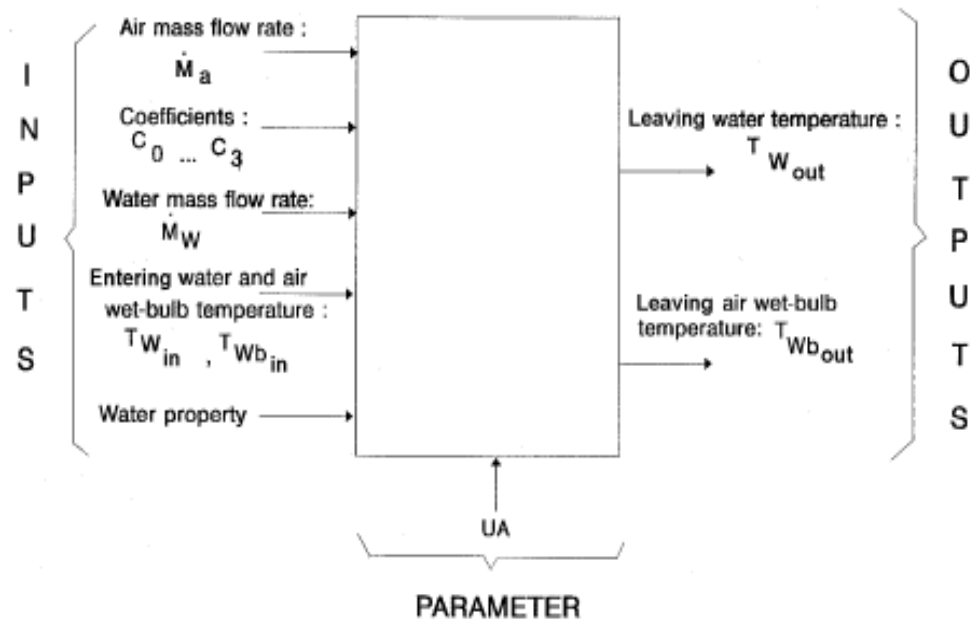


Figure 4.4: Information Flow Diagram for Steady State Cooling Tower Simulation (Courtesy ASHRAE Toolkit)

Input Variables

\dot{m}_w = Water mass flow rate (kg s^{-1})

\dot{m}_a = Air mass flow rate (kg s^{-1})

$T_{w,in}$ = Inlet water temperature (K)

$T_{wb,in}$ = Inlet air wet-bulb temperature (K)

$C_0 \dots C_3$ = Polynomial coefficients for $h = f(T_{wb})$;

These coefficients depend on the value of the atmospheric pressure.

Parameter²¹

UA = Actual heat transfer coefficient-area product (W K^{-1})

Output Variables

$T_{w,out}$ = Exit water temperature (K)

²¹ Parameters are “special inputs” used to characterise (define the nature of) a component.

$T_{wb,out}$ = Exit air wet-bulb temperature (K)

Water Property

c_{pw} = Specific heat of water in the liquid state ($\text{J kg}^{-1}\text{K}^{-1}$)

Selected Internal Variables

c_p = Moist air specific heat ($\text{J kg}^{-1}\text{K}^{-1}$)

h_{in} = Inlet moist air enthalpy (J kg^{-1})

h_{out} = Exit moist air enthalpy (J kg^{-1})

c_{pe} = Equivalent specific heat ($\text{J kg}^{-1}\text{K}^{-1}$)

C_w = Water heat capacity flow rate (W K^{-1})

C_e = Effective fluid heat capacity flow rate (W K^{-1})

C_r = Ratio of the minimum heat capacity flow rate to the maximum heat capacity flow rate

UA_e = Effective heat transfer coefficient-area Product (W K^{-1})

C_{Min} = Minimum heat capacity flow rate (W K^{-1})

C_{rMax} = Maximum heat capacity flow rate (W K^{-1})

NTU = Number of transfer units (-)

Q_{w-air} = Water-air heat transfer rate (W)

ε = Effectiveness of the heat exchanger (-)

The following polynomial equation calculates the saturated air enthalpy as a function of the air temperature (ASHRAE Toolkit, 1999):

$$h_s = C_0 + C_1 t + C_2 t^2 + C_3 t^3 \quad 4.10$$

where,

h_s = saturated air enthalpy

t = air temperature in degree Celsius

C_0, C_1, C_2, C_3 are coefficients whose value depends on the atmospheric pressure.

Since lines of constant wet-bulb temperature on the psychrometric chart closely follows lines of constant enthalpy, the moist air enthalpy can be calculated, with

reasonable accuracy, using the former equation with the temperature set to the air wet-bulb temperature (ASHRAE Toolkit, 1999):

$$h = C_0 + C_1 t_{wb} + C_2 t_{wb}^2 + C_3 t_{wb}^3 \quad 4.11$$

where,

$$t_{wb} = T_{wb} - 273.15$$

The consequence of this approximation is to slightly overestimate the value of the moist air enthalpy.

For the standard atmospheric pressure of 101325 Pa:

$$C_0 = 9362.5 \text{ J kg}^{-1}$$

$$C_1 = 1786.1 \text{ J kg}^{-1} \text{ K}^{-1}$$

$$C_2 = 11.35 \text{ J kg}^{-1} \text{ K}^{-2}$$

$$C_3 = 0.98855 \text{ J kg}^{-1} \text{ K}^{-3}$$

These values are valid for $275.15 \text{ K} \leq T_{wb} \leq 303.15 \text{ K}$.

An iterative process is used to determine the value of $T_{wb,out}$

By using a first guess value²² for the exit wet-bulb temperature, $T_{wb,out}$, the effective specific heat is calculated from:

$$c_{pe} = \frac{h_{out} - h_{in}}{T_{wb,out} - T_{wb,in}} \quad 4.12$$

the effective heat transfer coefficient-area product is deduced:

$$UA_e = UA * \frac{c_{pe}}{c_p} \quad 4.13$$

The effectiveness of the heat exchanger (counterflow cooling tower) is then calculated:

²² The first (initial) guess of the outlet (leaving) wet-bulb temperature of the air is based on estimated difference between the inlet and outlet wet-bulb temperatures.

$$\varepsilon = \frac{1 - \exp(-NTU(1 - C_r))}{1 - C_r * \exp(-NTU(1 - C_r))} \quad 4.14$$

where,

$$C_r = \frac{C_{\min}}{C_{\max}} \quad 4.15$$

$$C_{\min} = \min(C_w, C_e)$$

$$C_{\max} = \max(C_w, C_e)$$

$$C_w = c_{pw} * \dot{m}_w \quad 4.16$$

$$C_e = c_{pe} * \dot{m}_a \quad 4.17$$

$$NTU = \frac{UA_e}{C_{\min}} \quad 4.18$$

The water-air heat transfer rate is described by:

$$Q_{w-air} = \varepsilon * C_{\min} (T_{win} - T_{wb,in}) \quad 4.19$$

The exit air wet-bulb temperature is recalculated as:

$$T_{wb,out} = T_{wb,in} + \frac{Q_{w-air}}{C_e} \quad 4.20$$

The iterative process continues until convergence is reached.

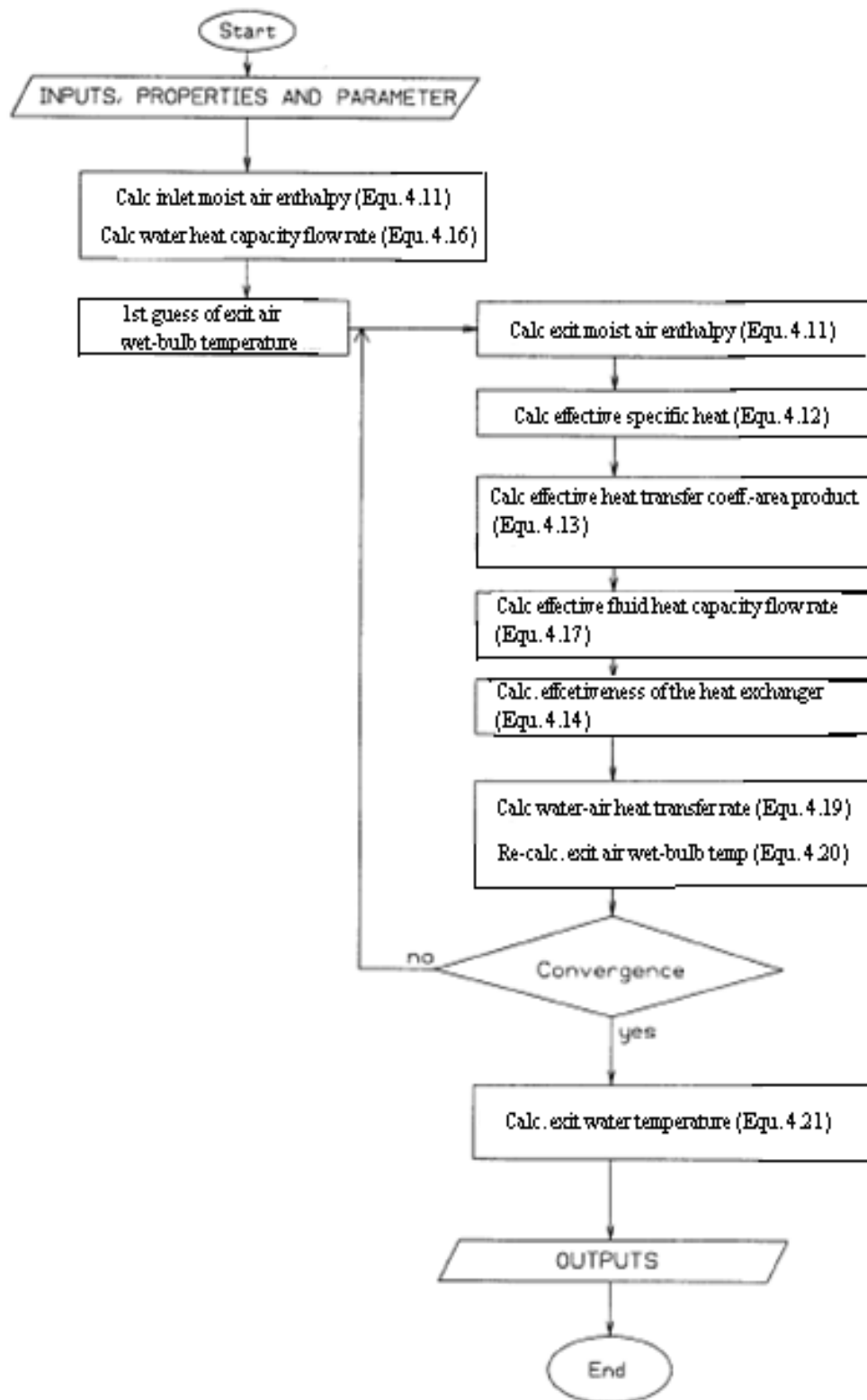
The cooling tower leaving water temperature is described by:

$$T_{w,out} = T_{w,in} - \frac{Q_{w-air}}{C_w} \quad 4.21$$

Note: In this study, the EES model calculation of the cooling tower leaving water temperature was done using the same equation as the TRNSYS model (Equation 4.8) in order to account for water loss due to evaporation. This modification of the “Toolkit” approach gives better accuracy to the model.

ALGORITHM

The flowchart below is a summary of the modelling steps as used in the EES Procedure.



- **Estimation of the Heat Transfer Coefficient of a Cooling Tower**

The purpose of the model is to develop the routine to determine the heat transfer coefficient-area product of the cooling tower associated with a given air mass flow rate on the basis of one operating point in the steady-state regime. The information flow diagram is presented in Figure 4.5.

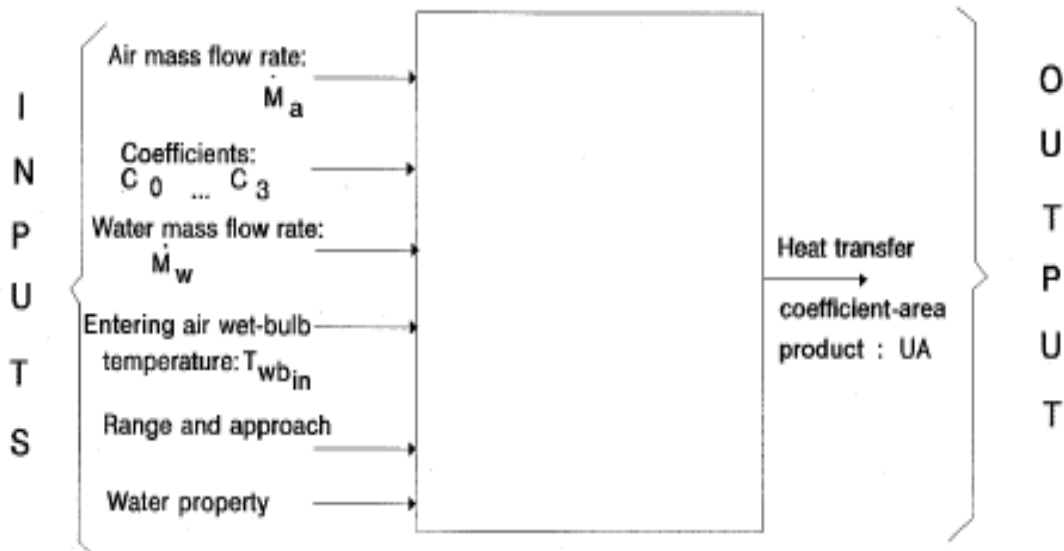


Figure 4.5: Information Flow Diagram for the Heat Transfer Coefficient Model
(Courtesy ASHRAE Toolkit)

Input Variables

\dot{m}_w = Water mass flow rate (kg s^{-1})

\dot{m}_a = Air mass flow rate (kg s^{-1})

Range = Difference between the inlet and exit water temperatures

Approach = Difference between the exit water temperature and the inlet air wet-bulb temperature

$T_{wb,in}$ = Inlet air wet-bulb temperature (K)

$C_0...C_3$ = Polynomial coefficients for $h = f(T_{wb})$;

Output Variables

UA = Actual heat transfer coefficient-area product (W K^{-1})

Water Property

c_{pw} = Specific heat of water in the liquid state ($\text{J kg}^{-1}\text{K}^{-1}$)

Selected Internal Variables

c_p = Moist air specific heat ($\text{J kg}^{-1}\text{K}^{-1}$)

$T_{w,in}$ = Inlet water temperature (K)

$T_{w,out}$ = Exit water temperature (K)

h_{in} = Inlet moist air enthalpy (J kg^{-1})

h_{out} = Exit moist air enthalpy (J kg^{-1})

$T_{wb,out}$ = Exit air wet-bulb temperature (K)

c_{pe} = Equivalent (effective) specific heat ($\text{J kg}^{-1}\text{K}^{-1}$)

C_W = Water heat capacity flow rate (W K^{-1})

C_e = Effective fluid heat capacity flow rate (W K^{-1})

UA_e = Effective heat transfer coefficient-area (W K^{-1})

Product

Q_{w-air} = Water-air heat transfer rate (W)

The inlet and exit water temperatures are easily calculated:

$$T_{w,out} = T_{wb,in} + Approach \quad 4.22$$

$$T_{w,in} = T_{w,out} + Range \quad 4.23$$

Equation 4.11 is used to calculate the inlet moist air enthalpy h_{in} and Equations 4.7 (b) gives the water heat capacity flow rate.

The water-air heat transfer rate is obtained using Equation 4.21.

An iterative process must be used on the value of $T_{wb,out}$ (i.e. on the value of c_{pe}).

By using a first guess, it is possible to calculate the effective specific heat c_{pe} using Equation 4.12 and then the effective fluid heat capacity flow rate, C_e using Equation 4.7 (c).

The exit air wet-bulb temperature can be recalculated by using Equation 4.20.

The iterative process is carried out until convergence is reached.

Assuming a counterflow heat exchanger, the effective heat transfer coefficient-area product is given by:

$$UA_e = \frac{Q_{w-air}}{(T_{w,in} - T_{wb,out}) - (T_{w,out} - T_{wb,in})} \ln \left(\frac{T_{w,in} - T_{wb,out}}{T_{w,out} - T_{wb,in}} \right) \quad 4.24$$

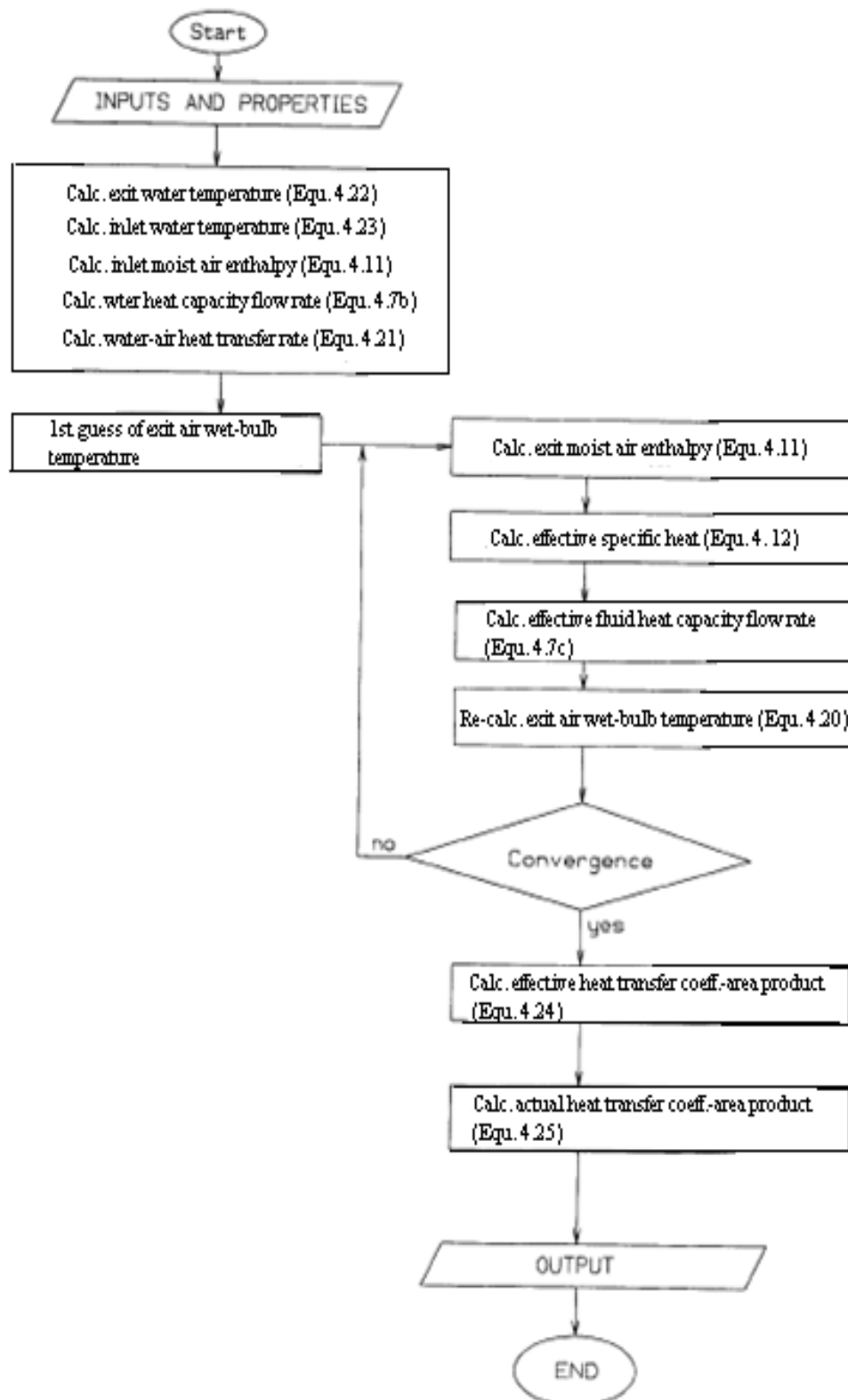
Under these conditions, the actual heat transfer coefficient-area product can be calculated by using the following equation:

$$UA = UA_e * \frac{c_p}{c_{pe}} \quad 4.25$$

ALGORITHM

The flowchart below is a summary of the modelling steps as used in the EES Procedure.

The Cooling Tower EES model is given in Appendix 2.



4.6.1.2 MATHEMATICAL DESCRIPTION OF THE LIQUID DESICCANT AIR CONDITIONING EES MODEL

As previously stated, the EES model of the cooling tower air conditioning unit was developed in this study as no ready-made model was available. The applicable equations used for writing the model routine are presented in Section 3.4 of Chapter 3. The EES “Procedure” was used in writing the program. Below are the mathematical details of the cooling tower air conditioning unit model:

Data:

- Lean TEG concentration = 98.8 wt%
- Density of TEG = $1120 \text{ kg/m}^3 = 1.12 \text{ g/cm}^3 = 1.12 \text{ kg/liter}$
- Mass rate of air through absorber (20% of design) = 1018.8 kg/s
- Density of air = 1.11 kg/m^3
- Volume rate of air = $917.8 \text{ m}^3/\text{s}$
- Molecular weight of air = 28.97 kg/kmol
- Universal gas constant = 8314 J/kmol K
- Density of air at standard condition of $15 \text{ }^\circ\text{C}$ and 101325 Pa (calculated using equation of state for ideal gas) = 1.225 kg/m^3
- Volume rate of air at standard condition $15 \text{ }^\circ\text{C}$ and 101325 Pa (calculated using equation of state for ideal gas), $\dot{v}_{std} = 831.5 \text{ std m}^3/\text{s} = 2993333 \text{ std m}^3/\text{h} = 71.8 \times 10^6 \text{ std m}^3/\text{d} = 71.8 \text{ MSm}^3/\text{d}$
- Air pressure (site ambient) = $1.006 \text{ bara} (100.6 \text{ kPa})$
- Inlet air temperature (dry bulb) = 30°C
- Wet bulb temperature of inlet air (WBT) = 28°C
- Relative humidity of absorber inlet air (RH) = 86%
- Molecular weight of water vapour = 18.02 kg/kmol
- Moisture (water vapour) content of absorber inlet air saturated at given temperature and pressure (from psychrometric chart), $w_{in} = 0.025878 \text{ kg/ m}^3 (0.02314 \text{ kg/kg}) = 25878 \text{ kg}/10^6 \text{ std m}^3$
- Specified (required) relative humidity of absorber outlet air = 10%

- Moisture content of absorber outlet air at given temperature, pressure & specified relative humidity of 10% (from psychrometric chart), $w_{out} = 0.00301 \text{ kg/m}^3$ (0.002712 kg/kg) = $3010 \text{ kg}/10^6 \text{ std m}^3$
- Actual number of trays in absorber = 12 (3 theoretical tray)
- Tray efficiency = 25%

Calculation

From Equation 3.26, the mol fraction water in the lean TEG entering the absorber, $x_o = 0.092$

From Figure 3.10, the activity coefficient for water concentration in lean TEG at 30°C and 98.8 wt% TEG, $\gamma = 0.66$

w_{in} is water content (mass of water) of absorber inlet air saturated at given temperature and pressure (30°C and 100.6 kPa) or $25878 \text{ kg}/10^6 \text{ std m}^3$ as given above.

From Equation 3.25, the water content of gas in equilibrium with incoming lean TEG:

$$w_0 = w_{in} * x_o * \gamma$$

4.26

$$=(25878) (0.63) (0.092) = 1498.5 \text{ kg}/10^6 \text{ std m}^3$$

From Equation 3.23, the absorber efficiency,

$$E_a = (25878 - 3010) / (25878 - 1498.5) = 0.938$$

From Figure 3.8, for theoretical tray, $N = 3$ and $E_a = 0.938$, the absorption factor, $A = 2$

The rate of lean TEG entering the top tray is given by:

$$l = (A) (k) (v_n) \tag{4.27}$$

From Equation 3.24, the equilibrium constant for water in a TEG-water system, $k = [(25878) (0.63)] / 761\,000 = 0.021423$

The given volume air flow rate entering the bottom tray of the absorber (in $\text{std m}^3/\text{h}$) is converted to molar flow rate (in kmol/h at standard P & T) using the ideal gas equation, hence,

$$\dot{v}_n = \frac{\text{std pressure} \times \text{std volume flowrate}}{\text{universal gas constant} \times \text{std temperature}} \quad 4.28$$

$$= \frac{101325 \times 2993333}{8314 \times 288.15} = 126603 \text{ kmol/h}$$

Hence, $l = (2) (0.021423) (126603) = 5424.4 \text{ kmol/h}$

From equation 3.27, the molecular weight of the lean TEG is

$$MW = (0.092) (18) + (150) (1 - 0.092) = 137.87 \text{ kg/kmol}$$

The equivalent mass flow rate of lean TEG entering absorber per hour is

$$m_{TEG} = \frac{l}{MW} \quad 4.29$$

$$= (5424.4) (137.87) = 747862 \text{ kg TEG/h}$$

Density of TEG is 1.12 kg/liter, hence

$$\text{TEG circulation volume rate, } q_{TEG} = \frac{m_{TEG}}{\rho_{TEG}} \quad 4.30$$

$$= 747862 / 1.12 = 667734 \text{ liter/h}$$

Mass of water absorbed from the air flowing through the absorber per hour

$$w = (w_{in} - w_{out}) \dot{v}_{std} \times 24 \quad 4.31$$

$$= (0.025878 - 0.00301) (2993333) = 68452 \text{ kg/h}$$

Hence, TEG circulation rate required to absorb one kg of water (circulation ratio):

$$CR = \frac{q_{TEG}}{w} \quad 4.32$$

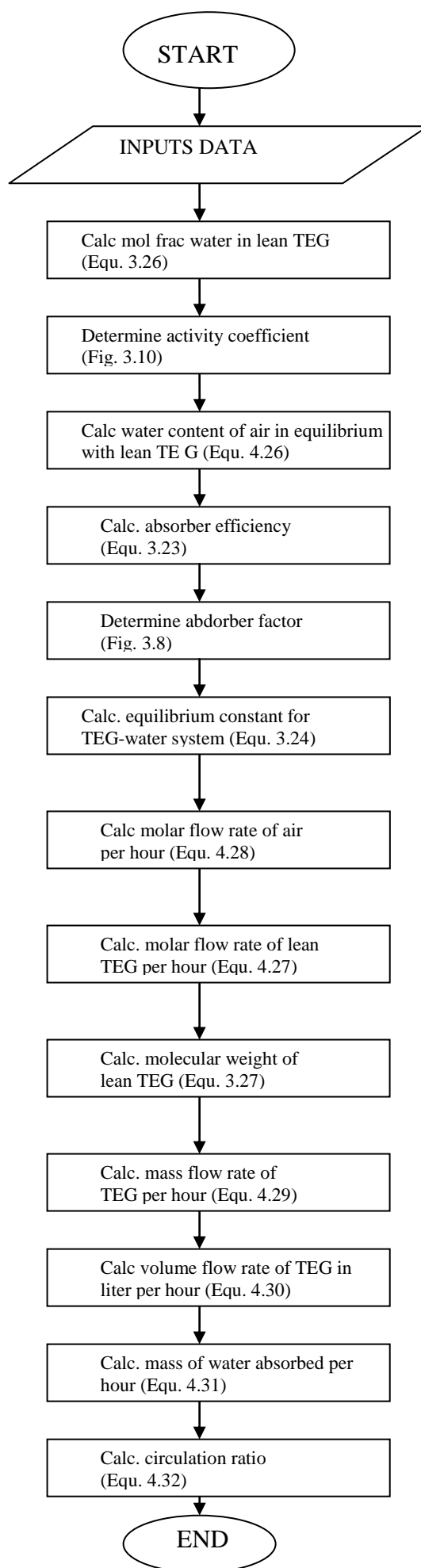
$$= 667734 / 68452 = 9.8 \text{ liter TEG per kg water absorbed}$$

ALGORITHM

The flowchart below is a summary of the modelling steps as used in the EES Procedure.

A snapshot of the calculation spreadsheet for the air conditioning unit is presented in Appendix 5.

The cooling tower air conditioning unit EES model is presented in Appendix 2.



4.6.1.3 MATHEMATICAL DESCRIPTION OF THE GAS LIQUEFACTION PLANT ESS MODEL

Like the air conditioning model, the NLNG gas liquefaction plant model was developed in this study. The plant was modelled using a combination of in-built mathematical and thermo-physical EES property functions and self-written conventional vapour-pressure refrigeration system equations (Khurmi, 2004). The model was written with the EES **Equation Window**. Design data as well as real-time measured data from NLNG plant were used as inputs into the model.

Since the Equation Window was used to build the model, there is no defined Algorithm in the solving of the model. In the EES Equation Window, the order that the equations are written has no effect on the solution as EES will **block** the equations and reorder them for efficient solution (convergence).

The **Residuals Window** in the model indicates the equation blocking and calculation order used by EES, and the status of the unit checking, in addition to the relative and absolute residual²³ values.

The relative residuals are monitored during iterative calculations to determine when the equations have been solved to the accuracy specified with the Stopping Criteria command.

Variables having values that can be determined directly, i.e., without simultaneously finding the values of other variables are determined first and assigned to a Block 0. The order in which the individual equations are solved in Block 0 is indicated by the order in which they appear in the Residuals window. After solving all equations in Block 0, EES simultaneously solves the equations in Block 1, then Block 2, and so on until all equations are solved. The variable(s) that are determined by the equation(s) in each block are shown in bold font.

The Gas Liquefaction system EES model is presented in Appendix 2.

EES Model Limitation

²³ The absolute residual of an equation is the difference between the values on the left and right hand sides of the equation. The relative residual is the magnitude of the absolute residual divided by the value of left side of the equation.

The major limitation of the EES model developed in this study is its inability to perform transient or dynamic simulation. The EES model is a static model (do not vary with time). This means that all outputs depend only on inputs for the time considered. Annual and monthly averaged data from the one year real-time measured site dataset were used as inputs to the cooling tower model.

4.6.2 TRNSYS

TRNSYS is a component based simulation program and is often used to validate new energy concepts, from simple domestic hot water systems to the design and simulation of buildings, plants and their equipment, including control strategy, alternative energy systems (wind, solar, photovoltaic, hydrogen systems), etc. TRNSYS applications include:

- Solar systems (solar thermal and PV)
- Low energy buildings and HVAC systems with advanced design features
- Renewable energy systems
- Cogeneration, fuel cells
- Anything that requires dynamic simulation of thermal systems

(TRNSYS, 2007).

As previously stated, TRNSYS was used for calibration / validation of the EES cooling tower model in this study. To enable the modelling of the cooling tower performance and subsequent calibration / validation of the EES model, the following TRNSYS components were used:

- The Cooling Tower component
- The Psychrometrics Data component
- The Data Reader component

These components are described in some more details in the following sub-sections.

4.6.2.1 MATHEMATICAL DESCRIPTION OF COOLING

TOWER TRNSYS MODEL

The cooling tower component models the performance of a multi-cell counterflow or crossflow cooling tower. There are two primary modes for this model. In mode 1, the

user enters the coefficients of the mass transfer correlation, c and n^{24} . In mode 2, the user enters overall performance data for the cooling tower and the model determines the parameters c and n that provides a best fit to the data in a least-squares sense. Values for c and n are output and can be used in subsequent simulations instead of the program recalculating them.

For this study, mode 1 was used for the steady state and real-time dynamic (actual site conditions) modelling of the cooling tower.

The TRNSYS cooling tower model is based on the effectiveness model developed by Braun (1989). Figure 4.6 shows a schematic of the cooling tower as represented in TRNSYS while Figure 4.7 shows a hypothetical process on a psychrometric chart for the air states through the cooling tower.

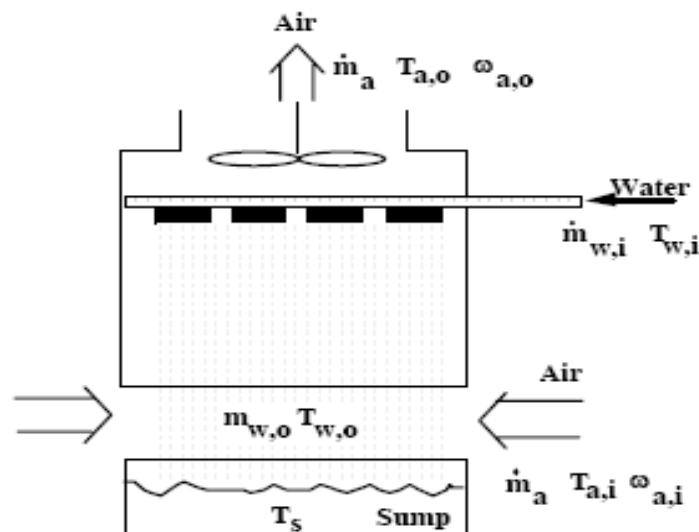


Figure 4.6: TRNSYS Schematic of a Single Cell CounterFlow Cooling Tower (Courtesy TRNSYS 16)

²⁴ The mass transfer coefficients are empirical constants specific to a particular tower design. They are determined by experimental test for a given cooling tower. They depend on the packing design and vary with changes in air and water flow rates.

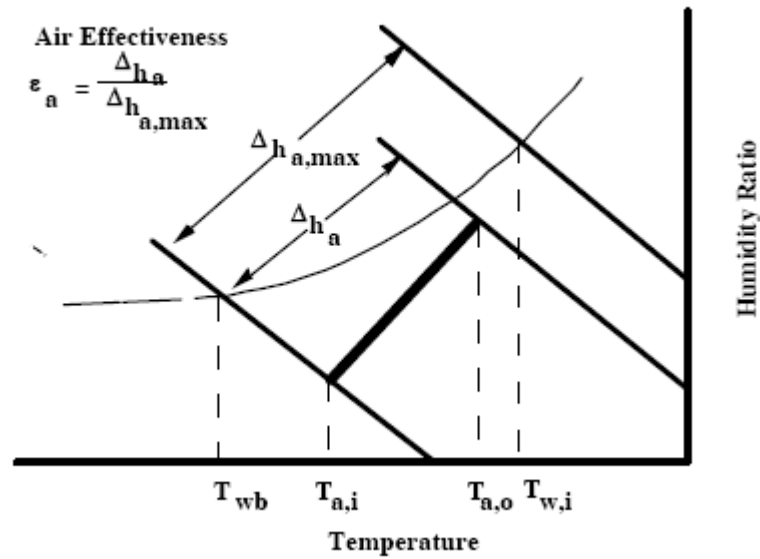


Figure 4.7: Psychrometric Chart for the Air States Through Cooling Tower
(Courtesy TRNSYS 16)

The air enters at a known state, characterized by a temperature, $T_{a,i}$ and humidity ratio, $\omega_{a,i}$ and exits at $T_{a,o}$ and $\omega_{a,o}$. The limiting exit state for the air would be if the air were saturated at a temperature equal to that of the incoming water stream. This corresponds to the maximum possible enthalpy of the exit air. Lines of constant enthalpy are shown for the air inlet and outlet and the saturated state corresponding to the water inlet. In terms of these enthalpies, the air-side heat transfer effectiveness is defined as the ratio of the air enthalpy difference to the maximum possible air enthalpy difference. For a known effectiveness, the heat rejection for an individual tower cell is then

$$\dot{Q}_{cell} = \epsilon_a \dot{m}_a (h_{s,w,i} - h_{a,i}) \quad 4.33$$

where

ϵ_a is the air-side effectiveness; \dot{m}_a is the mass flow rate of dry air; $h_{s,w,i}$ and $h_{a,i}$ are the enthalpies of the water vapour boundary film and the dry air, respectively, at tower inlet.

Using the assumption that the Lewis number equals one, Braun showed that the air effectiveness can be determined using the relationships for sensible heat exchangers

with modified definitions for the number of transfer units, Ntu , and the heat capacitance rate ratios m^* . For a counterflow cooling tower,

$$\varepsilon_a = \frac{1 - \exp\left[-Ntu \left(\frac{1 - m^*}{m^*}\right)\right]}{1 - m^* \exp\left[-Ntu \left(\frac{1 - m^*}{m^*}\right)\right]} \quad 4.34$$

with

$$Ntu = \frac{h_m A_v V_{cell}}{m_a} \quad 4.35$$

and

$$m^* = \frac{m_a C_s}{m_{w,i} C_{pw}} \quad 4.36$$

where,

h_m is the cooling tower mass transfer coefficient; A_v is the surface area of water droplet per tower cell exchange volume; V_{cell} is the total tower cell exchange volume; $m_{w,i}$ is mass flow rate of water at tower inlet and C_{pw} is the constant pressure specific heat of water.

The saturation specific heat, C_s , is defined as the average slope of the saturation enthalpy with respect to temperature curve. It is determined with the water inlet and outlet conditions and psychrometric data using:

$$C_s = \frac{h_{s,w,i} - h_{s,w,o}}{T_{w,i} - T_{w,o}} \quad 4.37$$

Tower Cell Performance Data

In order to determine tower effectiveness, it is necessary to have a relationship for the number of transfer units. General correlations for heat and mass transfer in cooling towers in terms of the physical tower characteristics are not readily available.

As presented in the ASHRAE Equipment Guide (ASHRAE, 1996), mass transfer data is generally correlated in the form

$$\frac{h_m A_V V_{cell}}{m_w} = c \left(\frac{\dot{m}_w}{\dot{m}_a} \right)^n \quad 4.38$$

Multiplying both sides of the above equation by $\frac{\dot{m}_w}{\dot{m}_a}$ and utilizing the definition for

Ntu gives

$$Ntu = c \left(\frac{\dot{m}_w}{\dot{m}_a} \right)^{1+n} \quad 4.39$$

The exponent n is typically between -0.35 and -1.1, while c may be in the range of 0.5 to 5. Simpson and Sherwood (1946) give data for a number of different tower designs. It is usually necessary to correlate specific data in order to determine the parameters for a particular tower.

Tower Cell Exit Conditions

From an overall energy balance, the outlet temperature from a tower cell is determined as (Braun, 1989)

$$T_{w,o} = \frac{\dot{m}_{w,i} C_{pw} (T_{w,i} - T_{ref}) - Q_{cell}}{\dot{m}_{w,o} C_{pw}} + T_{ref} \quad 4.40$$

where

T_{ref} = water reference temperature = 0°C

Note: Most analyses neglect the water loss and assume that $\dot{m}_{w,o} = \dot{m}_{w,i}$. Generally, the water loss rate is in the order of 1% to 4% of the entering water flow rate. Neglecting this loss can result in about a 1 degree Celsius error in the exit water temperature. It is also necessary to know the water loss for analyzing the performance of the cooling tower sump, in order to incorporate the effects of water makeup. The TRNSYS component model includes the effect of the water loss.

From an overall mass balance, the exit water flow rate is

$$\dot{m}_{w,o} = \dot{m}_{w,i} - \dot{m}_a (\omega_{a,o} - \omega_{a,i}) \quad 4.41$$

The exit humidity ratio is determined from an analytic solution to an equation for the mass transfer assuming an effective water surface condition and a Lewis Number of unity (Braun, 1989).

$$\omega_{a,o} = \omega_{s,w,e} + (\omega_{a,i} - \omega_{s,w,e}) \exp(-Ntu) \quad 4.42$$

The effective saturation humidity ratio, $\omega_{s,w,e}$, is found with psychrometric data using an effective saturation enthalpy computed from the solution to the heat transfer equation

$$h_{s,w,e} = h_{a,i} + \frac{h_{a,o} - h_{a,i}}{1 - \exp(-Ntu)} \quad 4.43$$

where the enthalpy of dry air at the cooling tower outlet is given by

$$h_{a,o} = h_{a,i} + \varepsilon_a (\omega_{s,w,i} - \omega_{a,i}) \quad 4.44$$

To determine the air conditions exiting the tower, the exit air from each of the cells are combined to find "bulk" air conditions. The total exiting air flow rate is the sum of the flow rates exiting the cells,

$$m_{a,exit} = \sum_{k=1}^{N_{cell}} m_{a,k} \quad 4.45$$

The exiting enthalpy is found by summing the energy flow rates exiting the cells and dividing by the total air flow rate,

$$h_{a,exit} = \frac{\sum_{k=1}^{N_{cell}} (m_a h_{a,o})_k}{m_{a,exit}} \quad 4.46$$

From a mass balance on the moist air, an overall exit air humidity ratio can be calculated,

$$\omega_{a,exit} = \frac{\sum (m_{w,i} - m_{w,o})}{m_{a,exit}} + \omega_{a,i} \quad 4.47$$

Using $h_{a,exit}$ and $\omega_{a,exit}$, psychrometrics data is used to find the bulk dry bulb temperature, $T_{a,exit}$, and the bulk wet bulb temperature, $T_{wb,exit}$.

4.6.2.2 THE TRNSYS PSYCHROMETRICS DATA COMPONENT

The psychrometrics data component in TRNSYS is identified as “Type 33”. It is a flexible component that can be used to calculate different properties of humid air from two independent variables. This component is merely a wrapper for a call to the TRNSYS Psychrometrics Utility Subroutine. It takes as Input:

- (1) dry bulb temperature and one other property, depending on the mode: wet bulb temperature, relative humidity, dew point temperature, humidity ratio, or enthalpy; or
- (2) humidity ratio and enthalpy.

It calls the Psychrometrics Subroutine to calculate humidity ratio, wet bulb temperature, enthalpy, density of the air-water mixture, density of dry air only, relative humidity (as percentage), dry bulb temperature and dew point temperature.

For the modelling done in this study, the real-time dry-bulb temperature and relative humidity (from site) were the inputs to the psychrometrics component.

4.6.2.3 THE TRNSYS DATA READER COMPONENT

This component serves the purpose of reading data at regular time intervals from a logical unit number, converting it to a desired system of units, and making it available to other TRNSYS Units as time varying forcing functions. The component used in the modelling for this study is “Type 9”.

The simulation start time is the moment in time at which the simulation is to begin. Other timesteps are still referred to using the time at the end of the time step.

Each variable is associated to four parameters that instruct TRNSYS whether or not to interpolate the data, whether the column in question contains instantaneous or interpolated data, and requires a simple unit conversion operation.

Each variable read by Type 9 can be an instantaneous or an average value over the data time step. TRNSYS always expects average values from Type 9 so instantaneous values are converted.

Each variable can be interpolated or not, whether or not it is provided as an instantaneous or an average value.

Appendix 2 shows the TRNSYS models of the cooling tower system.

4.7 MODEL VERIFICATION

An EES model of the NLNG Bonny Island plant cooling tower and proposed air conditioning system has been developed. This model will be used to assess the potential improvements in cooling tower performance due to the air conditioning system.

In order to have confidence in the predictions of this model, comprehensive verification process has been undertaken. This makes use of both empirical validation and inter-model comparison.

The first step in the verification process was to ensure that the predictions of the EES cooling tower model was in agreement with the manufacturer's design data (conditions) for the NLNG site cooling tower.

A TRNSYS cooling tower model, which involved use of measured time series boundary conditions (Bonny Island climatic conditions) data, was also developed which was used to verify the predictions of the EES model. This TRNSYS was also verified against static manufacturer's data.

The results of the verification of the EES model predictions against manufacturer's data are shown in Table 4.3. These results shows very close match of the EES cooling tower model, as well as the TRNSYS model cooling tower mode, to the manufacturer's data.

Table 4.3: Steady State Models Input and Output Parameters

Input Parameter	Manufacturer's Design Data	TYNSYS Model	EES Model	Output Parameter	Manufacturer's Design Data	TRNSYS Model	EES Model
<i>Water inlet temp (°C)</i>	40.1	40.1	40.1	<i>Water outlet temp (°C)</i>	32	32.2	32.2
<i>Inlet water flow rate (kg/s)</i>	7777.778	7777.778	7777.778	<i>Cooling tower duty (kW)</i>	266000	269722	268975
<i>Dry-bulb temp (°C)</i>	30.0	30.0	30.0	<i>Air density (kg/m³)</i>	1.06	1.11	1.11
<i>Wet-bulb temp (°C)</i>	28.0	28.0	28.0	<i>Water loss (kg/s)</i>	90	90	90
<i>Max total air flow rate (kg/s)</i>	4537	4537	4537				
<i>Fan power at max air flow rate (kW)</i>	3040	3040	3040				
<i>Mass transfer constant</i>	-0.7	-0.7	-0.7				
<i>Mass transfer constant</i>	2.4	2.4	N/A	<i>Mass transfer constant</i>	N/A	N/A	2.46

4.7.1 Verification of Measured Time Series Data

As currently configured, the EES cooling tower model can only be used for time averaged analysis of the cooling tower performance. It was therefore necessary to verify that this form of model gave results that were consistent with the real-time operation of the NLNG plant. This was done by comparing the time averaged results of the EES cooling tower model against the results of the TRNSYS model, which could be simulated using measured time series data from the NLNG plant. For use with the EES model, this time series data was converted to monthly average format. The results of the comparison carried out are shown in Table 4.4.

The results show that the EES cooling tower model is capable of producing performance data which accurately reflects that of the NLNG Bonny Island plant.

Table 4.4: Time Series Model Input and Output Parameters

Input Parameter	Measured NLNG Site Data	TYNSYS Model	EES Model	Output Parameter	Measured NLNG Site Data	TRNSYS Model	EES Model
<i>Water inlet temp (°C)</i>	40.9	40.9	40.9	<i>Water outlet temp (°C)</i>	33	32.2	32.19
<i>Inlet water flow rate (kg/s)</i>	8139	8139	8139	<i>Cooling tower duty (kW)</i>	269409	308333	311000
<i>Dry-bulb temp (°C)</i>	30.1	30.1	30.1	<i>Air density (kg/m³)</i>	N/A	1.11	1.11
<i>Relative humidity (°C)</i>	86.2	86.2	86.2	<i>Water loss (kg/s)</i>	95.8	101.9	104.1
<i>Max total air flow rate (kg/s)</i>	5345	5345	5345	<i>Wet-bulb temp (°C)</i>	28.12	28.1	28.12
<i>Fan power at max air flow rate (kW)</i>	3040	3040	3040				
<i>Mass transfer constant</i>	-0.7	-0.7	-0.7				
<i>Mass transfer constant</i>	N/A	2.268	N/A	<i>Mass transfer constant</i>	N/A	N/A	2.268

Note: The results show an average difference of +/- 1 K between the cooling tower cold water temperature from the measure site data and the EES / TRNSYS models. This could be due to the fact that the online temperature transmitter at the site is a single point transmitter located on a 72 inch (1.83 m) cold water discharge pipeline downstream the water circulation pump. Hence the temperature transmitter reading is unlikely to be representative of the “homogenous” temperature due to temperature difference between the fluid (water) in contact with the pipe wall (where the temperature transmitter is located) and the temperature of the “free stream” fluid. An attempt was made to calculate the temperature rise due to the 2.2 MW water circulating pump. However, the 0.016 K temperature rise calculated was considered

negligible. No attempt was made to calculate the effect of the temperature boundary layer of the discharge piping.

Figure 4.8 shows a plot of the actual and TRNSYS model cold (outlet) water temperature of the 9-cell (actual condition) cooling tower. Also included in the plot is the air density to show the site and model outlet (exit) water temperature difference (error) variation with the air density. This outlet water temperature error variation with air density is further highlighted by Figures 4.9(a) and 4.9(b). It is observed that the minimum error in outlet temperature occurs when the air density is 1.11 kg/m^3 . This value is close to the design air density of the NLNG cooling tower which is 1.06 kg/m^3 .

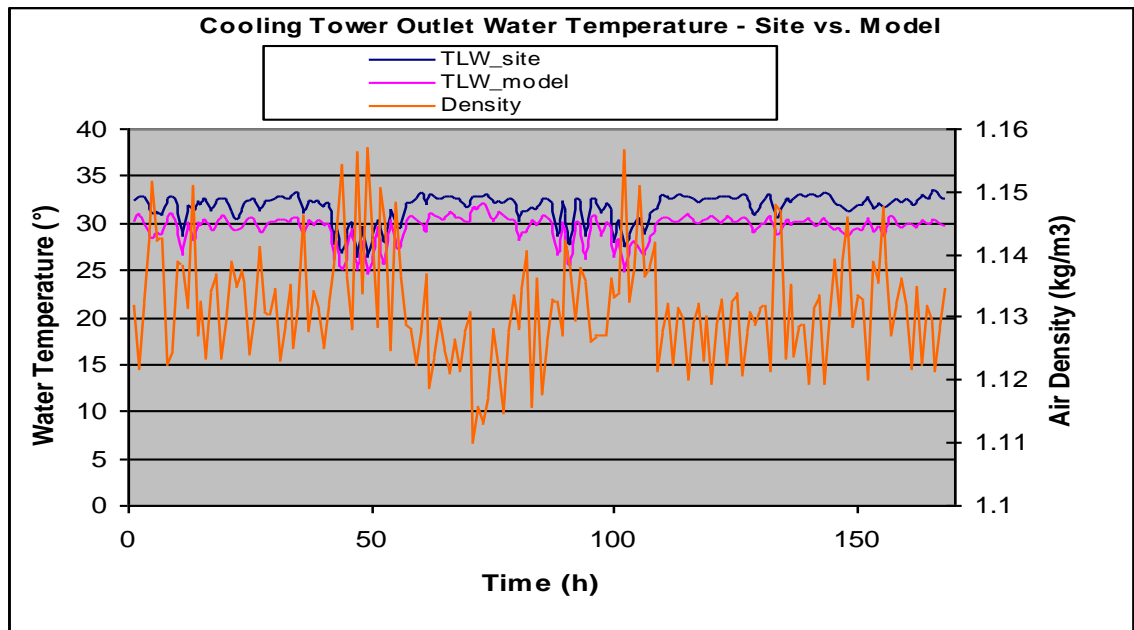


Figure 4.8: Site vs. Model Cold Water Temperature (Real-Time)

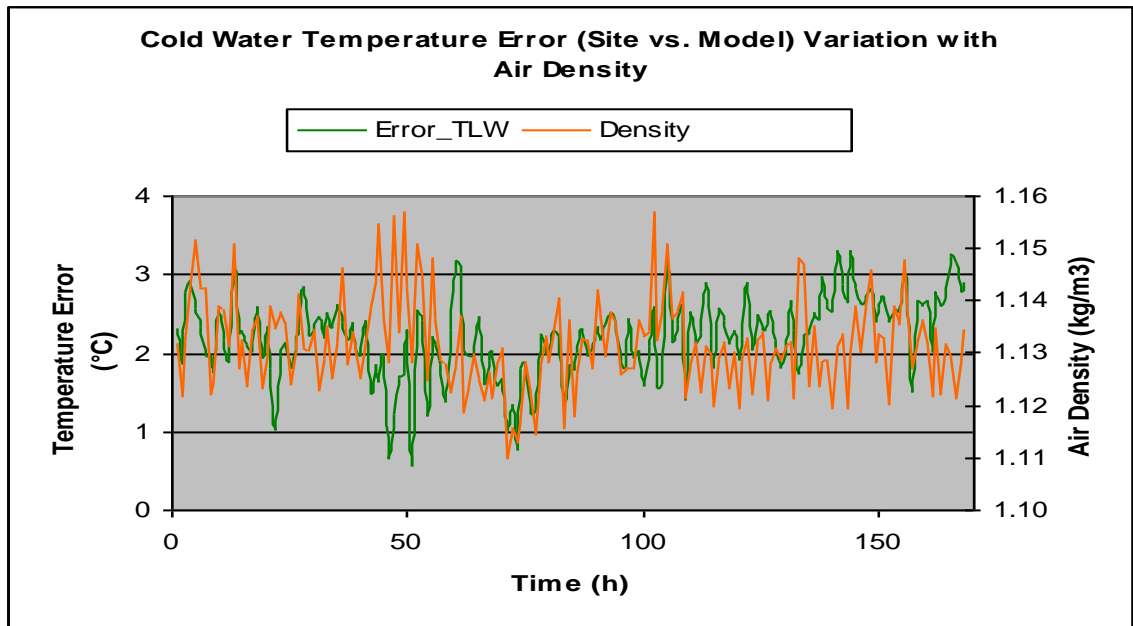


Figure 4.9(a): Site vs. Model Cold Water Temperature Error Variation with Air Density (Real-Time)

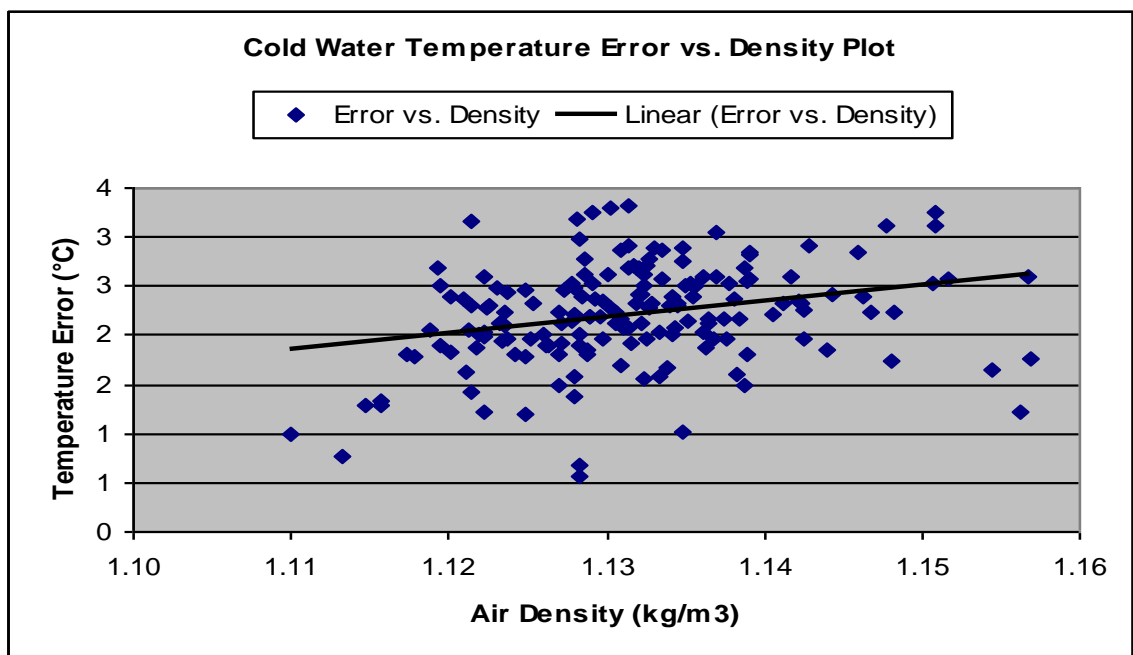


Figure 4.9(b): Cold Water Temperature Error vs. Air Density Plot (Real-Time)

4.8 MODEL ASSUMPTIONS

The following assumptions were made for the 9-cell real-time cooling tower TRNSYS and EES models:

- The air flow rate is assumed to be constant at design of 566 kg/s per cell. The cooling tower fans speed are always set at maximum at the NLNG site, hence this assumption would have negligible effect in the validity of the models. More details are provided in the uncertainty analysis on Section 4.10 below.
- The cooling tower mass transfer exponent was assumed to be constant at design value of -0.7. Usually this constant varies with changes in the water and air flow rates. However, keeping this value constant in the TRNSYS and EES models enable the calculation of the cooling tower mass transfer constant (coefficient) which also varies with the water and air flow rates. This assumption would have negligible effect in the validity of the models.
- It was assumed that the exhaust air from the cooling tower is saturated.

The absorption cycle consists of mass and heat transfer balances in the absorber, the regenerator and the interchange exchanger

The following basic assumptions were made to simplify the sizing and modelling of the air conditioning system:

- The TEG absorber is essentially isothermal (refer to Section 3.4.1). Also, heat losses and heat gains between the absorber and its environment (surroundings) are neglected.
- Changes in potential and kinetic energy between the inlet and outlet of each component are negligible.
- There are no friction or pressure losses in pipes and components.
- A lean TEG concentration of 98.8 wt% will be used for calculations (Campbell, 2004).
- A 12-tray absorber (3-theoretical tray) is used as the initial design selection.
- A tray efficiency of 25% is used (Campbell, 2004).
- Regeneration temperature is 204°C (Campbell, 2004).

Regeneration pressure is atmospheric - 760 mmHg (Campbell, 2004).

4.9 UNCERTAINTY ANALYSIS

The uncertainty or margin of error of a measurement is stated by giving a range of values which are likely to enclose the true value. Uncertainty depends on both the accuracy and precision of the measurement instrument. The lower the accuracy and precision of an instrument, the larger the measurement uncertainty is. This uncertainty imposes a limit on the confidence in the response or output of a model. There are two common ways to statistically describe uncertainty in measurements. One is with the standard deviation of a single measurement and the other is with the standard deviation of the mean, often called the standard error. Often, the uncertainty of a measurement is found by repeating the measurement enough times to get a good estimate of the standard deviation of the values. Then, a single value has an uncertainty equal to the standard deviation. However, if the values are averaged, then the mean measurement value has a much smaller uncertainty, equal to the standard error of the mean, which is the standard deviation divided by the square root of the number of measurements (Taylor and Kuyatt, 1994).

The uncertainty of the following parameters has been determined due to their impact (directly and/or indirectly) on the cold (exit) water temperature from the cooling tower, which is the primary variable of interest for this research study:

1. the site measured air temperature
2. the calculated ambient air density
3. the site measured relative humidity of the cooling tower inlet air
4. the calculated wet-bulb temperature the cooling tower inlet air
5. the air flow rate through the cooling tower

The uncertainty of the air temperature measured from NLNG site affects the accuracy of the value of the calculated air density that was used in the cooling tower EES model. The uncertainty of the air density, on the other hand, affects the mass flow rate of air through the cooling tower.

The uncertainty of the relative humidity of the ambient air measured from site affects the value of the wet-bulb temperature calculated by the model. Finally, the

uncertainty of the air flow rate through the cooling tower and the inlet wet-bulb temperature affects the temperature of the exit water from the cooling tower.

The air temperature, relative humidity and air flow rate were used as inputs in both the TRNSYS and EES cooling tower models. The air density was an input in the EES model and an output in the TRNSYS model. The cooling tower inlet wet-bulb temperature and exit water temperature were outputs in both models.

Using the standard error approach, the uncertainty of the measured variables from site was calculated as;

- air temperature = ± 0.14 °C
- relative humidity of ambient air = ± 1.17 %

The uncertainty propagation command in EES calculates how the uncertainties in each of the measured variables propagate into the value of a given calculated variable. The method for determining this uncertainty propagation is described in NIST (National Institute of Standards and Technology – USA) Technical Note 1297 (Taylor and Kuyatt, 1994). Assuming the individual measurements are uncorrelated and random, the uncertainty, U , in a calculated variable, Y , can be determined as a function of the measured variables, X_1 , X_2 , etc, using the equation:

$$U_y = \sqrt{\sum_i \left(\frac{\partial Y}{\partial X_i} \right)^2 U_{X_i}^2} \quad 4.48$$

After the calculations are completed, EES will display an abbreviated Solution Window containing the calculated and measured variables and their respective uncertainties. The partial derivative of the calculated variable with respect to each measured variable is also shown. In addition, the percentage of the total uncertainty in the calculated variable resulting from the uncertainty in each measured variable is displayed in the solution. This percentage is calculated as a ration of the square of the product of the partial derivative and uncertainty of each measured variable to the square of the uncertainty in the calculated variable (Klein, 2009).

Using the EES, the uncertainty of the following variables was calculated as shown in Table 4.5:

Note: The uncertainty of the air density was calculated as a function of the measured air temperature; the uncertainty of the air flow rate was calculated as a function of the air density; the uncertainty of the inlet air wet-bulb temperature was calculated as a function of the measured relative humidity of ambient air; the uncertainty of the exit water temperature was calculated as a function of the air flow rate and the measured relative humidity of ambient air.

Uncertainty calculation details are shown in the EES model in Appendix 2

Table 4.5: Uncertainty of Calculated Variables

Variable	Uncertainty	Partial derivative
Air density	$\pm 0 \text{ kg/m}^3$	0
Air flow rate	$\pm 0 \text{ kg/s}$	4815
Inlet air wet-bulb temperature	$\pm 0.2 \text{ }^\circ\text{C}$	16.72
Exit water temperature	$\pm 0.1 \text{ }^\circ\text{C}$	0 (with air flow rate) 7.35 (with relative humidity)

4.10 SENSITIVITY ANALYSIS

Sensitivity analysis is the study of how the variation (uncertainty) in the output of a model can be apportioned, qualitatively or quantitatively, to different sources of variation in the input of the model. Sensitivity analysis tries to identify what source of uncertainty weighs more on the model's outputs. Sensitivity analysis looks at the effect that varying the inputs of a model will have on the output of the model. Such an appreciation (understanding) of how the model behaves in response to changes in its inputs is of fundamental importance to ensure a correct use of the model.

A common approach in sensitivity analysis is that of changing one-factor-at-a-time (e.g., changing one input variable at a time) to see what effect this produces on the output. This appears a logical approach as any change observed in the output will unambiguously be due to the single factor changed. Furthermore, by changing one factor at a time all other factor can be kept fixed (constant) to their baseline value. This increases the comparability of the results (Breierova and Choudhari, 2001).

In this study, sensitivity analysis has been carried out on the site-measured input variables that directly affect the inlet air wet-bulb temperature and the temperature of the exit (cold) water temperature from the cooling tower. These variables are the measured ambient air temperature and the measured relative humidity of the ambient air.

The sensitivity of the cooling tower exit temperature to changes in the air temperature and the relative humidity is calculated using the Parametric Table capability of EES. The Parametric Table, similar to a spreadsheet, allows an equation set to be solved repeatedly while varying the values of one or more variables. By identifying the variables that are independent and entering them in the table cells, EES calculates the values of the dependent variables in the table. The relationship of the variables in the table can also be displayed in a plot.

In the sensitivity calculation, the air temperature is varied ± 5 °C about design value of 32 °C (to cover the range of seasonal temperature variation in Nigeria where the NLNG plant is located) while the relative humidity of the air is varied between 10 % (which is the air conditioning unit design specification for this study) and 90 % (which is the the seasonal high in Nigeria). The sensitivity of the inlet air wet-bulb temperature to these variations is shown in Tables 4.6 and 4.7 respectively. The associated sensitivity plots are shown in Figures 4.10 and 4.11 respectively. Similarly, the sensitivity of cooling tower exit temperature to these variations is shown in Tables 4.8 and 4.9 respectively. The associated sensitivity plots are shown in Figures 4.12 and 4.13 respectively.

Table 4.6: Wet-bulb Temperature Variation with Air Temperature

Air Temperature (°C)	Inlet Air Wet-bulb Temperature (°C)
27	23.09
28	24.01
29	24.92
30	25.84
31	26.76
32	27.67
33	28.59
34	29.51
35	30.43
36	31.35

Table 4.7: Wet-bulb Temperature Variation with Relative Humidity

Relative Humidity of Air (%)	Inlet Air Wet-bulb Temperature (°C)
10	14.0
15	15.34
20	16.62
25	17.84
30	19.01
35	20.13
40	21.2
45	22.24
50	23.23
55	24.19
60	25.11
65	26.0
70	26.86
75	27.69
80	28.5
85	29.29
90	30.04

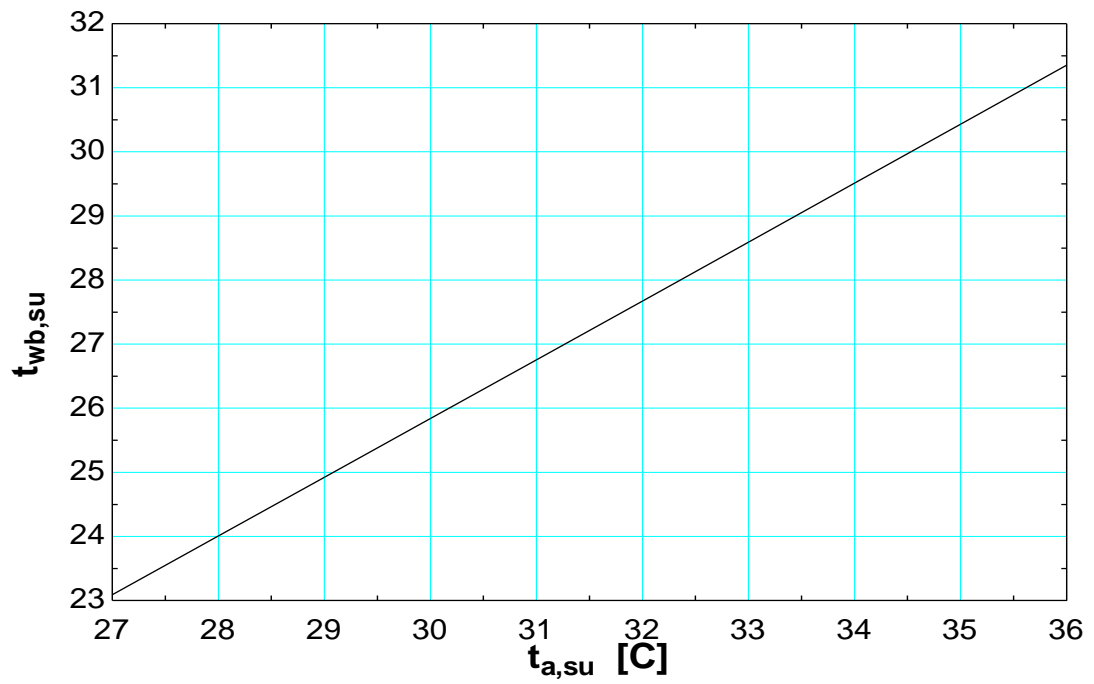


Figure 4.10: Wet-bulb Temperature Variation with Air Temperature

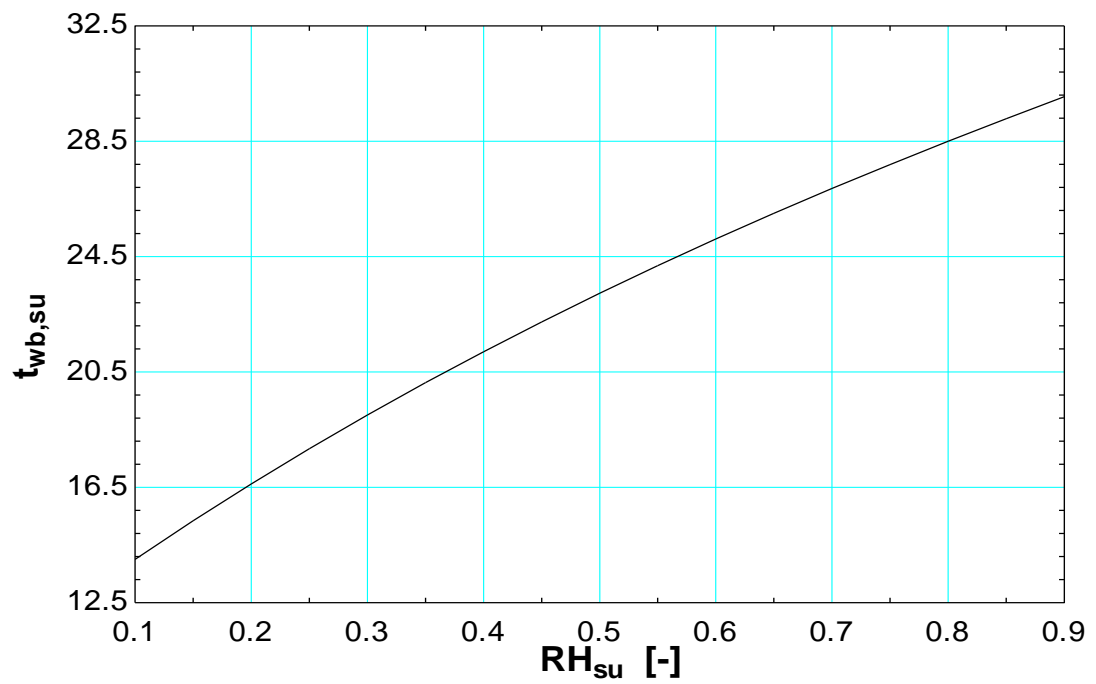


Figure 4.11: Wet-bulb Temperature Variation with Relative Humidity

Table 4.8: Cold Water Temperature Variation with Air Temperature

Air Temperature (°C)	Cold Water Temperature (°C)
27	29.58
28	30.02
29	30.48
30	30.95
31	31.44
32	31.94
33	32.46
34	32.99
35	33.54
36	34.11

Table 4.9: Wet-bulb Temperature Variation with Relative Humidity

Relative Humidity of Air (%)	Cold Water Temperature (°C)
10	26.0
15	26.46
20	26.92
25	27.38
30	27.84
35	28.3
40	28.76
45	29.22
50	29.68
55	30.13
60	30.59
65	31.05
70	31.5
75	31.95
80	32.39
85	32.84
90	33.28

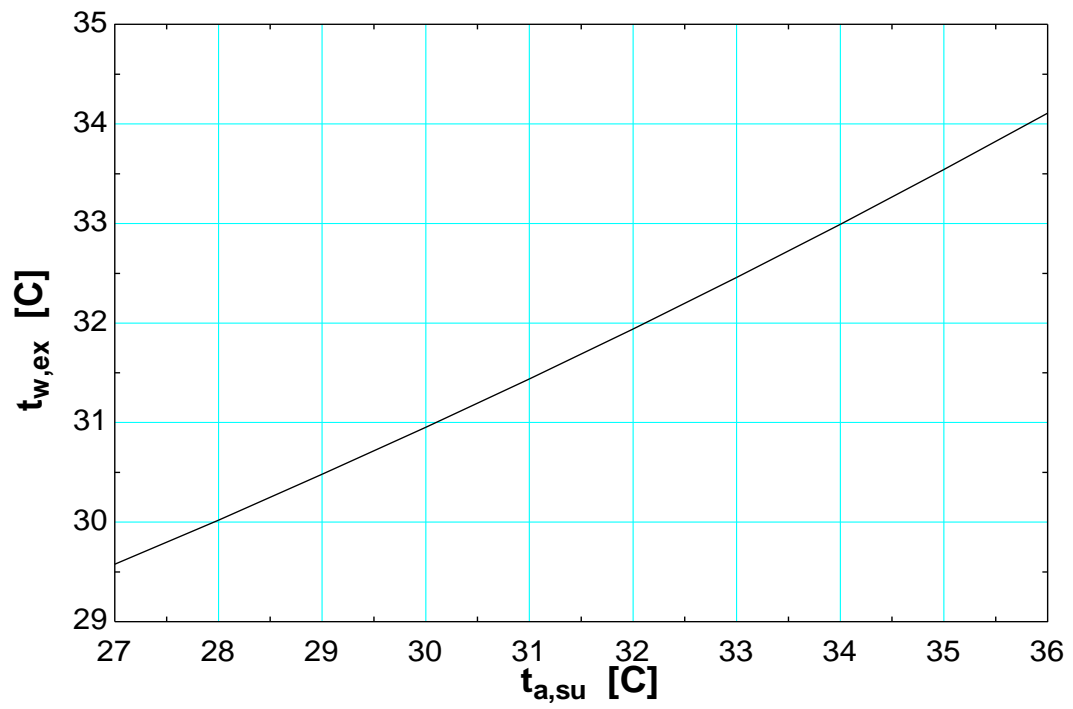


Figure 4.12: Cold Water Temperature Variation with Air Temperature

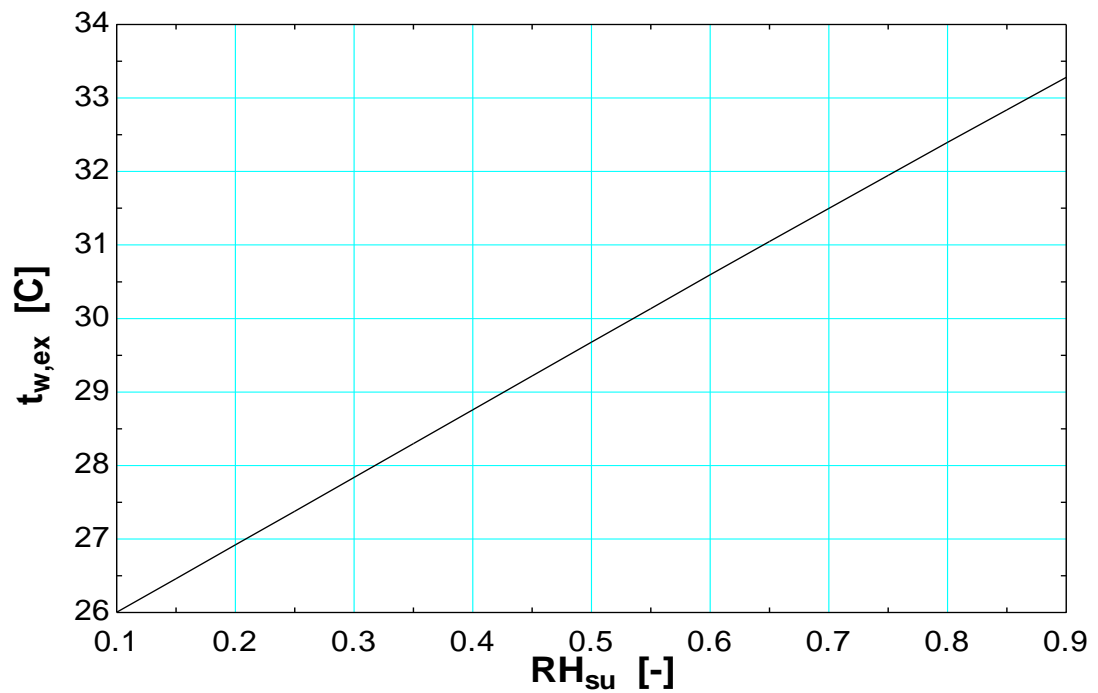


Figure 4.13: Cold Water Temperature Variation with Relative Humidity

In addition to the above, the sensitivity of the coefficient of performance (COP) of the gas liquefaction plant propane refrigerant cycle to the cold water temperature to the condensers (from the cooling tower) was analysed. Table 4.11 and Figure 4.14 shows the results of this analysis.

Figures 4.8(a), 4.8(b) and 4.8(c) presented and discussed earlier in Sub-section 4.6.3.1, shows the sensitivity of the cooling tower exit (cold) water temperature to changes in air density.

Finally, the results of the actual sensitivity analysis carried out by the NLNG plant to determine the sensitivity of liquefied natural gas production to the cold water temperature (from the cooling tower) is shown in Table 4.12 and Figure 4.15.

Table 4.10: COP Variation with Cold Water Temperature

Cold Water Temperature (%)	Compressor COP
28	3.745
28.53	3.742
29.05	3.738
29.58	3.734
30.11	3.729
30.63	3.724
31.16	3.718
31.68	3.712
32.21	3.705
32.74	3.696
33.26	3.687
33.79	3.678
34.32	3.667
34.84	3.655
35.37	3.642
35.89	3.627
36.42	3.612
36.95	3.595
37.47	3.577
38	3.558

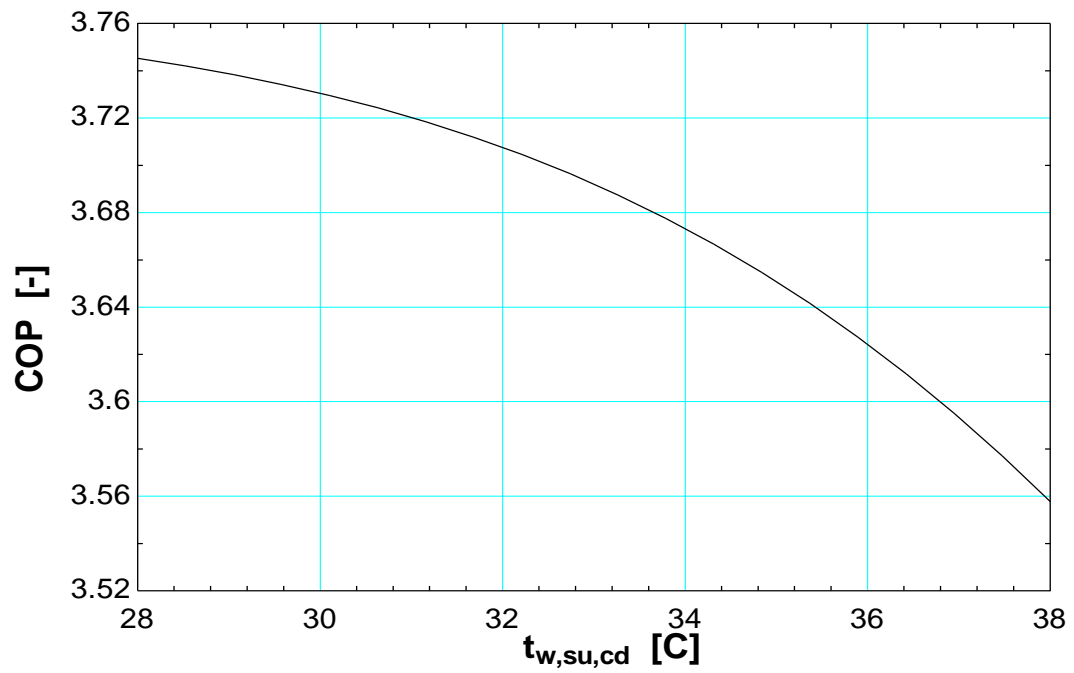


Figure 4.14: COP Variation with Cold Water Temperature

Table 4.11: LNG Production Variation with Cold Water Temperature

Cold Water Temperature (°C)	LNG Production (kg/s)
28	137
29	136.5
30	136
31	135.2
32	135
33	132.5
34	129
35	126

Plant feed and cooling water temperature at constant compositions and conditions

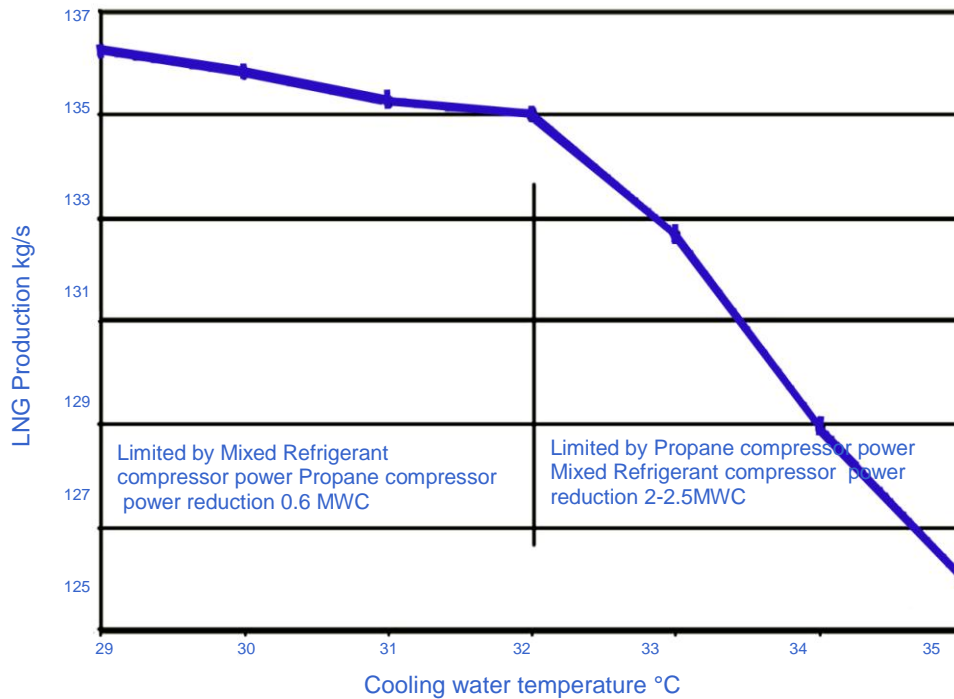


Figure 4.15: LNG Production Variation with Cold Water Temperature
(Courtesy NLNG)

These results from the sensitivity analysis are discussed in Chapter 5.

4.11 SUMMARY

In this chapter, the following were presented:

- The need for the use of modelling in this study
- The process description of the cooling tower air conditioning system
- The NLNG plant design conditions that formed the design criteria for the modelling and sizing of the cooling tower air conditioning system
- The specified design conditions for the modelling and sizing exercise. Date source references were also provided.
- An overview of cooling tower modelling as well as justification for the modelling tools used for this study.
- The modelling methodology used in this study.

- Details of the EES and TRNSYS underlying modelling equations and principles used for the modelling and simulation of the cooling tower, the air conditioning system and the gas liquefaction plant.
- The model verification process.
- Uncertainty analysis for the model.
- Sensitivity analysis of the model

The next chapter presents a detailed analysis of the modelling and sizing results.

CHAPTER 5

RESULTS ANALYSIS

5.1 INTRODUCTION

This chapter presents a detailed analysis of the results obtained from the cooling tower air conditioning system modelling and sizing. This involves the presentation and evaluation of the key results upon which the performance of the cooling tower, with and without the air conditioning system, is assessed. These results are also used to evaluate the liquefied natural gas plant performance, in terms of overall energy requirements and production, with and without the cooling tower air conditioning system. In addition, a simple economic analysis, based on estimated costing for investment, is presented. A general discussion on the identified draw backs, the improvement options, as well as detail technical and economic feasibility assessment tools, for the proposed cooling tower air conditioning system are also presented in this chapter.

5.2 RESULTS

The following are the key results obtained from the modelling and sizing of the cooling tower air conditioning system and the natural gas liquefaction plant.

5.2.1 Results from Modelling

The key results from the real-time dynamic simulation modelling with and without the cooling tower air conditioning system are presented in Table 5.1.

Note: These results are annual average values from the EES model.

Tables 5.2 and 5.3 show the variation of the exit (cold) water temperature and the inlet air wet-bulb temperature of the cooling tower, respectively, based on monthly averages, with and without air conditioning. These results are also shown in Figures 5.1 and 5.2 respectively.

Table 5.1: Key Results Comparison

Variable	Unit	Result without air conditioning	Result with air conditioning
Cooling water exit temperature (from cooling tower)	°C	31.7	30.6
Wet-bulb temperature of inlet air to cooling tower	°C	27.2	25.1
Cooling tower heat transfer	MW	320	360
LNG production	kg/s	134.6	135.4
Propane refrigerant condenser duty	W	1.081E+08	1.08E+08
Propane refrigerant compressor duty	W	2.571E+07	2.562E+07
Propane refrigerant compressor COP	-	3.712	3.725
Mixed refrigerant compressor duty	W	8.266E+07	8.263E+07
Total power consumption of gas liquefaction plant	W	1.084E+08	1.083+E08
Conditioning unit total power consumption	W	N/A	5.573+E07
Total power consumption	W	1.084E+08	1.64E+08

Table 5.2: Monthly Variation Comparison for Cold Water Temperature

Month	Without Air Conditioning (°C)	With Air Conditioning (°C)
January	29.47	28.59
February	29.89	28.96
March	29.88	28.95
April	27.26	26.67
May	30.65	29.68
June	30.3	29.34
July	29	28.22
August	28.82	28.08
September	30.36	29.41
October	29.96	29.03
November	29.41	28.49
December	25.68	25.54

Table 5.3: Seasonal Variation Comparison for Air Wet-bulb Temperature

Month	Without Air Conditioning (°C)	With Air Conditioning (°C)
January	23.63	21.75
February	24.38	22.44
March	24.36	22.42
April	20.47	19.1
May	25.32	23.35
June	25.14	23.18
July	23.09	21.41
August	22.48	20.84
September	24.86	22.92
October	24.41	22.49
November	23.82	21.94
December	14.77	14.39

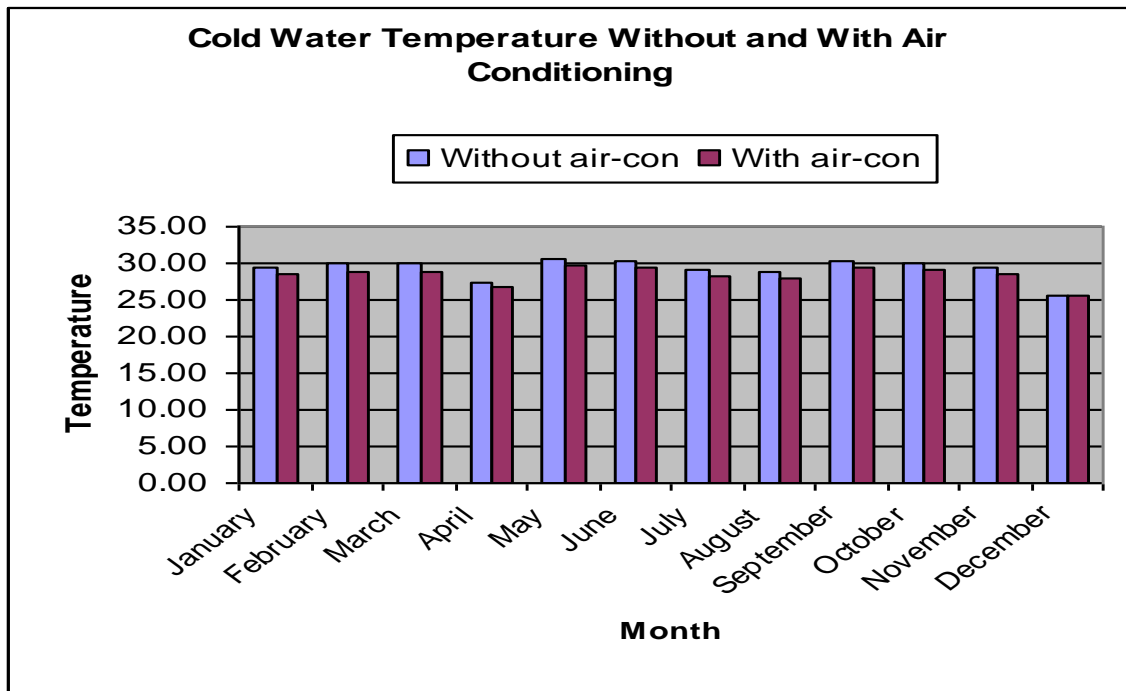


Figure 5.1: Monthly Variation Comparison for Cold Water Temperature

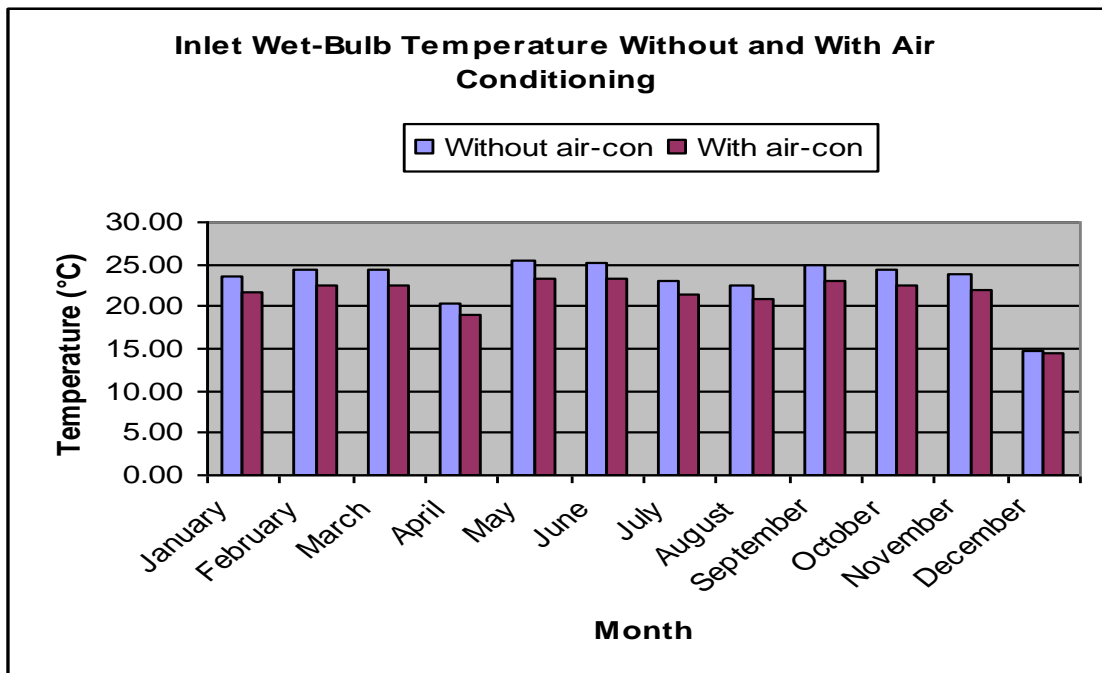


Figure 5.2: Monthly Variation Comparison for Wet-bulb Temperature

Table 5.4 shows the monthly variation of liquefied natural gas (LNG) production with and without air conditioning. These results are also shown in Figure 5.3.

Table 5.4: Monthly Variation Comparison for LNG Production

Month	Without Air Conditioning (kg/s)	With Air Conditioning (kg/s)
January	136.2	136.7
February	136	136.5
March	136	136.5
April	137.4	137.7
May	135.6	136.1
June	135.8	136.3
July	136.5	136.9
August	136.6	137
September	135.7	136.3
October	136	136.5
November	136.3	136.7
December	138.2	138.3

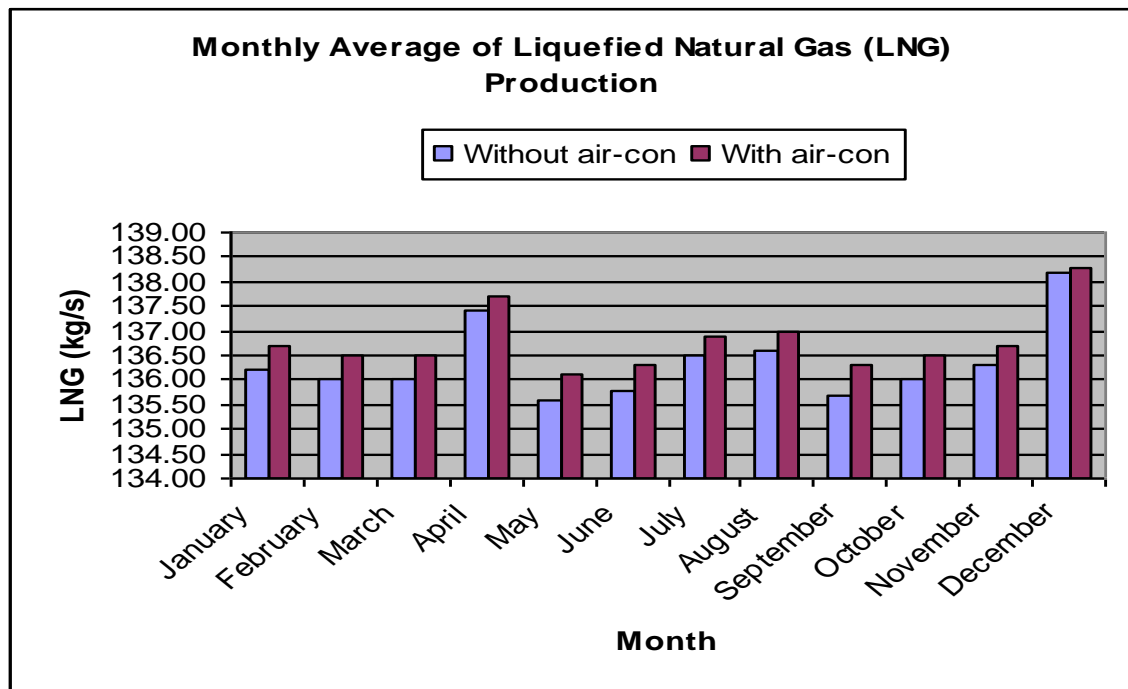


Figure 5.3: Seasonal Variation Comparison for LNG Production

Appendix 4 gives detailed results of the simulation model.

5.2.2 Results from Component Sizing

The sizing of the components of the cooling tower air conditioning system was based on the design process and equations for glycol dehydration system as presented in Chapter 3. Table 5.5 lists the components that were sized together with the parameters calculated.

Details of the Sizing Calculations are presented in Appendix 9, while a snapshot of the calculation spreadsheet is presented in Appendix 5.

Table 5.5: Component Sizing

S/N	Component	Sizing Parameter
1	Absorber Column	<ul style="list-style-type: none"> • Circulation Ratio (liter TEG/kg H₂O absorbed) • Diameter • Height
2	Regenerator	<ul style="list-style-type: none"> • Rich TEG Concentration • Reboiler Duty • Reboiler Number of Tubes • Reboiler Tube Length • Reboiler Tube Bundle Diameter • Still Column Diameter • Surge Vessel Volume
3	Interchange Exchanger	<ul style="list-style-type: none"> • Exchanger Duty • Exchanger Approach
4	Lean Glycol Cooler	<ul style="list-style-type: none"> • Cooler Duty
5	Glycol Circulation Pump	<ul style="list-style-type: none"> • Pump Head • Pump Duty
6	Air Blower	<ul style="list-style-type: none"> • Blower Duty

The following key results were obtained from the sizing of the air conditioning system:

5.2.2.1 Absorber Column

- Circulation ratio = 9.8
- Column diameter (tray type) = 25 m
- Column diameter (packing type) = 20 m
- Column height (tray type) = 7 m
- Column height (packing type) = 18 m

5.2.2.2 Regenerator

- Rich TEG concentration = 90.5 wt%
- Reboiler duty = 46370 kW
- Reboiler number of tubes = 1562
- Reboiler tube bundle diameter = 0.45 m
- Still column diameter = 5 m
- Surge vessel volume = 223 m³

5.2.2.3 Interchange Exchanger

- Exchanger duty = 67602 kW
- Exchanger approach = 20 °C

5.2.2.4 Lean Glycol Cooler

- Cooler duty = 9074 kW

5.2.2.5 Glycol Circulation Pump

- Pump head = 18 m
- Pump duty = 43 kW

5.2.2.6 Air Blower

- Blower duty = 245 kW

Appendices 5 and 9 give details of the sizing calculations

Figure 5.4 shows the process flow diagram with the key results.

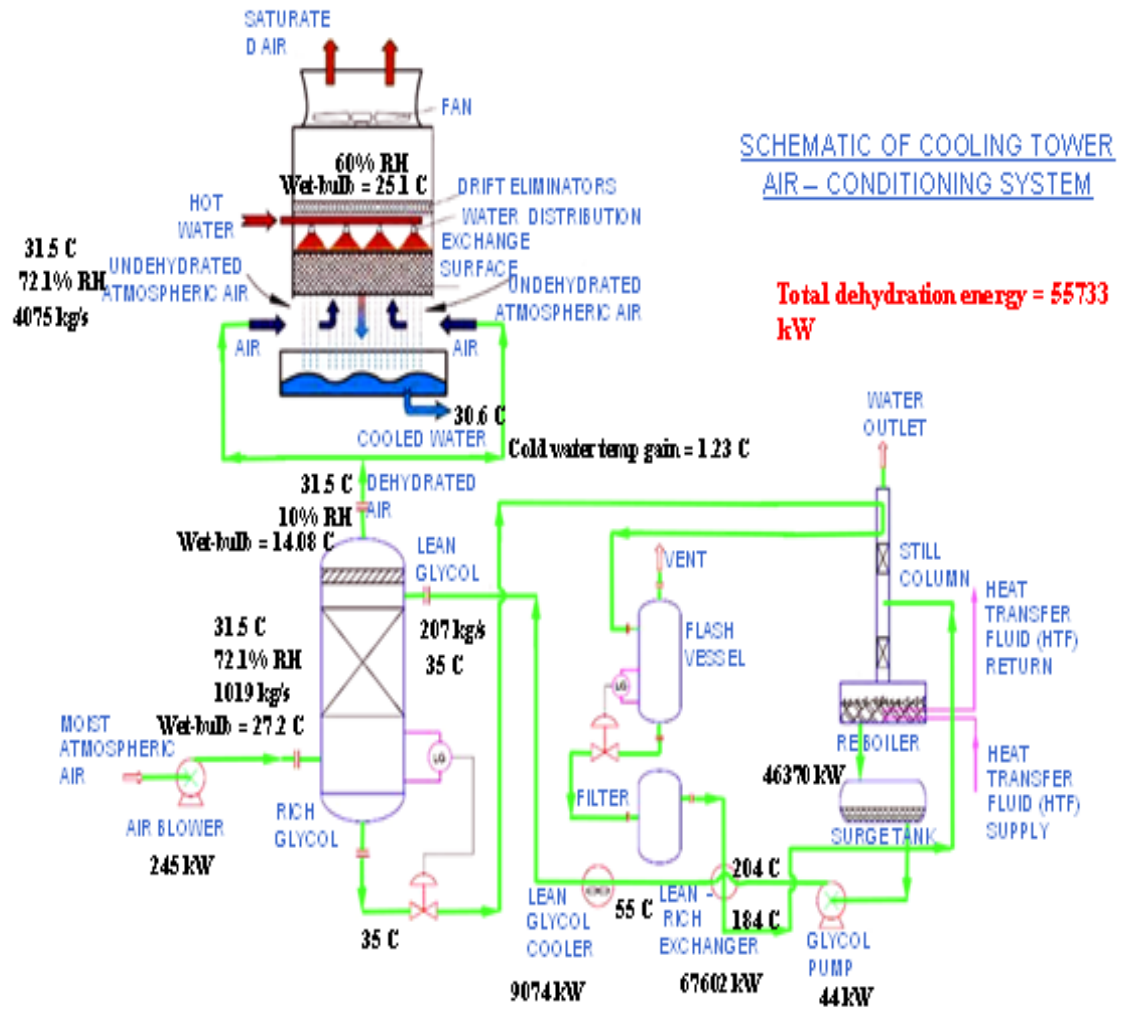


Figure 5.4: Process Flow Diagram with Key Variables

5.3 EVALUATION OF RESULTS

The results obtained from the design analysis and modelling of the cooling tower air conditioning system were evaluated by judging against the following criteria:

- Cooling tower effectiveness
- Tower characteristic
- Tower energy (heat transfer)
- Tower air-conditioning system energy balance
- Compressor performance (for liquefaction) plant
- Overall plant energy balance
- Plant production (of Liquefied Natural Gas)

The evaluation was based on the 9-cell “Actual Condition” results.

5.3.1 Cooling Tower Effectiveness

The cooling tower effectiveness is a measure of the cooling efficiency of the tower. It is used to define the actual cooling in the tower against the maximum possible cooling. It is computed from Equation 5.1:

$$\eta = \frac{T_{w,in} - T_{w,out}}{T_{w,in} - T_{wb}} \quad 5.1$$

Note: T_{wb} is the ambient (entering or inlet) air supply wet-bulb temperature.

Using this equation, the cooling tower effectiveness without air conditioning is 0.66. With air conditioning this effectiveness is calculated to be 0.74. These results are presented in Figure 5.5.

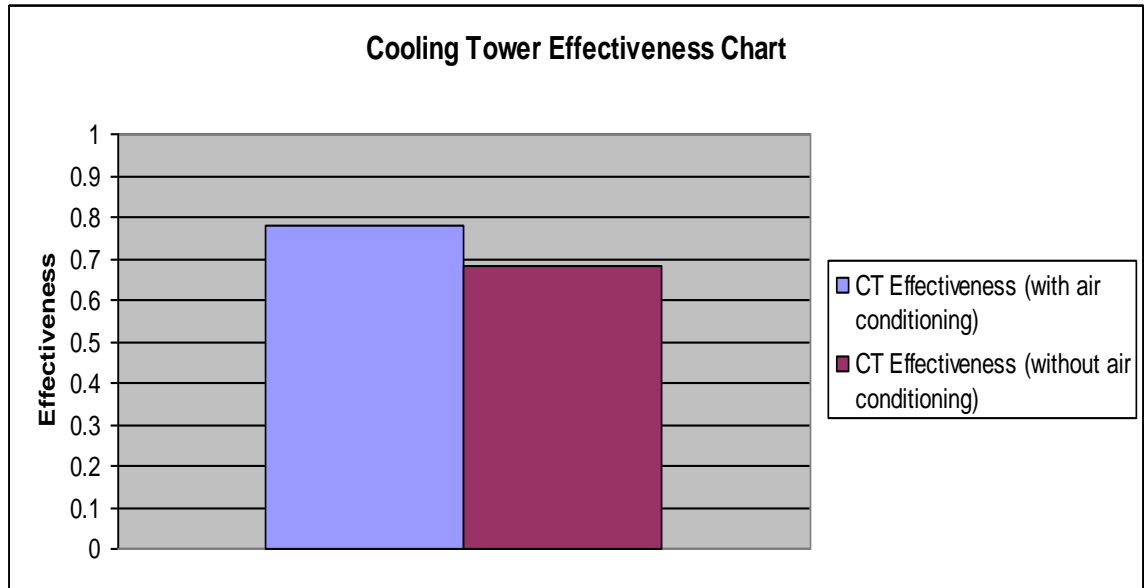


Figure 5.5: Cooling Tower Effectiveness Results

The results show that, by conditioning 20% of the supply ambient air, the cooling tower effectiveness is increased by approximately 10 % for a corresponding decrease in cold water temperature and inlet air wet-bulb temperature of 1.1 K and 2.1 K respectively.

5.3.2 Cooling Tower Characteristics

The tower characteristic, (KaV/L) , is the coefficient of performance of the tower. It can be calculated from Equation 3.1. Table 5.6 and Table 5.7 give a summary of the calculated results of the tower characteristics with and without conditioning of the supply air, respectively.

Table 5.6: Computation of Cooling Tower Characteristic with Pre-Conditioned Air

T (°C)	h_{water} (kJ/kg)	h_{air} (kJ/kg)	$h_w - h_a$ (kJ/kg)	$1/\Delta h$ (kJ/kg)
$T_2 = 30.6$		$h_1 = 76.7$		
$T_2 + 0.1 (T_1 - T_2) = 31.6$	108.35	$h_1 + 0.1L/G (T_1 - T_2) \times c = 83.62$	$\Delta h_1 = 24.73$	0.0404
$T_2 + 0.4 (T_1 - T_2) = 34.6$	126.4	$h_1 + 0.4L/G (T_1 - T_2) \times c = 104.38$	$\Delta h_2 = 22.02$	0.0454
$T_1 - 0.4 (T_1 - T_2) = 36.5$	139.2	$h_2 - 0.4L/G (T_1 - T_2) \times c = 118.21$	$\Delta h_3 = 20.99$	0.0477
$T_1 - 0.1 (T_1 - T_2) = 39.5$	161.92	$h_2 - 0.1L/G (T_1 - T_2) \times c = 138.97$	$\Delta h_4 = 22.95$	0.0436
$T_1 = 40.5 \text{ C}$		$h_2 = 145.89$		sum = 0.1771
$\text{KaV/L} = \frac{(40.5 - 30.6)}{4} [0.1771] \times 4.19 = 1.84$				

Table 5.7: Computation of Cooling Tower Characteristic without Pre-Conditioned Air

T (°C)	h_{water} (kJ/kg)	h_{air} (kJ/kg)	$h_w - h_a$ (kJ/kg)	$1/\Delta h$ (kJ/kg)
$T_2 = 31.7$		$h_1 = 85.95$		
$T_2 + 0.1 (T_1 - T_2) = 32.6$	114.09	$h_1 + 0.1L/G (T_1 - T_2) \times c = 92.10$	$\Delta h_1 = 21.99$	0.0455
$T_2 + 0.4 (T_1 - T_2) = 35.2$	130.32	$h_1 + 0.4L/G (T_1 - T_2) \times c = 110.55$	$\Delta h_2 = 19.77$	0.0506
$T_1 - 0.4 (T_1 - T_2) = 37.0$	142.77	$h_2 - 0.4L/G (T_1 - T_2) \times c = 122.85$	$\Delta h_3 = 19.92$	0.0502
$T_1 - 0.1 (T_1 - T_2) = 39.6$	162.73	$h_2 - 0.1L/G (T_1 - T_2) \times c = 141.30$	$\Delta h_4 = 21.43$	0.0467
$T_1 = 40.5 \text{ C}$		$h_2 = 147.45$		sum = 0.1929
$\text{KaV/L} = \frac{(40.5 - 31.7)}{4} [0.1929] \times 4.19 = 1.78$				

Figure 5.6 shows the comparison of the tower characteristics with and without air conditioning air supply to the cooling tower is pre-conditioned and when the air is not pre-conditioned.

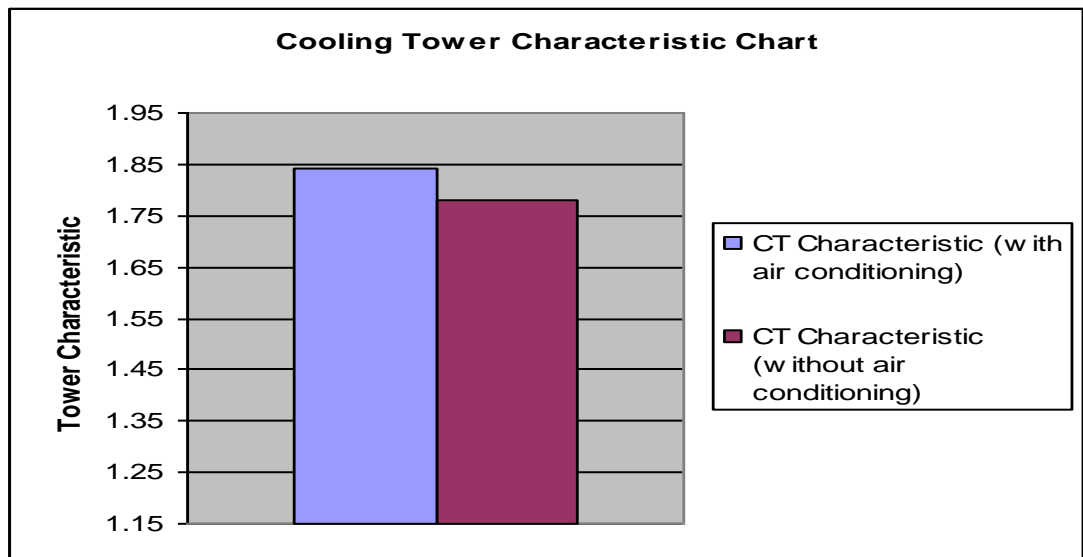


Figure 5.6: Cooling Tower Characteristic Chart

From the results, it is observed that the cooling tower characteristic has increased by 3.5% by conditioning 20% of the supply air for the same water-to-air ratio.

Appendix 6 gives details of the cooling tower characteristic calculation.

5.3.3 Cooling Tower Duty

Figure 5.7 compares the cooling tower heat transfer with and without air conditioning. Also shown on the chart is the design duty of the cooling tower.

From Figure 5.7 it can be seen that the heat transfer by the cooling tower increased from 320 MW to approximately 360 MW through conditioning of 20% of the supply air. It is noted that in both cases, the heat duty of the tower is greater than the design value. This is due to the fact that 9 cells of the tower are continuously on-stream on site (due to higher-than-design production) as against the design of 8 cells.

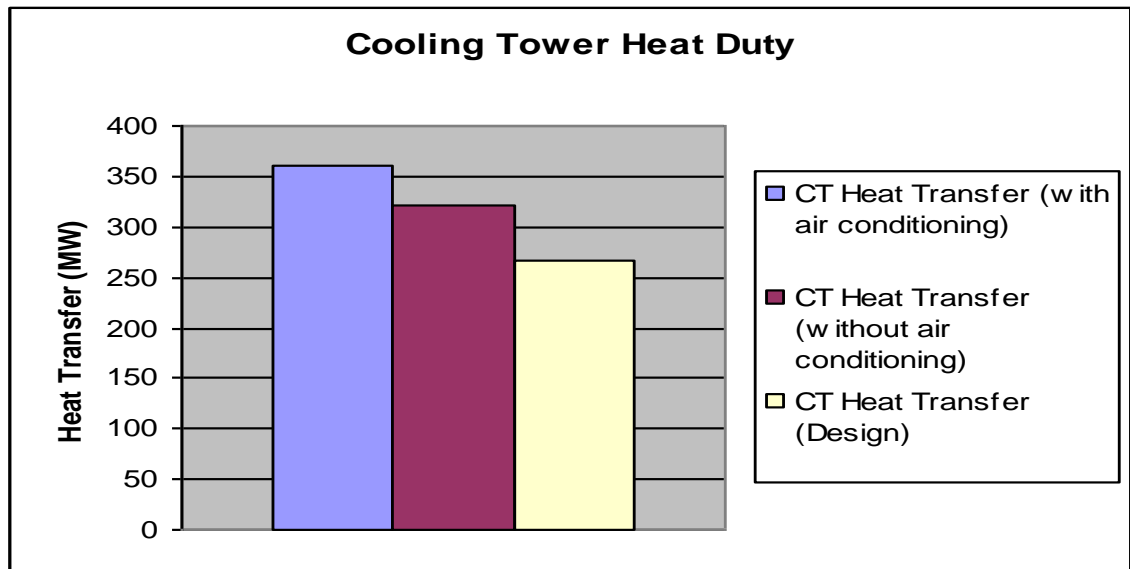


Figure 5.7: Cooling Tower Heat Duty

5.3.4 Gas Liquefaction Refrigerant Compressors Energy Consumption

As explained in Chapter 1, the Nigeria Liquefied Natural Gas (NLNG) plant used as a case study for this research has two refrigerant compressors. These are the propane refrigerant compressor and the mixed refrigerant compressor. The propane refrigerant cycle pre-cools the natural gas and cools / condenses the mixed refrigerant, while the mixed refrigerant cools and liquefies the natural gas. The cooling water from the cooling tower is used mainly by the propane refrigerant cycle for the water-cooled condensers.

Figure 5.8 shows the coefficient of performance (COP) of the propane refrigeration cycle with and without the cooling tower air conditioning system.

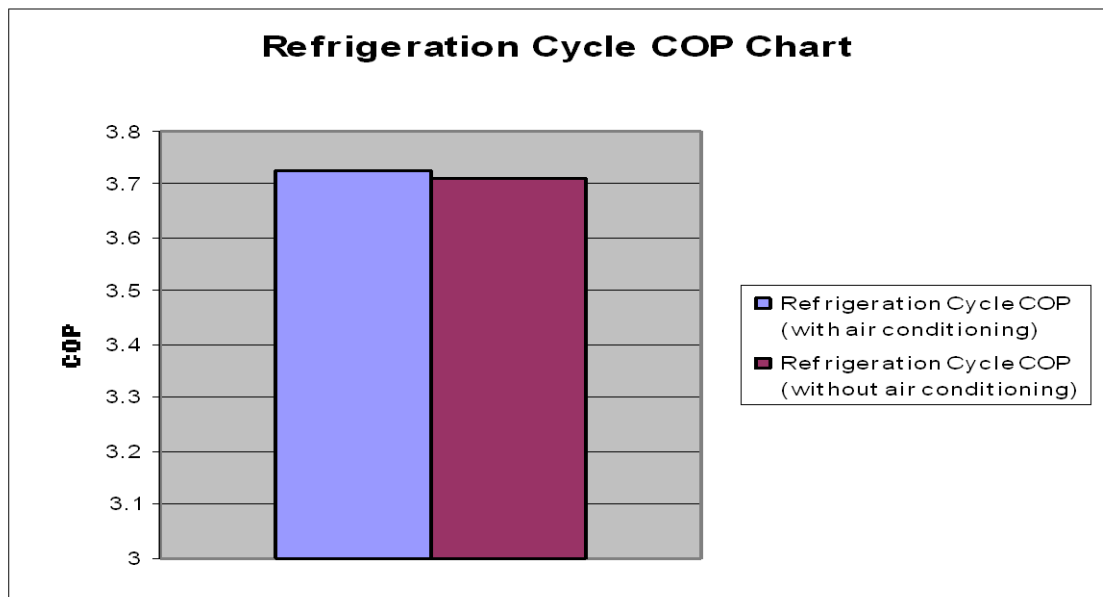


Figure 5.8: Refrigeration Cycle Coefficient of Performance

An increase of 0.013 in the propane compressor COP was obtained as a result of the cooling water temperature reduction of 1.1 K obtained by conditioning 20% of the supply air to the cooling tower. It is however, noted that this increase is marginal in terms of compressor performance.

Figure 5.9 and Figure 5.10 show the compressor power consumed for the propane and mixed refrigerant (MR) compressors, respectively, with and without air conditioning.

The power consumption for the propane refrigerant compressor was reduced by 0.1 MW while that for the mixed refrigerant compressor was reduced by 0.3 MW as a result of reducing the cold water temperature by 1.1 K. via the conditioning 20% of the cooling tower supply air.

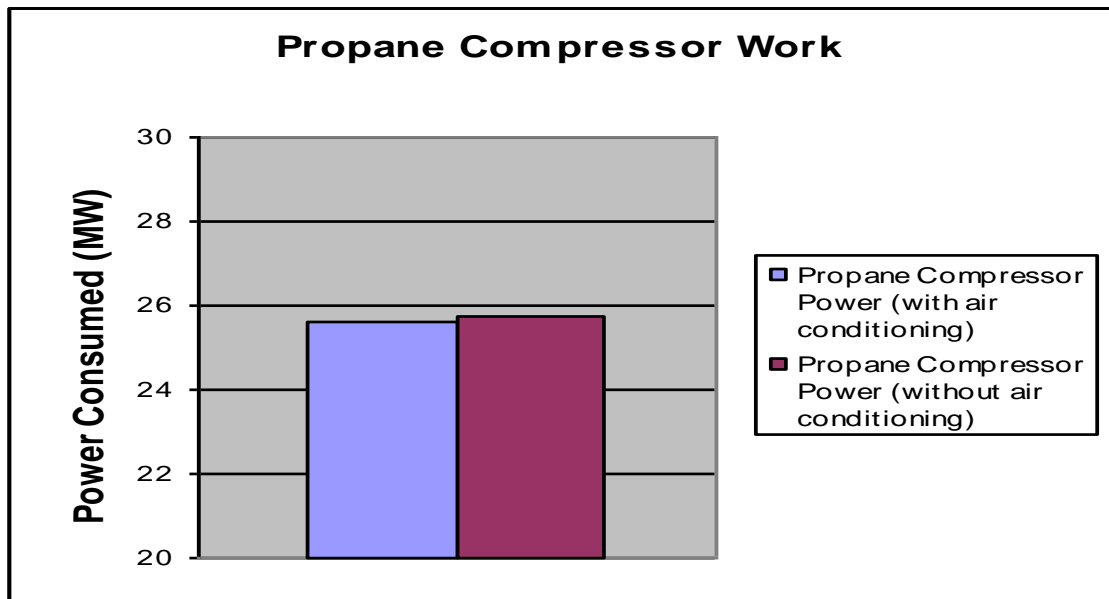


Figure 5.9: Propane Compressor Work

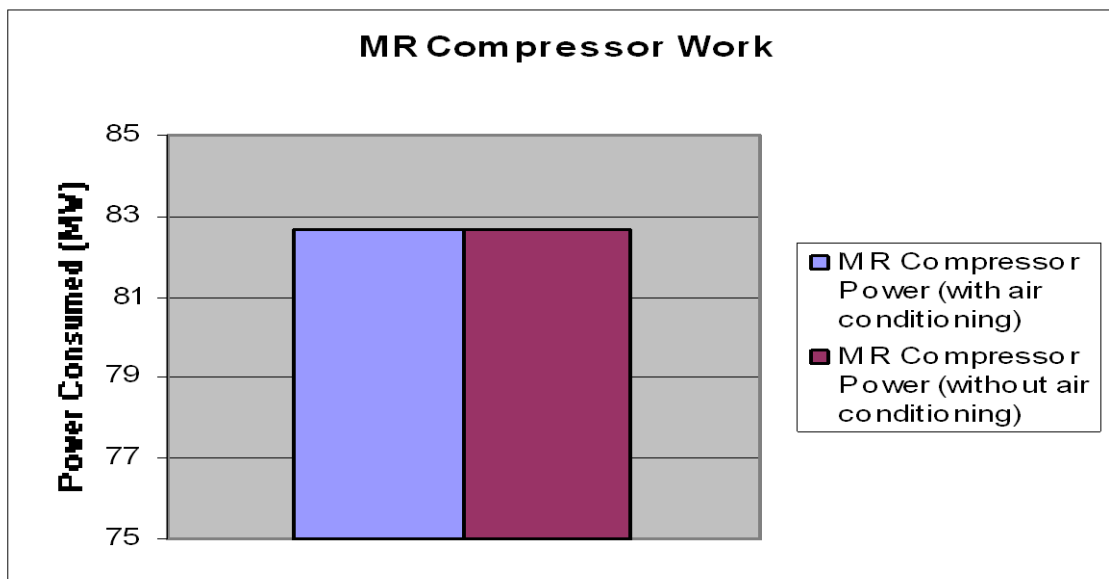


Figure 5.10: Mixed Refrigerant Compressor Work

5.3.5 Plant Production

The impact of conditioning of 20% of the cooling tower supply air on the production of liquefied natural gas (based on the actual sensitivity curve from NLNG) is shown on Figure 5.11.

An increase of 0.8 kg/s was achieved in the production of liquefied natural gas (LNG) as a result of the conditioning of 20% of the cooling tower supply air. The benefit of this increase is dependent on the unit cost of LNG matched against the cost of achieving this increase in production.

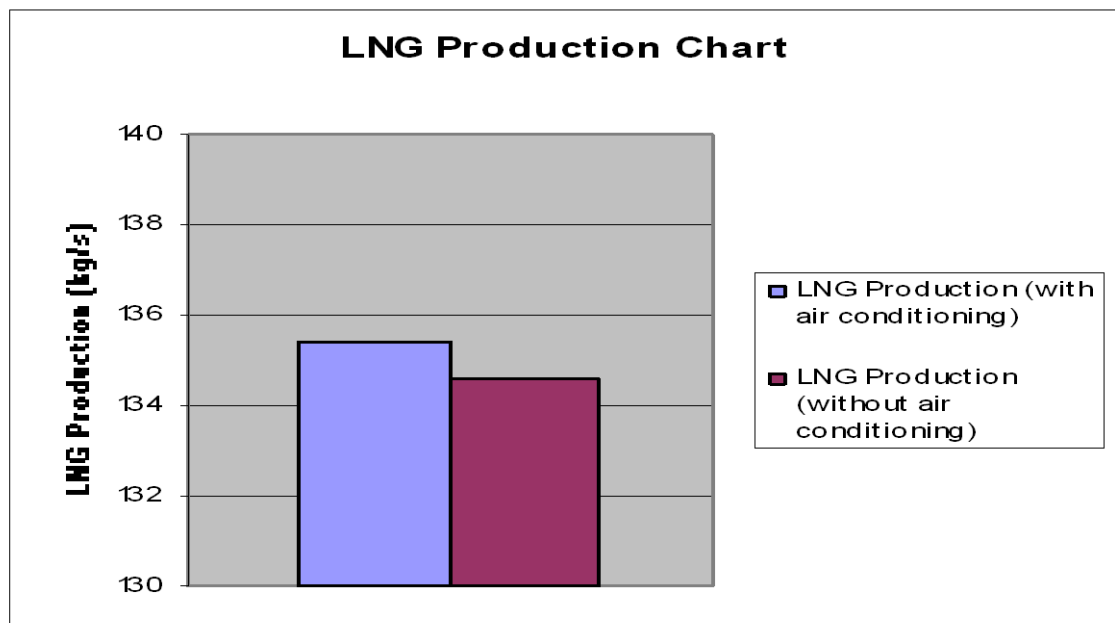


Figure 5.11: Liquefied Natural Gas Production

5.3.6 Cooling Tower Air Conditioning Unit Energy Consumption

A total energy consumption of 56 MW has been calculated for the air conditioning circuit of the cooling tower. Most of the energy (60%) is consumed by the reboiler used for regenerating the glycol. However, the design intent is to use low grade (waste heat) from the plant for the reboiler in order to reduce the energy costs of the air conditioning plant. Also, operational (best practice) and technical optimization

options can further reduce the energy consumption of the air conditioning unit. Some of these strategies are discussed in Chapter 5.5.

5.3.7 Overall Plant Energy Balance

Figure 5.12 shows the overall energy balance of the liquefied natural gas plant in terms of the energy consumed (for production) with and without the conditioning of 20% of the cooling tower supply air as well as the energy required to condition the air.

It is observed that the energy consumption of the liquefied natural gas plant was reduced by 0.1 MW as a result of conditioning 20% of the cooling tower supply air. However, the air conditioning unit used for the conditioning of the air has an energy consumption of 56 MW, which is way beyond the energy savings of the gas liquefaction plant.

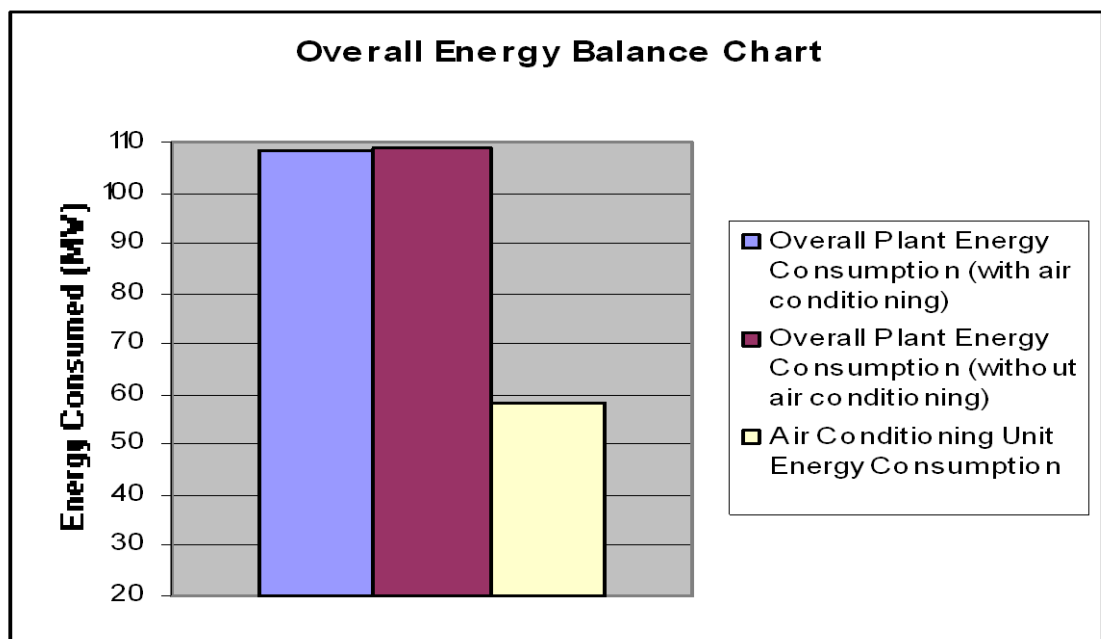


Figure 5.12: Overall Plant Energy Balance

5.4 COST ANALYSIS

This section presents a “basic” economic analysis of the air conditioning unit for the cooling tower with respect to the Nigeria Liquefied Natural Gas (NLNG) plant which has been used as a case study. The economic analysis involves estimation of the initial capital investment costs for the unit design, operating costs together with expected cashflows.

As explained earlier, the cooling tower air conditioning unit has the following main components:

- The Absorber (Contactor)
- The Reboiler
- The Rich – Lean Heat Exchanger
- The Lean Glycol Cooler
- The Circulation Pump
- The Air Blower

The *Matches* online process equipment cost model was used for the cost analysis (Matches, 2003). Matches provides internet based conceptual process engineering cost analysis to optimise investment in research and development, acquisitions, production expansion, addition of new products or changes in operation. Matches has a wide range of process and equipment information. It provides a capital cost with a generic description for a single piece of equipment or equipment module. In addition if available, the equipment information includes utility consumption and costs. The site contains free cost correlations for over 270 pieces of equipment. By supplying the relevant input information, e.g., mass flow, operating pressure, temperature, etc, the matches online model can provide cost estimate for the given equipment.

The costing (in US\$) for the different components of the proposed cooling tower air conditioning unit are as follows:

5.4.1 Costing for Absorber

The absorber specification is as follows:

Working pressure = 1 bar (10^5 Pa)

Air inlet temperature = 30°C

Glycol (TEG) inlet temperature = 35°C

Air flow rate = 1019 kg/s = 72 MMSCM/day (2537 MMSCF/day)

Air inlet water (moisture) content = 0.02331 kg/kg (0.02588 kg/m³)

Specified air outlet water content = 0.00271 kg/kg (0.00301 kg/m³)

TEG circulation volume = 667735 liter/hour

Circulation ratio (liter TEG/kg water absorbed) = 9.8

Diameter (for bubble cap tray type) = 25 m

Diameter (for packing type) = 20 m

Height (for bubble cap tray type) = 7 m

Height (for packing type) = 18 m

Number of trays = 12

Based on the above specification, the costs of the absorber are as follows:

Bubble cap tray type absorber = **\$945,000**

Packing type absorber = **\$804,700**

5.4.2 Costing for Reboiler

The reboiler specification is as follows:

Hot side fluid = Hot oil (HTF)

Cold side fluid = Triethylene glycol (TEG)

HTF inlet temperature = 235°C

HTF outlet temperature = 192°C

TEG inlet temperature = 184°C

TEG outlet temperature = 204°C

HTF mass flow rate = 373 kg/s

TEG mass flow rate = 207 kg/s

Reboiler duty = 46370 kW

Working pressure = 1 bar (10⁵ Pa)

Heat transfer surface area = 1932 m²

Overall heat transfer coefficient = 1398 W/m² K

Cross sectional area per tube = 0.000195 m²

Length per tube = 20 m

Total number of tubes = 1562

Number of tube passes = 4

Tube bundle diameter = 0.6 m

Note: Specified lean glycol quality is 98.8 wt% and calculated rich glycol quality is 90.5 wt%

Based on the above specifications, the cost of the reboiler is **\$823,612**.

5.4.3 Costing for Still Column

The still column specification is as follows:

Working temperature = 240°C

Working pressure = 1 bar (10^5 Pa)

Design temperature range – min / max = 20°C / 320°C

Design pressure = 10 bar (10×10^5 Pa)

TEG mass flow rate = 207 kg/s

TEG density = 1120 kg/m³

Still column diameter = 5 m

Based on the above specifications, the cost of the still column is **\$26,600**.

5.4.4 Costing for Lean -Rich Glycol Heat Exchanger

The lean-rich glycol heat exchanger specification is as follows:

Hot side fluid = Lean TEG

Cold side fluid = Rich TEG

Lean TEG inlet temperature = 204°C

Lean TEG outlet temperature = 55°C

Rich TEG inlet temperature = 35°C

Rich TEG outlet temperature = 184°C

TEG mass flow rate = 207 kg/s

Exchanger duty = 67602 kW

Based on the above specifications, the cost of the lean-rich heat exchanger is **\$29,100**.

5.4.5 Costing for Lean Glycol Cooler

The lean glycol cooler specification is as follows:

Hot side fluid = Lean TEG

Cold side fluid = Air

Lean TEG inlet temperature = 55°C

Lean TEG outlet temperature = 35°C

Air temperature = 30°C

Approach = 5°C

TEG mass flow rate = 207 kg/s

Exchanger duty = 9074 kW

Based on the above specifications, the cost of the lean glycol cooler is **\$33,500**.

5.4.6 Costing for Glycol Circulation Pump

The glycol circulation pump specification is as follows:

Pumping fluid = Triethylene glycol (TEG)

TEG temperature = 204°C

TEG density = 1120 kg/m³

TEG volumetric flow rate = 0.185 m³/s

Pump head = 18 m

Pump efficiency = 0.88

Motor efficiency = 0.95

Pump duty = 43 kW

Based on the above specifications, the cost of the glycol circulation pump is **\$6,200**.

5.4.7 Costing for Air Blower

The conditioning unit air blower specification is as follows:

Pressure drop across absorber = 200 Pa

Air volumetric flow rate = 918 m³/s

Blower efficiency = 0.75

Blower duty = 245 kW

Based on the above specifications, the cost of the air blower is **\$412,400**.

5.4.8 Costing for Surge Vessel

The glycol vessel specification is as follows:

Working temperature = 240°C

Working pressure = 1 bar (10⁵ Pa)

Design temperature range – min / max = 20°C / 320°C

Design pressure = 10 bar (10×10^5 Pa)

TEG mass flow rate = 207 kg/s

TEG density = 1120 kg/m³

Retention time = 20 minutes

Surge vessel volume = 223 m³

Based on the above specifications, the cost of the surge vessel is **\$420,000**.

5.4.9 Total Cost for Cooling Tower Air Conditioning Unit Design

A summary of the estimated total initial investment (capital) cost for the cooling tower air conditioning unit design is shown in Table 5.8.

5.4.10 Operating (Running) Costs Analysis

The bulk of the operating costs for the cooling tower air conditioning unit come from the purchase of glycol (TEG) inventory and maintenance of the unit.

5.4.10.1 Costs of Glycol Inventory

The required glycol (TEG) inventory is based on the following data:

Lean glycol quality = 98.8%

Circulation ratio (liter TEG per kg water absorbed) = 9.8

Water absorbed per second = 19 kg/s

Required TEG volume (per second) = 185 liters

TEG retention time in air conditioning unit = 20 minutes

Total TEG inventory required = $185 \times 20 = 3700$ liters

Cost per liter of TEG = US\$8

Total cost of TEG inventory required = $3700 \times 8 = \mathbf{US\$29600}$

Additional glycol inventory would also be required for make-up due to losses. The rate of make-up depends on the loss rate and this, in turn, depends on the absorber and still column design as well as operating practice.

In addition to losses, glycol degradation may also lead to the need to replace the inventory. However, degradation is most likely to be avoided if the glycol is not heated above the degradation temperature of 204° in the reboiler.

5.4.10.2 Costs of HTF Inventory

The costs of the heat transfer fluid (HTF) inventory would increase the operating costs. Similarly, the costs of heating the HTF the operating costs. However, for this study, it has been assumed that waste heat (low grade heat, e.g., from turbine exhaust flue gas) from the plant would be used for heating up the HTF.

5.4.10.1 Energy Costs

The electricity costs due to energy required to run the air conditioning unit (pumps, blowers, etc) would further increase the overall operating costs of the system. Although a power savings of 1 MW from the liquefaction plant has been achieved (due to reduced temperature of cooling tower) by conditioning 20 % of the cooling tower inlet air, the calculated power required to run the air conditioning unit (56 MW) is far in excess of the power savings. This will result in overall increase in energy costs for the NLNG plant. Electricity used within the plant is generated by NLNG by use of ten gas turbines of 34 MW each. The unit cost of electricity generated within the plant is a function of the capital (initial investment) costs and the operating costs (for fuel, oil, spare parts, routine maintenance. The detail of this costing is beyond the scope of this research.

5.4.11 Costs of Maintenance

Routine maintenance would have to be carried out on the cooling tower air conditioning unit at regular interval. The cost of maintenance would depend on the maintenance philosophy (e.g., run-to-failure or preventive maintenance), schedule (frequency) and scope (major, minor, partial or complete shutdown maintenance). The cost of maintenance would also depend on the personnel and material requirements as well as service charge. These costs vary with time and location and hence cannot be easily estimated.

It is however important to note that, maintenance costs can be reduced by operational excellence.

Common problems in a glycol dehydration unit are fouling and foaming as a result of excessive amount of liquid and solid contaminants present in the process stream. Foaming in the contactor (absorber) can result in huge glycol losses, reduced operating capacity and off-spec product. Fouling can result in reboiler tube failure, tray plugging in the contactor and the regenerator, exchanger failures and frequent pump seal replacements. Since the glycol is a circulating system, the solids concentration will gradually increase unless removed by a filter. Hence, installing the right type of filter is very important (Campbell, 2004).

Table 5.8: Initial Investment Costs for Cooling Tower Air Conditioning Unit

S/N	ITEM	QUANTITY	UNIT COST	TOTAL COST
1	Absorber	1	\$945,000	\$945,000
2	Reboiler	1	\$28,700	\$823,612
3	Still Column	1	\$26,600	\$26,600
4	Lean-Rich Exchanger	1	\$29,100	\$29,100
5	Lean Glycol Cooler	1	\$33,500	\$33,500
6	Circulation Pump	1	\$6,200	\$6,200
7	Air Blower	1	\$412,400	\$412,400
8	Surge Vessel	1	\$420,000	\$420,000
GRAND TOTAL			\$2,696,412	

Note: These are estimated costs.

5.4.12 Expected Cash Inflow

Cash inflow expected as a result of the cooling tower air conditioning system are derived from the following

- Increased LNG production - The expected cash inflow is calculated as follows:
 Liquefied Natural Gas (LNG) production gain (due to air conditioning) = 0.8 kg/s = 69.12 Metric Ton per day
 Unit price of LNG = \$300 per Metric Ton
 Savings on LNG per day = \$20736
 Savings on LNG per year = \$7,568,640
- Fuel savings - Currently, the gas turbines (two) driving the propane and mixed refrigerant compressors consume a total of approximately 600 metric tons per day (~7 kg/s) of natural gas. The design power of the turbine driving the propane compressor is 39 MW, while that for the turbine driving the mixed refrigerant compressor is 87 MW. Savings on fuel consumption by the propane and mixed refrigerant turbines due to the 0.1 MW and 0.3 MW power reductions on the respective compressors have not been calculated in this study. It is however, noted that the cost of one metric ton of natural gas fuel for the turbine is ~ \$300.

5.5 DISCUSSION

5.5.1 Results

Comparison of the results obtained by conditioning (reducing the wet-bulb temperature) of 20% of the cooling tower inlet air against non-conditioning of the inlet air, shows the potential of this concept to improve the thermal performance of the cooling tower and indeed the overall energy and production efficiency of the gas liquefaction plant that was used as a case study. However, it has been observed that the net energy gain is negative as the energy requirement for the air conditioning unit is much greater than the energy savings due to the conditioned air supplied to the cooling tower. On the other hand, cash inflow is expected to accrue, based on the LNG production increase and the savings in fuel consumption of the gas turbines

driving the propane and mixed refrigerant compressors. However, operating and maintenance costs, as well as actual cost of procurement and installation of equipment, have to be taken into account in a detailed cost analysis (net present value, internal rate of return, etc) before conclusions, on investment benefits or otherwise, can be drawn.

Results from the sensitivity analysis results, as presented in Section 4.11 of Chapter 4, shows that variations in the wet-bulb temperature (or relative humidity) of air does have an impact on the exit water temperature of the cooling tower. There is approximately a 0.5 K decrease in the cooling tower exit water temperature for every 1 K decrease in the wet-bulb temperature of the air. The sensitivity of the exit water temperature to variations in the wet-bulb temperature of air is greater than that of variations in the dry-bulb temperature of the air. This is so because over 80 % of the heat transfer in the cooling tower is as a result of evaporation of water into air, which is a direct function of the moisture content (and hence wet-bulb temperature) of the air. This implies that the relative measured relative humidity of the inlet air to the cooling tower, used as input into the model, have to be very accurate in order for the model results to be reliable.

The seasonal (monthly) comparison for cold water temperature, inlet air wet-bulb temperature and liquefied natural gas production, without (base case) and with air conditioning shows marginal differences, as shown by Figures 5.1, 5.2 and 5.3. This is due to the fact that Bonny Island, where the NLNG plant is located, is in the tropical region and subject to very minimal variations in climatic conditions (ambient temperature, pressure, humidity, etc.) during the year. The only period with noticeable climatic change is in the month of December. During this month, there is a marked drop in the humidity (and hence wet-bulb temperature) of the ambient air due to the dust blowing from the Sahara desert that absorbs the moisture from the air. This period is known as the Harmattan season. As the cooling tower performance depends on humidity of the air for the major part of its heat transfer, there is always a drop in the cold water temperature from the cooling tower during this period and an associated increase in liquefied natural gas production at the NLNG plant.

5.5.2 Optimisation Strategies

Various technical and operational optimisation solutions can be explored in order to reduce the energy demand / utilization of the cooling tower air conditioning unit. One obvious option would be to utilize waste heat (e.g., heat from turbine exhaust flue gas) for heating up the reboiler heat transfer fluid (HTF) or as gas for heating up the reboiler directly. Another option would be to run the cooling tower air conditioning unit on a demand-basis, i.e., the unit would be put in service (brought on-stream) only when the cold water temperature of the cooling tower exceeds the plant threshold of 32°C. As explained in Chapter 1, above this threshold temperature the propane refrigerant compressor is constrained due to the high discharge line pressure set by the water cooled condensers. It is beyond this threshold temperature of the cold water from the cooling tower that the production loss (due to the constrained propane refrigerant compressor) becomes significant – approximately 3 kg/s of LNG loss in production per degree rise in cooling water temperature to the condensers beyond 32°C. Below this threshold temperature the production loss is only 0.5 kg/s per degree rise in cooling water temperature. Figure 6.1 shows the actual cooling water temperature versus production sensitivity curve of the Nigeria Liquefied Natural Gas (NLNG).

Note: Figure 5.13 is identical to Figure 4.15. It has been “repeated” here for a better understanding of the discussions.

Further reduction in the energy consumption of the cooling tower air conditioning unit can be achieved by designing the system such that the regeneration of the TEG is performed by direct contact with flue gas from a gas turbine exhaust (there are ten 34 MW gas turbines at the NLNG plant).

Using more “robust” design components (e.g., high efficient absorber tray / packing designs, heat exchangers, pump, blower fan, etc) could provide a technical optimization for the cooling tower air conditioning unit. This could minimize energy consumption, minimize glycol inventory / loss and minimize component sizes.

Plant feed and cooling water temperature at constant compositions and conditions

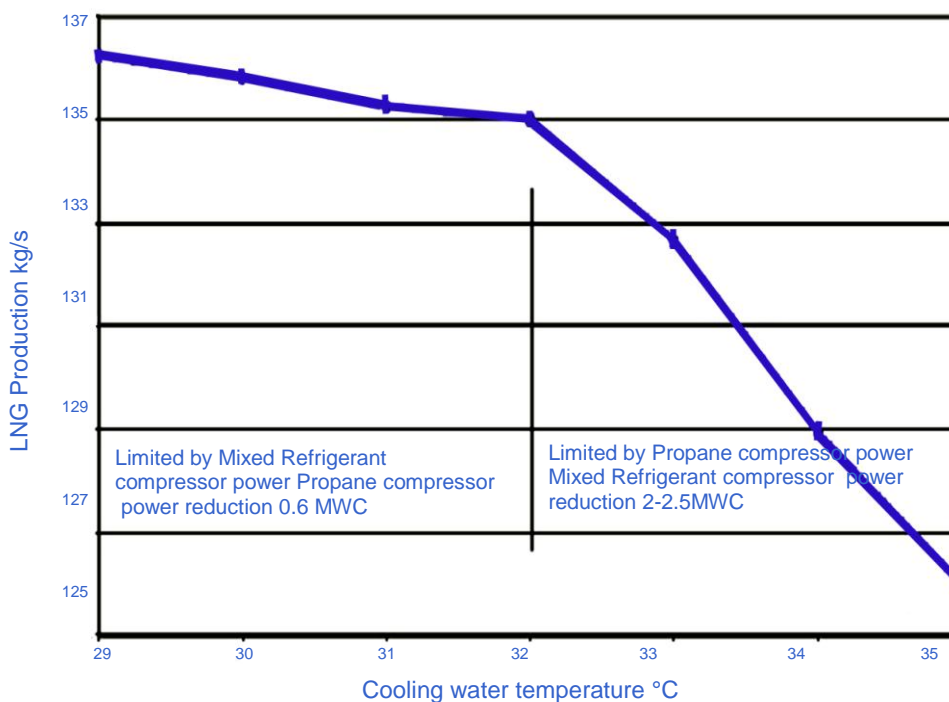


Figure 5.13: NLNG Plant Sensitivity Curve – *production vs. cooling water temperature* (courtesy of NLNG Operating Manual)

One drawback of the cooling tower air conditioning design is the rather large size of the absorber column (20 m diameter) which is as a result of the high flow rate of air. Physical construction of this size of column is not likely to be feasible (unless it is a tank!). This research has been carried out using a “real life” Liquefied Natural Gas (LNG) plant, which has a high design capacity (about 4 mega tons of LNG production per year). Only 20% of the total air flow rate to the cooling tower of this plant (5094 kg/s) was conditioned in this study. One option to manage the size and air flow rate calculated is to use multiple parallel streams or trains for the absorber, for example, one air blower supply air to four absorber columns of 5 m each. In terms of reduction in investment cost, one option could be to replace the air blower upstream of the absorber with a centrifugal fan covered by just a shroud. The cost of a fan is much lower than that of a blower.

5.5.3 Statistical Significance Test

One of the challenges faced in trying to improve a process or system is to be reasonably certain that the new process really is better before making the necessary change. The new process will have to be compared with an existing one. The method of testing the data to see if the results are considerably or significantly better is called a significance test, which involves the testing of a hypothesis. The hypothesis to be tested is called the null hypothesis and is denoted by H_0 . Any other hypothesis is called the alternative hypothesis, and is denoted by H_1 . The null hypothesis theory assumes there is no difference between the two processes. In statistical analysis, the null hypothesis is assumed to be true until the data indicates otherwise. Thus, the burden of proof is on H_1 and departures from H_0 , is the point of interest (Chatfield, 1970).

For example, in the case of this research study, it must be decided whether to accept or reject the hypothesis that there is no difference between this new concept of conditioning the inlet air to cooling towers and current existing process with no conditioning of air. Denoting the annual mean temperature of the cold (exit) water temperature from the cooling tower for the existing (without air conditioning) and new (with air conditioning) processes as μ_0 and μ respectively, then the null hypothesis is given by

$$H_0: \mu = \mu_0$$

The second possible theory in this situation is that the new process is better than the existing process and this is the alternative hypothesis given by

$$H_0: \mu > \mu_0$$

A detailed statistical significance test is beyond the scope of this study and has been recommended for future work.

5.5.4 Investment Appraisal

In addition to the statistical significance test, a detailed cost analysis would have to be carried out before a business decision can be made to invest or not to invest in the installation of the proposed cooling tower air conditioning system. Some of the available investment appraisal techniques that can be used for this purpose include:

- Payback period - this is a simple technique for assessing an investment by seeing how long it would take to be repaid. It is quite useful when used to compare similar investments. However, as a stand-alone tool to compare “investment” with “no-investment”, payback period has no explicit criteria for decision-making. It is considered a method of analysis with serious limitations and qualifications for its use, because it does not account for time value of money, risk, financing or other important considerations such as operating costs. It is generally agreed that this tool for investment decisions should not be used in isolation (Business Link, 2009).
- Net present value (NPV) – this uses discounting cashflow to evaluate potential investment. The NPV calculates the present value of all cashflow associated with an investment: the initial investment outflow and the future cashflow returns. The higher the NPV the better. The key advantage of NPV is that it takes into account the time value of money. The main disadvantage of NPV is that it is a sophisticated and relatively complicated way of evaluating a potential investment; choosing the right discount rate to use to calculate NPV is difficult. The discount rate needs to take into account the riskiness of an investment project and should at least match the cost of the capital (Business Link, 2009).
- Internal Rate of Return (IRR) – this is a rate of return used in capital budgeting to measure and compare the profitability of investments. It is also called the discounted cash flow rate of return or simply cash flow rate. It is an annualised effective compounded return rate that can be earned on the investment capital. In other words, the IRR of an investment is the interest rate at which the costs of the investment lead to the benefits of the investment. This means that all gains from the investment are inherent to the time value of money and that the investment has a zero net present value at this interest rate. Because the IRR is a rate quantity, it is an indicator of the efficiency, quality or yield of an investment. This is in contrast with the net present value, which is an indicator of the value or magnitude of an investment. An investment is considered acceptable if its internal rate of

return is greater than an established acceptable minimum rate of return (e.g., a minimum return rate agreed to by equity holders in a firm).

A detailed investment appraisal, for the cooling tower air conditioning system, is beyond the scope of this study and has been recommended for future work.

5.6 SUMMARY

In this chapter, the following were presented:

- The key performance results from the modelling and components sizing carried out.
- A detail evaluation of the results with respect to performance criteria.
- A basic economic analysis involving the estimation of the initial investment cost, the operating cost and expected cash inflow for the proposed cooling tower air conditioning system.
- A discussion on the results and evaluations carried out with respect to the technical and economic feasibility, as well as operating strategies / improvement options, of the proposed cooling tower air conditioning system.

In the next chapter, a detail discussion of the results analysis is presented. Also presented in the next chapter, are the conclusions drawn from this research study as well as the recommendations therefrom.

CHAPTER 6

DISCUSSION, CONCLUSIONS AND RECOMMENDATIONS

6.1 INTRODUCTION

This chapter presents the conclusions that have been drawn from this research study and recommendations thereupon are also presented.

6.1 CONCLUSIONS

Based on the analysis and discussions that have been carried out in this research study on improving the thermal performance of cooling towers by conditioning of the air, the following conclusions have been drawn:

1. This research study has demonstrated that it is possible to reduce the wet-bulb temperature of the ambient air supply to a cooling tower.
2. Reducing the wet-bulb temperature of the supply air by means of a liquid desiccant does improve the thermal performance of the cooling tower used for the case study. The effectiveness of the cooling tower increased by 10% and the cold water temperature reduced by 1.1 K.
3. The reduction in cold water temperature from the cooling tower resulted in 0.2 MW reduction in energy consumption by the refrigeration cycles in the gas liquefaction plant used as a case study.
4. The overall energy balance of the plant shows a net energy loss as the energy consumption of the cooling tower air conditioning unit was much higher than the energy savings in the refrigeration cycles of the gas liquefaction plant.
5. A production increase of 0.8 kg/s of LNG was achieved as a result of the increased thermal performance of the cooling tower. Expected cash inflow, based on the LNG production increase, have been calculated to be \$7,568,640 per annum
6. The technical and operational optimization options that have been proposed have the potential to reduce the energy consumption as well as costs of the cooling tower air conditioning unit.

7. The results of the analysis indicated the feasibility of this new and novel approach. However, significant further work will be required before the concept can be implemented in reality.

6.2 RECOMMENDATIONS

Based on this research study the following are recommended:

1. A complete statistical significance test to determine if there are enough benefits (based on technical and commercial considerations) to justify implementation of the new concept of conditioning of cooling tower inlet air, for industrial processes.
2. Further study on the applicability of the air conditioning concept to other industrial plants such as power generation plants, oil refinery plants, chemical plants, nuclear plants, etc.
3. Further study on the applicability of the air conditioning concept to natural draught cooling towers.
4. Further study on the use of other liquid desiccants, such as Lithium Bromide, Lithium Chloride, etc, for the cooling tower air conditioning unit. The presence of oxygen does enhance the degradation of glycol desiccant, which has been used in this study.
5. Building of a prototype cooling tower air conditioning unit using the models and design approach of this research study to verify the practical application of this new concept.
6. A detail cost analysis, taking into consideration the initial investment (capital) costs, the operating costs (glycol / HTF inventory, electricity requirement for equipment, maintenance, etc) and the expected returns (LNG production increase and savings in fuel consumption by gas turbines), to draw conclusions on the benefit-to-costs ratio for the cooling tower air conditioning unit and hence form the basis for investment decision-making.

6.3 SUMMARY

In this chapter, the following were presented:

- Conclusions drawn from this research study
- Recommendations for further studies on this research subject

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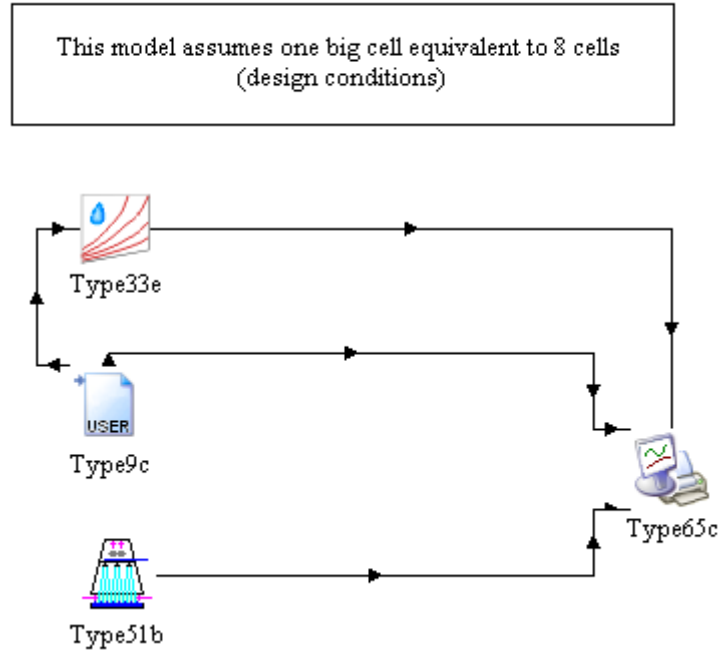
Appendix 1: One Year Measured Time Series (Real-Time) Dataset from NLNG Plant

1	Supply (hot) water temp	Supply water flowrate	Dry-bulb temp of supply air	Relative humidity of supply air	Exhaust (cold) water temp	Make-up water flow rate	Design air flow per cell	Design air density
2	°C	kg/h	°C	%	°C	kg/s	kg/s	kg/m ³
3	40.21	8076	27.46	75.8	32.37	52	566	1.1
4	40.49	8076	27.77	75.8	32.43	52	566	1.1
5	40.55	8076	29.55	75.8	32.92	52	566	1.1
6	40.55	8076	27.64	75.8	32.77	52	566	1.1
7	39.73	8076	25.83	75.8	32.01	52	566	1.1
8	38.45	8076	23.64	84.8	31.03	52	566	1.1
9	38.99	8076	25.55	81.0	31.40	52	566	1.1
10	38.99	8076	26.14	72.0	30.85	52	566	1.1
11	40.15	8076	29.86	70.8	32.71	52	566	1.1
12	40.43	8076	28.36	84.0	32.74	52	566	1.1
13	39.33	8076	26.14	81.2	32.07	52	566	1.1
14	35.12	8076	29.86	37.6	28.65	52	566	1.1
15	38.96	8076	27.18	84.6	32.04	52	566	1.1
16	38.72	8076	23.78	85.1	31.37	52	566	1.1
17	39.54	8076	28.82	73.4	32.43	52	566	1.1
18	38.96	8076	27.46	78.1	32.07	52	566	1.1
19	39.54	8076	29.68	70.3	32.55	52	566	1.1
20	38.45	8076	27.64	72.1	31.37	52	566	1.1
21	39.54	8076	26.45	81.5	32.31	52	566	1.1
22	40.00	8076	29.27	75.5	32.71	52	566	1.1
23	40.03	8076	27.91	79.0	32.68	52	566	1.1
24	38.38	8076	25.41	92.0	31.24	52	566	1.1
80	39.73	8076	31.91	63.2	32.22	52	566	1.1
678	41.28	8076	26.28	84.1	31.97	52	566	1.1

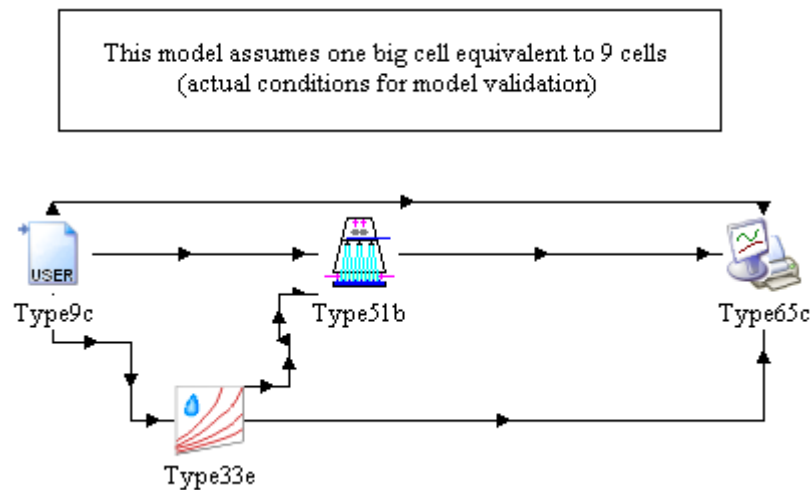
Appendix 2: TRNSYS and ESS Simulation Models

- **TRNSYS MODELS**

1. Steady-State Condition Model



2. Real-Time Dynamic (actual) Condition Model



- **ESS MODELS**

"!COOLING TOWER PERFORMANCE IMPROVEMENT MODEL (With Air Conditioning)"

"ESTIMATION OF HEAT TRANSFER COEFFICIENT OF A COOLING TOWER (FOR TOWER SIZING)"

PROCEDURE DCCTSIZING

(M_dot_w,M_dot_a,t_w_su,t_w_ex,t_wb_su,C_0,C_1,C_2,C_3:AU_n,Q_dot_n,NTU_f)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Assumption"

"1. Initial guess of exhaust wet-bulb temperature"

"Calculate the supply moist air enthalpy"

$h_{a_su}=C_0+C_1*t_{wb_su}+C_2*t_{wb_su}^2+C_3*t_{wb_su}^3$

"Calculate the water side heat capacity flow rate"

$C_{dot_w}=c_{p_w}*M_{dot_w}$

"Calculate the air-water heat transfer"

$Q_{dot_n}=C_{dot_w}*(t_{w_su}-t_{w_ex})$

"1st guess value of the exhaust air wet-bulb temperature"

$t_{wb_ex}=t_{wb_su}+5[C]$

10:

"Calculate the exhaust moist air enthalpy"

$h_{a_ex}=C_0+C_1*t_{wb_ex}+C_2*t_{wb_ex}^2+C_3*t_{wb_ex}^3$

"Calculate the fictitious specific heat and the fictitious fluid heat capacity flow rate"

$c_{p_af}=(h_{a_ex}-h_{a_su})/(t_{wb_ex}-t_{wb_su})$

$C_{dot_af}=c_{p_af}*m_{dot_a}$

"Experimental wet-bulb temperature"

$t_{wb_exp}=t_{wb_ex}$

"Re-calculate the exhaust air wet-bulb temperature"

$t_{wb_ex}=t_{wb_su}+(Q_{dot_n}/C_{dot_af})$

"Convergence test"

$Err_Rel=abs((t_{wb_ex}-t_{wb_exp})/t_{wb_exp})$

IF(Err_Rel>0.001) THEN

GOTO 10

ENDIF

"Calculate the fictitious fluid heat transfer coefficient"

$DT_{lm}=((t_{w_su}-t_{wb_ex})-(t_{w_ex}-t_{wb_su}))/ln((t_{w_su}-t_{wb_ex})/(t_{w_ex}-t_{wb_su}))$

$AU_f=Q_{dot_n}/DT_{lm}$

"Calculate the number of transfer units"

$C_{dot_min} = \min (C_{dot_w}, C_{dot_af})$

$C_{\dot{max}} = \max(C_{\dot{w}}, C_{\dot{af}})$

$NTU_f = AU_f / C_{\dot{min}}$

"Calculate the actual heat transfer coefficient"

$AU_n = AU_f * c_{p_{as}} / c_{p_{af}}$

END

"DETERMINATION OF THE MOISTURE CONTENT OF DEHYDRATED AIR LEAVING ABSORBER"

PROCEDURE DCCTAIRCON

(M_dot_a, M_ratio, w_s_su, wt_g, A, B, gamma, rho_g, t_a, X_dehyd: RH_dehyd, M_dot_a_dehyd, M_dot_g_hr)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std, R, T_std, MW_s

"Assumption"

"1. Maximum air flow per cooling tower cell for actual condition is same as design condition"

"Calculate mol fraction water in lean TEG"

$X_0 = \frac{((100 - wt_g) / 18)}{((100 - wt_g) / 18) + (wt_g / 150)}$

"Calculate dehydrated air flow rate"

$M_{\dot{a_dehyd}} = X_{dehyd} * M_{\dot{a}}$

"Calculate standard volumetric flow rate of air per hour"

$V_{\dot{a_std}} = (M_{\dot{a_dehyd}} / \rho_{a_std}) * 3600$

"Calculate molar flow rate of air per hour"

$n_{\dot{a}} = (P_{atm} * V_{\dot{a_std}}) / (R * T_{std})$

"Calculate moisture content of saturated inlet air (kg) per kg of dry air"

$w_{su} = w_{s_su} / \rho_a$

"Calculate TEG-water system equilibrium constant"

$K_{eq} = (w_{s_su} * 10^6 * \gamma) / B$

{ NB: $w_{s_su} * 10^6$ is moisture content of saturated inlet air per million standard cubic meter dry air }

"Calculate molar flow rate of lean TEG entering absorber per hour"

$n_{\dot{g}} = A * K_{eq} * n_{\dot{a}}$

"Calculate molecular weight of lean TEG"

$MW_g = (18 * X_0) + (150 * (1 - X_0))$

"Calculate equivalent mass flow rate of TEG per hour"

$M_{\dot{g_hr}} = n_{\dot{g}} * MW_g$

"Calculate equivalent volume flow rate of TEG in liter per hour"

$V_{\dot{g}} = M_{\dot{g_hr}} / \rho_g$

"Calculate mass of water absorbed per hour"

$M_{\dot{as}} = V_{\dot{g}} / M_{ratio}$

"Calculate water content of absorber outlet air (kg) per m³ of dry air"

```

w_a_ex=w_s_su-(M_dot_as/V_dot_a_std)
"Calculate water content of absorber outlet air (kg) per kg of dry air"
w_ex=w_a_ex/rho_a
"Calculate water content of air in equilibrium with lean TEG to absorber"
w_0=w_s_su*gamma*X_0
"Calculate absorber efficiency"
Eff_ab=(w_s_su-w_a_ex)/(w_s_su-W_0)
"Calculate relative humidity of outlet air from absorber"
RH_dehyd=RelHum(AIRH2O,T=t_a,w=w_ex,P=P_atm)
END

```

"DETERMINATION OF THE MOISTURE CONTENT OF MIXED DEHYDRATED + UNDEHYDRATED AIR ENTERING COOLING TOWER"

PROCEDURE DCCTMIXING

```

(t_a_dehyd,t_a_undehyd,RH_undehyd,RH_dehyd,M_dot_a_dehyd,M_dot_a:RH_mix,t_wb_
mix,t_a_mix)
$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s
"Calculate dehydrated air properties"
h_dehyd=ENTHALPY(AIRH2O, T=t_a_dehyd,R=RH_dehyd,P=p_atm)
w_dehyd=HUMRAT(AIRH2O, T=t_a_dehyd,R=RH_dehyd,P=p_atm)
t_wb_dehyd=WETBULB(AIRH2O, T=t_a_dehyd,R=RH_dehyd,P=p_atm)
"Calculate undehydrated air properties"
h_undehyd=ENTHALPY(AIRH2O, T=t_a_undehyd,R=RH_undehyd,P=p_atm)
w_undehyd=HUMRAT(AIRH2O, T=t_a_undehyd,R=RH_undehyd,P=p_atm)
t_wb_undehyd=WETBULB(AIRH2O, T=t_a_undehyd,R=RH_undehyd,P=p_atm)
"Calculate undehydrated air flow rate from air mass balance"
M_dot_a_undehyd=M_dot_a-M_dot_a_dehyd
"Calculate mixed air (dehydrated + undehydrated) enthalpy from energy balance"
h_mix=(M_dot_a_undehyd*h_undehyd+M_dot_a_dehyd*h_dehyd)/M_dot_a
"Calculate mixed air moisture content from water vapour mass balance"
w_mix=(M_dot_a_undehyd*w_undehyd+M_dot_a_dehyd*w_dehyd)/M_dot_a
"Calculate mixed air wet-bulb temperature"
t_wb_mix=WETBULB(AIRH2O,h=h_mix,w=w_mix,P=p_atm)
"Calculate mixed air relative humidity"
RH_mix=RELHUM(AIRH2O,h=h_mix,w=w_mix,P=p_atm)
"Calculate mixed air dry-bulb temperature"
t_a_mix=TEMPERATURE(AIRH2O,h=h_mix,w=w_mix,P=p_atm)
END

```

"DETERMINATION OF THE LEAVING WATER TEMPERATURE AND THE LEAVING AIR WET-BULB TEMPERATURE (TOWER SIMULATION)"

PROCEDURE DCCTSIM

(M_dot_w_su,t_w_su,t_w_ref,t_a_su,M_dot_a,RH_mix,AU_n,M_dot_a_n,M_dot_w_n,C_0,C_1,C_2,C_3:Q_dot,t_w_ex,t_wb_ex)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Assumption"

"1. Initial guess of exhaust wet-bulb temperature"

"2. Hypothesis: saturated air at CT exhaust"

RH_ex=1

"Calculate supply air wetbulb temperature"

RH_su=RH_mix

t_wb_su=WETBULB(AIRH2O,T=t_a_su,P=P_atm,R=RH_su)

"Calculate the supply moist air enthalpy"

h_a_su=C_0+C_1*t_wb_su+C_2*t_wb_su^2+C_3*t_wb_su^3

"Calculate supply air humidity ratio"

w_su=HumRat(AirH2O,h=h_a_su,R=RH_su,P=P_atm)

"Calculate the water side heat capacity flow rate"

C_dot_w=c_p_w*M_dot_w_su

"1st guess value of the exhaust air wet-bulb temperature"

t_wb_ex=t_wb_su+5[C]

10:

"Calculate the exhaust moist air enthalpy"

h_a_ex=C_0+C_1*t_wb_ex+C_2*t_wb_ex^2+C_3*t_wb_ex^3

"Calculate the fictitious specific heat and the fictitious fluid heat capacity flow rate"

c_p_af=(h_a_ex-h_a_su)/(t_wb_ex-t_wb_su)

C_dot_af=c_p_af*m_dot_a

"Calculate the ratio of the minimum heat capacity flow rate to the maximum one"

C_dot_min =min (C_dot_w, C_dot_af)

C_dot_max =max (C_dot_w, C_dot_af)

C_dot_ratio=C_dot_min/C_dot_max

"Calculate the actual heat transfer coefficient"

m=0.43

n=1.03

AU=AU_n*(M_dot_w_su/M_dot_w_n)^m*(M_dot_a/M_dot_a_n)^n " according to

J.Lebrun - (the given values for the exponents m & n are best fit values)"

"Calculate the fictitious heat transfer coefficient and the number of transfer units"

```

AU_f=AU*c_p_af/c_p_as
NTU=AU_f/C_dot_min
"Calculate the effectiveness of the heat exchanger"
IF(C_dot_ratio<0.995) THEN
Eff=(1-EXP(-NTU*(1-C_dot_ratio)))/(1-C_dot_ratio*EXP(-NTU*(1-C_dot_ratio)))
ELSE
Eff=NTU/(1+NTU)
ENDIF
"Calculate the air-water heat transfer"
Q_dot=Eff*C_dot_min*(t_w_su-t_wb_su)
"Experimental wet-bulb temperature"
t_wb_exp=t_wb_ex
"Re-calculate the exhaust air wet-bulb temperature"
t_wb_ex=t_wb_su+(Q_dot/C_dot_af)
"Convergence test"
Err_Rel=abs((t_wb_ex-t_wb_exp)/t_wb_exp)
IF(Err_Rel>0.001) THEN
GOTO 10
ENDIF
"Calculate the exhaust water temperature"
t_w_ex=t_w_su-Q_dot/C_dot_w
"Calculate exhaust air temperature"
t_a_ex=Temperature(AirH2O,h=h_a_ex,R=1,P=P_atm)
"Calculate exhaust air humidity ratio"
w_ex=HumRat(AirH2O,h=h_a_ex,R=1,P=P_atm)
"Calculate water evaporation rate"
M_dot_w_consumed=M_dot_a*(w_ex-w_su)
"Calculate exhaust water flow rate"
M_dot_w_ex=M_dot_w_su-M_dot_w_consumed
"Calculate the exhaust water temperature"
t_w_ex=((M_dot_w_su*c_p_w*(t_w_su-t_w_ref)-Q_dot)/(M_dot_w_ex*c_p_w))+t_w_ref
END

```

"DETERMINATION OF TEMPERATURE RISE OF LEAVING WATER IN COOLING WATER CIRCULATING PUMP"

PROCEDURE PUMPDELTA

(power_pump,pump_efficiency,m_dot_w_pump,c_p_w:t_delta_pump)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Calculate the cooling water temperature rise in circulation pump"

$t_delta_pump = ((power_pump * (1 - pump_efficiency)) / (c_p_w * m_dot_w_pump))$

END

!"ENERGY ANALYSIS FOR AIR CONDITIONING UNIT"

"DETERMINATION OF GLYCOL REBOILER SPECIFICATIONS"

PROCEDURE RBSPECS

(m_dot_g, m_dot_h, t_g_su_rb, t_g_ex_rb, t_h_su_rb, rho_h, rho_g, vel_h, c_p_h, tube_OD_rb, tube_TK_rb, Q_flux_rb, Q_rb:Q_dot_rb, t_h_ex_rb, AU_rb)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std, R, T_std, MW_s

"Calculate the equivalent glycol flow rate in liter/s"

$m_dot_g_eqv = m_dot_g / rho_g$

"Calculate the reboiler heat transfer rate"

$Q_dot_rb = Q_rb * m_dot_g_eqv$

"Calculate the heat transfer fluid (HTF) outlet temperature"

$t_h_ex_rb = (t_h_su_rb - ((Q_dot_rb) / (m_dot_h * c_p_h)))$

"Calculate the heat transfer surface area"

$area_s_rb = Q_dot_rb / Q_flux_rb$

"Calculate the log mean temperature difference"

$DTm_rb = (((t_h_su_rb - t_g_ex_rb) - (t_h_ex_rb - t_g_su_rb)) / LN((t_h_su_rb - t_g_ex_rb) / (t_h_ex_rb - t_g_su_rb)))$

"Calculate the overall heat transfer coefficient"

$U_rb = Q_dot_rb / (area_s_rb * DTm_rb)$

"Calculate tube inside diameter"

$tube_ID_rb = tube_OD_rb - (2 * tube_TK_rb)$

"Calculate the cross-sectional area per tube"

$area_s_tube = ((3.142 * tube_ID_rb^2) / 4)$

"Calculate total cross-sectional area of reboiler tubes"

$area_tot_tube = m_dot_h / (rho_h * vel_h)$

"Calculate the number of tubes required in reboiler"

$N_tube = area_tot_tube / area_s_tube$

"Calculate the length of tube required"

$L_tube = area_s_rb / (N_tube * 3.142 * tube_OD_rb)$

"Calculate the heat transfer coefficient-area product"

$AU_rb = U_rb * area_s_rb$

END

"DETERMINATION OF LEAN-RICH EXCHANGER SPECIFICATIONS"

PROCEDURE LNEXCHSPECS

(t_su_rich_g,t_ex_rich_g,t_su_lean_g,c_p_g,m_dot_g:Q_dot_exchanger,t_ex_lean_g)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Calculate the heat exchanger approach temperature" $t_{app_exch}=t_{su_lean_g}-t_{ex_rich_g}$ **"Calculate the exhaust temperature of lean glycol"** $t_{ex_lean_g}=t_{su_rich_g}+t_{app_exch}$ **"Calculate the heat transfer"** $Q_{dot_exchanger}=m_{dot_g}*c_{p_g}*(t_{su_lean_g}-t_{ex_lean_g})$

END

"DETERMINATION OF LEAN COOLER SPECIFICATIONS"

PROCEDURE LNCOOLSPECS

(t_app_cooler,t_su_lean_g_cooler,t_a,c_p_g,m_dot_g:Q_dot_cooler,t_ex_lean_g_cooler)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Calculate the lean glycol exhaust temperature" $t_{ex_lean_g_cooler}=t_a+t_{app_cooler}$ **"Calculate the heat transfer"** $Q_{dot_cooler}=m_{dot_g}*c_{p_g}*(t_{su_lean_g_cooler}-t_{ex_lean_g_cooler})$

END

"DETERMINATION OF AIR BLOWER SPECIFICATIONS"

PROCEDURE AIRBLOWSPECS (rho_a,DP_a_absorber,eff,m_dot_a_dehyd:W_dot_blower)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Calculate the equivalent air volumetric flow rate" $q_{dot_flow_a}=m_{dot_a_dehyd}/rho_a$ **"Calculate the blower (fan) power"** $W_{dot_blower}=(DP_{a_absorber}*q_{dot_flow_a})/eff$

END

"DETERMINATION OF GLYCOL PUMP SPECIFICATIONS"

PROCEDURE PUMPSPECS

(acc_gravity,Head_pump,eff_pump,eff_motor,rho_g,m_dot_g:W_dot_pump)

\$COMMON P_atm, rho_a, c_p_a, c_p_s, rho_w, c_p_w, c_p_as, rho_a_std,R,T_std,MW_s

"Calculate equivalent glycol density in kg/m³" $rho_{g_cubic}=rho_g*1000$ **"Calculate equivalent glycol volumetric flow rate in m³/s"** $q_{dot_flow_g}=m_{dot_g}/rho_{g_cubic}$

"Calculate the pump power"

$W_{\text{dot_pump}} = (q_{\text{dot_flow_g}} \cdot \rho_{\text{g_cubic}} \cdot \text{acc_gravity} \cdot \text{Head_pump}) / (\text{eff_pump} \cdot \text{eff_motor})$

END

!"SENSITIVITY ANALYSIS - Cooling Water Temperature vs. Liquefied Natural Gas Production"

"DETERMINATION OF SENSITIVITY OF PRODUCTION TO COOLING WATER SUPPLY TEMPERATURE"

PROCEDURE SENSITIVITY (X:LNG_prod)

"Calculate Liquefied Natural Gas production variation with cooling water supply temperature"

IF (X <= 32) THEN

$\text{LNG_prod} = (-0.53 \cdot X) + 151.84$

ELSE

$\text{LNG_prod} = (-3.05 \cdot X) + 232.8$

ENDIF

END

!"TOTAL POWER CONSUMED BY AIR CONDITIONING UNIT"

$\text{energy_dehyd_total} = \text{SUM}(Q_{\text{dot_rb}}, Q_{\text{dot_cooler}}, W_{\text{dot_blower}}, W_{\text{dot_pump}})$

"DETERMINATION OF UNCERTAINTY OF THE CALCULATED COLD WATER TEMPERATURE"

"Assumptions"

"1. The uncertainty of the air flow rate is taken as +/- 50 kg/s while that of the ambient air relative humidity is taken as 0.05"

"2. The uncertainty of the calculated cold water temperature is based on the air flow rate and the relative humidity uncertainties only"

$U = \text{UncertaintyOf}(t_{\text{w_ex}})$

!"SIZING"

"Design / Norminal data (based on 8-cell tower)"

$M_{\text{dot_w_n}} = 7777.778$ [kg/s]

"design water mass flow rate to cooling tower"

$M_{\text{dot_a_n}} = \text{convert}(\text{m}^3/\text{h}, \text{m}^3/\text{s}) \cdot 15408000$ [m³/h] * rho_a

"design mass flow rate of air to cooling tower"

$t_{\text{w_su_n}} = 40.1$ [C]

"design temperature of water supply to cooling tower"

$t_{\text{w_ex_n}} = 32$ [C]

"design temperature of water exhaust from cooling tower"

```

t_wb_su_n=28 [C]
"design wet-bulb temperature of air supply to cooling tower"
C_0=9362.5 [J/kg]
"polynomial coefficient"
C_1=1786.1 [J/kg-K]
"polynomial coefficient"
C_2=11.35 [J/kg-K^2]
"polynomial coefficient"
C_3=0.98855 [J/kg-K^3]
"polynomial coefficient"
"Identification of parameters"
CALL DCCTSIING
(M_dot_w_n,M_dot_a_n,t_w_su_n,t_w_ex_n,t_wb_su_n,C_0,C_1,C_2,C_3:AU_n,Q_dot_n,
NTU_f)

"!AIR CONDITIONING"
{M_dot_a = convert(m^3/h,m^3/s) * 15408000 [m^3/h] * rho_a }
"mass flow rate of air to cooling tower (based on 8-cell tower)"
M_dot_a = convert(m^3/h,m^3/s) * 17334000 [m^3/h] * rho_a
"mass flow rate of air to cooling tower (based on 9-cell tower)"
M_ratio= 9.8 [liter/kg]
"glycol circulation ratio"
w_s_su=0.025878 [kg/m^3]
"moisture content (humidity ratio) saturated inlet air"
wt_g=98.8 [-]
"glycol concentration"
A=2.0 [-]
"absorber factor"
B=761000 [-]
"constant"
gamma=0.63 [-]
"activity coefficient for water concentration in lean glycol"
rho_g=1.12 [kg/liter]
"glycol density"
t_a=31.9 [C]
"temperature of ambient air"
X_dehyd=0.2 [-]
"dehydrated air fraction"
"Identification of parameters"

```

CALL DCCTAIRCON

(M_dot_a,M_ratio,w_s_su,wt_g,A,B,gamma,rho_g,t_a,X_dehyd:RH_dehyd,M_dot_a_dehyd,
M_dot_g_hr)

!MIXING

t_a_dehyd=t_a

"temperature of dehydrated air supply to cooling tower"

t_a_undehyd=t_a

"temperature of undehydrated air supply to cooling tower"

RH_undehyd=0.632 [-]

"relative humidity of undehydrated air supply to cooling tower"

"Identification of parameters"

CALL DCCTMIXING

(t_a_dehyd,t_a_undehyd,RH_undehyd,RH_dehyd,M_dot_a_dehyd,M_dot_a:RH_mix,t_wb_
mix,t_a_mix)

!SIMULATION

"Actual data (based on 9-cell tower)"

M_dot_w_su=8305.556 [kg/s]

"mass flow rate of water supply to cooling tower"

t_w_su=39.7 [C]

"temperature of water supply to cooling tower"

t_w_ref=0 [C]

"reference temperature for water"

t_a_su=t_a_mix

"temperature of air supply to cooling tower"

"Identification of parameters"

CALL DCCTSIM

(M_dot_w_su,t_w_su,t_w_ref,t_a_su,M_dot_a,RH_mix,AU_n,M_dot_a_n,M_dot_w_n,C_0,C
_1,C_2,C_3:Q_dot,t_w_ex,t_wb_ex)

!COOLING WATER CIRCULATION PUMP

"Calculation data"

power_pump=2200000 [W]

"cooling water circulation pump power"

pump_efficiency=0.87 [-]

"cooling water circulation pump efficiency"

$m_dot_w_pump = M_dot_w_su * 0.5$

"mass flow rate of cooling water through pump"

"Identification of parameters"

CALL PUMPDELTA (power_pump, pump_efficiency, m_dot_w_pump, c_p_w: t_delta_pump)

"!Fluid properties"

$P_atm = 100600$ [Pa]

"atmospheric pressure"

$T_std = 288.15$ [K]

"standard atmospheric temperature"

$MW_s = 18.02$ [kg/kmol]

"molecular weight of water vapour"

$c_p_a = 1005$ [J/kg-K]

"specific heat capacity of air"

$c_p_s = 1820$ [J/kg-K]

"specific heat capacity of water vapour"

$c_p_w = 4190$ [J/kg-K]

"specific heat capacity of water"

$c_p_as = 1025$ [J/kg-K]

"specific heat capacity of moist (humid) air"

$R = 8314$ [J/kmol-K]

"universal gas constant"

$\rho_a = 1.11$ [kg/m³]

"density of air"

$\rho_a_std = 1.225$ [kg/m³]

"density of air at standard temperature and pressure conditions"

$\rho_w = 1000$ [kg/m³]

"density of water"

"!REBOILER"

"Calculation data"

$m_dot_g = \text{convert}(\text{kg/h}, \text{kg/s}) * M_dot_g_hr$

"mass flow rate of glycol through reboiler"

$m_dot_h = 373$ [kg/s]

"mass flow rate of heat transfer fluid (HTF) through reboiler"

$t_g_su_rb = 184$ [C]

"temperature of glycol supply to reboiler"

$t_g_ex_rb = 204$ [C]

"temperature of glycol exhaust reboiler"

```

t_h_su_rb=235 [C]
"temperature of glycol supply to reboiler"
rho_h=681 [kg/m^3]
"density of HTF"
vel_h=2 [m/s] "velocity of water"
c_p_h=2907 [J/kg-K]
"specific heat capacity of HTF"
tube_OD_rb=0.01905 [m]
"reboiler tube outer diameter"
tube_TK_rb=0.00165 [m]
"reboiler tube thickness"
Q_flux_rb=24000 [W/m^2]
"heat flux of reboiler"
Q_rb=2.5E05 [J/liter]
"heat duty of reboiler (initial selection for design)"
"Identification of parameters"
CALL RBSPECS
(m_dot_g,m_dot_h,t_g_su_rb,t_g_ex_rb,t_h_su_rb,rho_h,rho_g,vel_h,c_p_h,tube_OD_rb,tu
be_TK_rb,Q_flux_rb,Q_rb:Q_dot_rb,t_h_ex_rb,AU_rb)

```

"!LEAN-RICH GLYCOL EXCHANGER"

```

"Calculation data"
t_su_rich_g=35 [C]
"temperature of rich glycol supply to exchanger"
t_ex_rich_g=t_g_su_rb
"temperature of rich glycol exhaust from exchanger"
t_su_lean_g=t_g_ex_rb
"temperature of lean glycol supply to exchanger"
c_p_g=2184 [J/kg-K]
"specific heat capacity of glycol"
"Identification of parameters"
CALL LNEXTSPECS
(t_su_rich_g,t_ex_rich_g,t_su_lean_g,c_p_g,m_dot_g:Q_dot_exchanger,t_ex_lean_g)

```

"!LEAN GLYCOL COOLER"

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"Calculation data"
t_app_cooler=5 [C]
"approach temperature of cooler"

```

t_su_lean_g_cooler=t_ex_lean_g
 "temperature of lean glycol supply to cooler"
 "Identification of parameters"
 CALL LNCOOLSPECS
 (t_app_cooler,t_su_lean_g_cooler,t_a,c_p_g,m_dot_g:Q_dot_cooler,t_ex_lean_g_cooler)

!"AIR BLOWER"

"Calculation data"
 DP_a_absorber=200 [Pa]
 "differential pressure across absorber"
 eff=0.75 [-]
 "efficiency of blower"
 "Identification of parameters"
 CALL AIRBLOWSPECS (rho_a,DP_a_absorber,eff,m_dot_a_dehyd:W_dot_blower)

!"GLYCOL PUMP"

"Calculation data"
 acc_gravity=9.81 [m/s²]
 "acceleration due to gravity"
 Head_pump=18 [m]
 "pump head"
 eff_pump=0.88 [-]
 "efficiency of pump"
 eff_motor=0.95 [-]
 "efficiency of electric motor (pump driver)"
 "Identification of parameters"
 CALL PUMPSPECS
 (acc_gravity,Head_pump,eff_pump,eff_motor,rho_g,m_dot_g:W_dot_pump)

!"SENSITIVITY"

"Calculation data"
 X=t_w_ex
 "temperature of cooling water exhaust from cooling tower"
 "Identification of parameters"
 CALL SENSITIVITY (X:LNG_prod)

!"ENERGY ANALYSIS FOR THE GAS LIQUEFACTION PLANT"

"Assumptions"

"1. Temperature of exhaust cooling water from all exchangers is same as the temperature of supply cooling water to the cooling tower"

"2. Values given are design values"

"!1. PROPANE REFRIGERANT CYCLE"

"!1.1. Inputs"

"!1.1.1 Natural Gas Inputs"

"Supply conditions"

$t_{ng_su_ev}=39$ [C]

"temperature of natural gas to propane evaporators"

$P_{ng_su_ev}=53.5E5$ [Pa]

"pressure of natural gas to propane evaporators"

$M_{dot_ng_ev}=142$ [kg/s]

"mass flow rate of natural gas through propane evaporators"

"!1.1.2 Cooling Water Inputs"

"Cooling water temperature"

$m_{dot_w_desup}=600$ [kg/s]

"mass flow rate of cooling water through propane desuperheater"

$t_{w_su_desup}=t_{w_ex}$

"temperature of cooling water supply to propane desuperheater"

$t_{w_ex_desup}=t_{w_su}$

"temperature of cooling water exhaust from propane desuperheater"

$t_{w_su_cd}=t_{w_ex}$

"temperature of cooling water supply to propane condenser"

$t_{w_ex_cd}=t_{w_su}$

"temperature of cooling water supply from propane condenser"

"!1.1.3 Mixed Refrigerant Inputs"

"Mixed Refrigerant (mr) common Inputs"

$M_{dot_mr_ev} = 274$ [kg/s]

"mass flow rate of mixed refrigerant through propane mr_evaporators"

$p_{mr_ev} = 48E5$ [Pa]

"pressure of mixed refrigerant to propane mr_evaporators"

"Molar fractions of the components of mixed refrigerant"

$X_{N2} = 0.084$

"mol fraction of nitrogen in mixed refrigerant"

$X_{CH4} = 0.383$

"mol fraction of propane in mixed refrigerant"

$X_{C2H6} = 0.411$
 "mol fraction of ethane in mixed refrigerant"
 $X_{C3H8} = 0.122$
 "mol fraction of butane in mixed refrigerant"
 "Molar masses of the components of mixed refrigerant"
 $MM_{N2} = \text{molarmass}(\text{nitrogen})$
 "molecular weight of nitrogen"
 $MM_{CH4} = \text{molarmass}(\text{methane})$
 "molecular weight of propane"
 $MM_{C2H6} = \text{molarmass}(\text{ethane})$
 "molecular weight of ethane"
 $MM_{C3H8} = \text{molarmass}(\text{propane})$
 "molecular weight of butane"
 "Mass flow rates of the components of mixed refrigerant"
 $M_{\dot{N}2} = M_{\dot{m}r_ev} * X_{N2}$
 "mass flow rate of nitrogen in mixed refrigerant"
 $M_{\dot{C}H4} = M_{\dot{m}r_ev} * X_{CH4}$
 "mass flow rate of propane in mixed refrigerant"
 $M_{\dot{C}2H6} = M_{\dot{m}r_ev} * X_{C2H6}$
 "mass flow rate of ethane in mixed refrigerant"
 $M_{\dot{C}3H8} = M_{\dot{m}r_ev} * X_{C3H8}$
 "mass flow rate of butane in mixed refrigerant"
 $M_{\dot{m}r_ev_bis} = \text{SUM}(M_{\dot{N}2}, M_{\dot{C}H4}, M_{\dot{C}2H6}, M_{\dot{C}3H8})$
 "total mass flow rate of mixed refrigerant"
 "HHP mr_evaporator inputs"
 $c_{p_mr_HHP_ev} = 2992 \text{ [J/kg-K]}$
 "specific heat capacity of mixed refrigerant in HHP propane mr_evaporator"
 $t_{mr_su_HHP_ev} = 35 \text{ [C]}$
 "temperature of mixed refrigerant supply to HHP propane mr_evaporator"
 $t_{mr_ex_HHP_ev} = 11.5 \text{ [C]}$
 "temperature of mixed refrigerant exhaust from HHP propane mr_evaporator"
 "HP mr_evaporator inputs"
 $c_{p_mr_HP_ev} = 3077 \text{ [J/kg-K]}$
 "specific heat capacity of mixed refrigerant in HP propane mr_evaporator"
 $t_{mr_su_HP_ev} = 11.5 \text{ [C]}$
 "temperature of mixed refrigerant supply to HP propane mr_evaporator"
 $t_{mr_ex_HP_ev} = -4.5 \text{ [C]}$
 "temperature of mixed refrigerant exhaust from HP propane mr_evaporator"
 "MP mr_evaporator inputs"

$c_{p_mr_MP_ev}=3010$ [J/kg-K]
 "specific heat capacity of mixed refrigerant in MP propane mr_evaporator"
 $t_{mr_su_MP_ev}=-4.5$ [C]
 "temperature of mixed refrigerant supply to MP propane mr_evaporator"
 $t_{mr_ex_MP_ev}=-18.8$ [C]
 "temperature of mixed refrigerant exhaust from MP propane mr_evaporator"
 "LP mr_evaporator inputs"
 $c_{p_mr_LP_ev}=2963$ [J/kg-K]
 "specific heat capacity of mixed refrigerant in LP propane mr_evaporator"
 $t_{mr_su_LP_ev}=-18.8$ [C]
 "temperature of mixed refrigerant supply to LP propane mr_evaporator"
 $t_{mr_ex_LP_ev}=-31.6$ [C]
 "temperature of mixed refrigerant exhaust from LP propane mr_evaporator"

!1.2. Outputs

"propane refrigerant compressor mechanical power"
 "propane refrigeration cycle coefficient of performance (COP)"

!1.3. Parameters

$fluid1\$='propane'$
 $fluid2\$='methane'$
 "Compressor"
 $epsilon_s_HHP_cp=0.76$
 "isentropic efficiency of High High Pressure stage of propane refrigerant compressor"
 $epsilon_s_HP_cp=0.74$
 "isentropic efficiency of High Pressure stage of propane refrigerant compressor"
 $epsilon_s_MP_cp=0.77$
 "isentropic efficiency of Medium Pressure stage of propane refrigerant compressor"
 $epsilon_s_LP_cp=0.82$
 "isentropic efficiency of Low Pressure stage of propane refrigerant compressor"
 "Heat exchangers (evaporators and condenser)"
 $AU_HHP_ev_n=918080$ [W/K]
 "heat transfer coefficient-area product of High High Pressure propane evaporator"
 $AU_HP_ev_n=918136$ [W/K]
 "heat transfer coefficient-area product of High Pressure evaporator"
 $AU_MP_ev_n=816888$ [W/K]
 "heat transfer coefficient-area product of Medium Pressure evaporator"
 $AU_LP_ev_n=762420$ [W/K]
 "heat transfer coefficient-area product of Low Pressure evaporator"

AU_mr_HHP_ev=2088394 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_HP_ev=4388696 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_MP_ev=3390400 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_LP_ev=2736708 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_desup=1022200 [W/K]
 "heat transfer coefficient-area product of propane desuperheater"
 AU_cd=35613420 [W/K]
 "heat transfer coefficient-area product of condenser"
 DELTAt_sc=8.5
 "subcooling (at outlet of condenser)"
 DELTAt_oh=5
 "superheating (at outlet of evaporators)"
 "Control (boundary conditions)"
 t_ng_ex_HHP_ev=21 [C]
 "temperature of natural gas exhaust from High High Pressure propane evaporator"
 t_ng_ex_HP_ev=-4.5 [C]
 "temperature of natural gas exhaust from High Pressure propane evaporator"
 t_ng_ex_MP_ev=-18.8 [C]
 "temperature of natural gas exhaust from Medium Pressure propane evaporator"
 t_ng_ex_LP_ev=-29.7 [C]
 "temperature of natural gas exhaust from Low Pressure propane evaporator"

 "Note: x is the propane quality (vapour mass fraction) in the condenser. x=1 denotes saturated vapour state and x=0 denotes saturated liquid state"

!1.4. Equations

!Desuperheater

"supply"

T_r_su_desup=t[1]

"temperature of propane refrigerant supply to desuperheater"

h_r_su_desup=ENTHALPY(fluid1\$,T=T_r_su_desup,P=P_cd)

"enthalpy of propane refrigerant supply to desuperheater"

"refrigerant side"

T_r_ex_desup=TEMPERATURE(fluid1\$,P=P_cd,x=1)

"temperature of propane refrigerant exhaust from desuperheater"

$h_{r_ex_desup} = \text{ENTHALPY}(\text{fluid1}\$, T=T_{r_ex_desup}, x=1)$

"enthalpy of propane refrigerant exhaust from desuperheater"

$Q_{dot_desup} = m_{dot_r_HHP_cp} * (h_{r_su_desup} - h_{r_ex_desup})$

"heat duty of desuperheater"

"!Condenser"

"supply"

$h_{r_su_cd} = h_{r_ex_desup}$

"enthalpy of propane refrigerant supply to condenser"

$T_{r_su_cd} = T_{r_ex_desup}$

"temperature of propane refrigerant supply to condenser"

"refrigerant side"

$Q_{dot_cd} = m_{dot_r_HHP_cp} * (h_{r_su_cd} - h_{r_ex_cd})$

"heat duty of condenser"

$T_{r_ex_cd} = T_{cd} - \text{DELTA}t_{sc}$

"temperature of propane refrigerant exhaust from condenser (including subcooling)"

$T_{cd} = \text{TEMPERATURE}(\text{fluid1}\$, P=P_{cd}, x=0)$

"temperature of propane refrigerant exhaust from condenser (excluding subcooling)"

$h_{r_ex_cd} = \text{ENTHALPY}(\text{fluid1}\$, P=P_{cd}, T=T_{r_ex_cd})$

"enthalpy of propane refrigerant exhaust from condenser" {need to fix subcooling}

"water side"

$Q_{dot_cd} = m_{dot_w_cd} * c_{p_w} * (t_{w_ex_cd} - t_{w_su_cd})$

"heat duty of condenser"

"heat exchange effectiveness"

$Q_{dot_cd} = m_{dot_w_cd} * c_{p_w} * \epsilon_{cd} * (t_{cd} - t_{w_su_cd})$

"heat duty of condenser"

$\epsilon_{cd} = 1 - \exp(-\text{NTU}_{cd})$

"effectiveness of condenser"

$\text{NTU}_{cd} = \text{AU}_{cd} / (m_{dot_w_cd} * c_{p_w})$

"number of transfer units of condenser"

"!HHP Expansion valve"

$h_{r_ex_HHP_expv} = h_{r_ex_cd}$

"enthalpy of propane refrigerant exhaust from HHP expansion valve"

"!HHP evaporator"

"supply"

$h_{r_su_HHP_ev} = h_{r_ex_HHP_expv}$

"enthalpy of propane refrigerant supply to HHP evaporator"

$t_{ng_su_HHP_ev}=t_{ng_su_ev}$
 "temperature of natural gas supply to HHP evaporator"
 $P_{ng_su_HHP_ev}=P_{ng_su_ev}$
 "pressure of natural gas supply to HHP evaporator"
 "refrigerant side"
 $Q_{dot_HHP_ev}=m_{dot_r_HHP_ev}*(h_{r_ex_HHP_ev}-h_{r_su_HHP_ev})$
 "heat duty of HHP evaporator"
 $h_{r_ex_HHP_ev}=ENTHALPY(fluid1$,P=P_{HHP_ev},t=t_{r_ex_HHP_ev})$
 "enthalpy of propane refrigerant exhaust from HHP evaporator"
 $t_{r_ex_HHP_ev}=t_{HHP_ev}+DELTA_{t_oh}$
 "temperature of propane refrigerant exhaust from HHP evaporator (including superheat)"
 $t_{HHP_ev}=TEMPERATURE(fluid1$,P=P_{HHP_ev},x=1)$
 "temperature of propane refrigerant exhaust from HHP evaporator (excluding superheat)"
 "natural gas side"
 $Q_{dot_HHP_ev}=m_{dot_ng_ev}*(h_{ng_su_HHP_ev}-h_{ng_ex_HHP_ev})$
 "heat duty of HHP evaporator"
 $P_{ng_ex_HHP_ev}=P_{ng_su_HHP_ev}$
 "pressure of natural gas exhaust from HHP evaporator (no head loss)"
 $h_{ng_su_HHP_ev}=ENTHALPY(fluid2$,t=t_{ng_su_HHP_ev},P=P_{ng_su_HHP_ev})$
 "enthalpy of natural gas supply to HHP evaporator"
 "heat exchange effectiveness"
 $Q_{dot_HHP_ev}=m_{dot_ng_ev}*c_{p_ng_HHP_ev}*\epsilon_{HHP_ev}*(t_{ng_su_HHP_ev}-t_{HHP_ev})$ "heat duty of HHP evaporator"
 $c_{p_ng_HHP_ev}=CP(fluid2$,t=t_{ng_su_HHP_ev},P=P_{ng_su_HHP_ev})$
 "specific heat of natural gas supply to HHP evaporator"
 $\epsilon_{HHP_ev}=1-\exp(-NTU_{HHP_ev})$
 "effectiveness of HHP evaporator (semi-isothermal heat exchange)"
 $NTU_{HHP_ev}=AU_{HHP_ev}/(m_{dot_ng_ev}*c_{p_ng_HHP_ev})$
 "number of transfer units of HHP evaporator"
 $AU_{HHP_ev}=AU_{HHP_ev_n}$ "heat transfer coefficient-area product of HHP propane evaporator (modification needed to take effect of flow rate variations)"
 $t_{ng_ex_HHP_ev}=TEMPERATURE(fluid2$,h=h_{ng_ex_HHP_ev},P=P_{ng_ex_HHP_ev})$
 "temperature of natural gas exhaust from HHP evaporator"

 "!HHP mr_evaporator"
 "supply"
 $h_{r_su_HHP_mr_ev}=h_{r_ex_HHP_expv}$
 "enthalpy of propane refrigerant supply to HHP mr_evaporator"
 "refrigerant side"

$$Q_dot_HHP_mr_ev = m_dot_r_HHP_mr_ev * (h_r_ex_HHP_mr_ev - h_r_su_HHP_mr_ev)$$

"heat duty of HHP mr_evaporator"

$$h_r_ex_HHP_mr_ev = ENTHALPY(fluid1$, P=P_HHP_mr_ev, t=t_r_ex_HHP_mr_ev)$$

"enthalpy of propane refrigerant exhaust from HHP mr_evaporator"

$$P_HHP_mr_ev = P_HHP_ev - 1.8E5 \text{ [Pa]}$$

"pressure of natural gas supply to HHP evaporator"

$$t_r_ex_HHP_mr_ev = t_HHP_mr_ev + \Delta T_{oh}$$

"temperature of propane refrigerant exhaust from HHP mr_evaporator (including superheat)"

$$t_HHP_mr_ev = TEMPERATURE(fluid1$, P=P_HHP_mr_ev, x=1)$$

"temperature of propane refrigerant exhaust from HHP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

$$h_N2_su_HHP_ev = \text{enthalpy}(\text{nitrogen}, T=t_mr_su_HHP_ev, p=p_mr_ev)$$

"enthalpy of nitrogen component of mixed refrigerant supply to HHP mr_evaporator"

$$h_CH4_su_HHP_ev = \text{enthalpy}(\text{methane}, T=t_mr_su_HHP_ev, p=p_mr_ev)$$

"enthalpy of methane component of mixed refrigerant supply to HHP mr_evaporator"

$$h_C2H6_su_HHP_ev = \text{enthalpy}(\text{ethane}, T=t_mr_su_HHP_ev, p=p_mr_ev)$$

"enthalpy of ethane component of mixed refrigerant supply to HHP mr_evaporator"

$$h_C3H8_su_HHP_ev = \text{enthalpy}(\text{propane}, T=t_mr_su_HHP_ev, p=p_mr_ev)$$

"enthalpy of butane component of mixed refrigerant supply to HHP mr_evaporator"

"Calculation of the exhaust enthalpies:"

$$h_N2_ex_HHP_ev = \text{enthalpy}(\text{nitrogen}, T=t_mr_ex_HHP_ev, p=p_mr_ev)$$

"enthalpy of nitrogen component of mixed refrigerant exhaust from HHP mr_evaporator"

$$h_CH4_ex_HHP_ev = \text{enthalpy}(\text{methane}, T=t_mr_ex_HHP_ev, p=p_mr_ev)$$

"enthalpy of methane component of mixed refrigerant exhaust from HHP mr_evaporator"

$$h_C2H6_ex_HHP_ev = \text{enthalpy}(\text{ethane}, T=t_mr_ex_HHP_ev, p=p_mr_ev)$$

"enthalpy of ethane component of mixed refrigerant exhaust from HHP mr_evaporator"

$$h_C3H8_ex_HHP_ev = \text{enthalpy}(\text{propane}, T=t_mr_ex_HHP_ev, p=p_mr_ev)$$

"enthalpy of butane component of mixed refrigerant exhaust from HHP mr_evaporator"

"Calculate evaporator duty"

$$Q_dot_N2_HHP_ev = M_dot_N2 * (h_N2_su_HHP_ev - h_N2_ex_HHP_ev)$$

"heat duty of nitrogen component of mixed refrigerant HHP mr_evaporator"

$$Q_dot_CH4_HHP_ev = M_dot_CH4 * (h_CH4_su_HHP_ev - h_CH4_ex_HHP_ev)$$

"heat duty of methane component of mixed refrigerant HHP mr_evaporator"

$$Q_dot_C2H6_HHP_ev = M_dot_C2H6 * (h_C2H6_su_HHP_ev - h_C2H6_ex_HHP_ev)$$

"heat duty of ethane component of mixed refrigerant HHP mr_evaporator"

$$Q_dot_C3H8_HHP_ev = M_dot_C3H8 * (h_C3H8_su_HHP_ev - h_C3H8_ex_HHP_ev)$$

"heat duty of butane component of mixed refrigerant HHP mr_evaporator"

$Q_{\text{dot_HHP_mr_ev}} =$
 $\text{SUM}(Q_{\text{dot_N2_HHP_ev}}, Q_{\text{dot_CH4_HHP_ev}}, Q_{\text{dot_C2H6_HHP_ev}}, Q_{\text{dot_C3H8_HHP_ev}})$ "heat duty of HHP mr_evaporator"

!"HHP Compressor"

"Calculate exhaust enthalpy (based on energy balance at inlet of HHP compressor)"

$h_{\text{r_su_HHP_cp}} = (m_{\text{dot_r_HHP_ev}} * h_{\text{r_ex_HHP_ev}} + m_{\text{dot_r_HHP_mr_ev}} * h_{\text{r_ex_HHP_mr_ev}} + m_{\text{dot_r_HP_cp}} * h_{\text{r_ex_HP_cp}}) / m_{\text{dot_r_HHP_cp}}$

"mass balance at inlet of HHP compressor"

$m_{\text{dot_r_HHP_cp}} = m_{\text{dot_r_HHP_ev}} + m_{\text{dot_r_HHP_mr_ev}} + m_{\text{dot_r_HP_cp}}$

"mass flow rate of propane refrigerant through HHP compressor"

$s_{\text{r_su_HHP_cp}} = \text{ENTROPY}(\text{fluid1}, h = h_{\text{r_su_HHP_cp}}, P = P_{\text{HHP_ev}})$

"entropy of propane refrigerant supply to HHP compressor"

$h_{\text{r_ex_HHP_cp_s}} = \text{ENTHALPY}(\text{fluid1}, P = P_{\text{cd}}, s = s_{\text{r_su_HHP_cp}})$

"HHP compressor exhaust enthalpy at isentropic condition"

$\epsilon_{\text{s_HHP_cp}} = (h_{\text{r_ex_HHP_cp_s}} - h_{\text{r_su_HHP_cp}}) / (h_{\text{r_ex_HHP_cp}} - h_{\text{r_su_HHP_cp}})$ "isentropic efficiency of HHP compressor"

$W_{\text{dot_HHP_cp}} = m_{\text{dot_r_HHP_cp}} * (h_{\text{r_ex_HHP_cp}} - h_{\text{r_su_HHP_cp}})$

"HHP compressor work"

!"HP expansion valve"

$h_{\text{r_ex_HP_expv}} = h_{\text{r_ex_HHP_expv}}$

"enthalpy of propane refrigerant exhaust from HP expansion valve"

!"HP evaporator"

"supply"

$h_{\text{r_su_HP_ev}} = h_{\text{r_ex_HP_expv}}$

"enthalpy of propane refrigerant supply to HP evaporator"

$t_{\text{ng_su_HP_ev}} = t_{\text{ng_ex_HHP_ev}}$

"temperature of natural gas supply to HP evaporator""(no heat loss)"

$P_{\text{ng_su_HP_ev}} = P_{\text{ng_ex_HHP_ev}}$

"pressure of natural gas supply to HP evaporator""(no head loss)"

"refrigerant side"

$Q_{\text{dot_HP_ev}} = m_{\text{dot_r_HP_ev}} * (h_{\text{r_ex_HP_ev}} - h_{\text{r_su_HP_ev}})$

"heat duty of HP evaporator"

$h_{\text{r_ex_HP_ev}} = \text{ENTHALPY}(\text{fluid1}, P = P_{\text{HP_ev}}, t = t_{\text{r_ex_HP_ev}})$

"enthalpy of propane refrigerant exhaust from HP evaporator"

$t_{\text{r_ex_HP_ev}} = t_{\text{HP_ev}} + \text{DELTA}t_{\text{oh}}$

"temperature of propane refrigerant exhaust from HP evaporator (including superheat)"

$t_{HP_ev} = \text{TEMPERATURE}(\text{fluid1}\$, P=P_{HP_ev}, x=1)$
 "temperature of propane refrigerant exhaust from HP evaporator (excluding superheat)"
 "natural gas side"
 $Q_{dot_HP_ev} = m_{dot_ng_ev} * (h_{ng_su_HP_ev} - h_{ng_ex_HP_ev})$
 "heat duty of HP mr_evaporator"
 $P_{ng_ex_HP_ev} = P_{ng_su_HP_ev}$
 "pressure of natural gas exhaust from HP evaporator (no head loss)"
 $h_{ng_su_HP_ev} = \text{ENTHALPY}(\text{fluid2}\$, t=t_{ng_su_HP_ev}, P=P_{ng_su_HP_ev})$
 "enthalpy of natural gas supply to HP evaporator"
 "heat exchange effectiveness"
 $Q_{dot_HP_ev} = m_{dot_ng_ev} * c_{p_ng_HP_ev} * \epsilon_{HP_ev} * (t_{ng_su_HP_ev} - t_{HP_ev})$
 "heat duty of HP evaporator"
 $c_{p_ng_HP_ev} = \text{CP}(\text{fluid2}\$, t=t_{ng_su_HP_ev}, P=P_{ng_su_HP_ev})$
 "specific heat of natural gas supply to HP evaporator"
 $\epsilon_{HP_ev} = 1 - \exp(-NTU_{HP_ev})$
 "effectiveness of HP evaporator (semi-isothermal heat exchange)"
 $NTU_{HP_ev} = AU_{HP_ev} / (m_{dot_ng_ev} * c_{p_ng_HP_ev})$
 "number of transfer units of HP evaporator"
 $AU_{HP_ev} = AU_{HP_ev_n}$
 "heat transfer coefficient-area product of HP propane evaporator (modification needed to take effect of flow rate variations)"
 $t_{ng_ex_HP_ev} = \text{TEMPERATURE}(\text{fluid2}\$, h=h_{ng_ex_HP_ev}, P=P_{ng_ex_HP_ev})$
 "temperature of natural gas exhaust from HP evaporator"

"HP mr_evaporator"
 "supply"
 $h_{r_su_HP_mr_ev} = h_{r_ex_HP_expv}$
 "enthalpy of propane refrigerant supply to HP mr_evaporator"
 "refrigerant side"
 $Q_{dot_HP_mr_ev} = m_{dot_r_HP_mr_ev} * (h_{r_ex_HP_mr_ev} - h_{r_su_HP_mr_ev})$
 "heat duty of HP mr_evaporator"
 $h_{r_ex_HP_mr_ev} = \text{ENTHALPY}(\text{fluid1}\$, P=P_{HP_mr_ev}, t=t_{r_ex_HP_mr_ev})$
 "enthalpy of propane refrigerant exhaust from HP mr_evaporator"
 $P_{HP_mr_ev} = P_{HP_ev}$
 "pressure of natural gas supply to HP evaporator"
 $t_{r_ex_HP_mr_ev} = t_{HP_mr_ev} + \Delta T_{at_oh}$
 "temperature of propane refrigerant exhaust from HP mr_evaporator (including superheat)"
 $t_{HP_mr_ev} = \text{TEMPERATURE}(\text{fluid1}\$, P=P_{HP_mr_ev}, x=1)$
 "temperature of propane refrigerant exhaust from HP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

$h_{N2_su_HP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_su_HP_ev}, p=p_{mr_ev})$

"enthalpy of nitrogen component of mixed refrigerant supply to HP mr_evaporator"

$h_{CH4_su_HP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_su_HP_ev}, p=p_{mr_ev})$

"enthalpy of methane component of mixed refrigerant supply to HP mr_evaporator"

$h_{C2H6_su_HP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_su_HP_ev}, p=p_{mr_ev})$

"enthalpy of ethane component of mixed refrigerant supply to HP mr_evaporator"

$h_{C3H8_su_HP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_su_HP_ev}, p=p_{mr_ev})$

"enthalpy of butane component of mixed refrigerant supply to HP mr_evaporator"

"Calculation of the exhaust enthalpies:"

$h_{N2_ex_HP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_ex_HP_ev}, p=p_{mr_ev})$

"enthalpy of nitrogen component of mixed refrigerant exhaust from HP mr_evaporator"

$h_{CH4_ex_HP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_ex_HP_ev}, p=p_{mr_ev})$

"enthalpy of methane component of mixed refrigerant exhaust from HP mr_evaporator"

$h_{C2H6_ex_HP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_ex_HP_ev}, p=p_{mr_ev})$

"enthalpy of ethane component of mixed refrigerant exhaust from HP mr_evaporator"

$h_{C3H8_ex_HP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_HP_ev}, p=p_{mr_ev})$

"enthalpy of butane component of mixed refrigerant exhaust from HP mr_evaporator"

"Calculate evaporator duty"

$Q_{dot_N2_HP_ev} = M_{dot_N2} * (h_{N2_su_HP_ev} - h_{N2_ex_HP_ev})$

"heat duty of nitrogen component of mixed refrigerant HP mr_evaporator"

$Q_{dot_CH4_HP_ev} = M_{dot_CH4} * (h_{CH4_su_HP_ev} - h_{CH4_ex_HP_ev})$

"heat duty of methane component of mixed refrigerant HP mr_evaporator"

$Q_{dot_C2H6_HP_ev} = M_{dot_C2H6} * (h_{C2H6_su_HP_ev} - h_{C2H6_ex_HP_ev})$

"heat duty of ethane component of mixed refrigerant HP mr_evaporator"

$Q_{dot_C3H8_HP_ev} = M_{dot_C3H8} * (h_{C3H8_su_HP_ev} - h_{C3H8_ex_HP_ev})$

"heat duty of butane component of mixed refrigerant HP mr_evaporator"

$Q_{dot_HP_mr_ev} =$

$SUM(Q_{dot_N2_HP_ev}, Q_{dot_CH4_HP_ev}, Q_{dot_C2H6_HP_ev}, Q_{dot_C3H8_HP_ev})$

"heat duty of HHP mr_evaporator"

"!HP Compressor"

"Calculate exhaust enthalpy (based on energy balance at inlet of HP compressor)"

$h_{r_su_HP_cp} = (m_{dot_r_HP_ev} * h_{r_ex_HP_ev}$

$+ m_{dot_r_HP_mr_ev} * h_{r_ex_HP_mr_ev} + m_{dot_r_MP_cp} * h_{r_ex_MP_cp}) / m_{dot_r_HP_c}$

p

"mass balance at inlet of HP compressor"

$$m_{\dot{r}_{HP_cp}} = m_{\dot{r}_{HP_ev}} + m_{\dot{r}_{HP_mr_ev}} + m_{\dot{r}_{MP_cp}}$$

"mass flow rate of propane refrigerant through HP compressor"

$$s_{r_su_HP_cp} = \text{ENTROPY}(\text{fluid1}\$, h = h_{r_su_HP_cp}, P = P_{HP_ev})$$

"entropy of propane refrigerant supply to HP compressor"

$$h_{r_ex_HP_cp_s} = \text{ENTHALPY}(\text{fluid1}\$, P = P_{HHP_ev}, s = s_{r_su_HP_cp})$$

"HP compressor exhaust enthalpy at isentropic condition"

$$\epsilon_{s_HP_cp} = (h_{r_ex_HP_cp_s} - h_{r_su_HP_cp}) / (h_{r_ex_HP_cp} - h_{r_su_HP_cp})$$

"isentropic efficiency of HP compressor"

$$W_{\dot{HP_cp}} = m_{\dot{r}_{HP_cp}} * (h_{r_ex_HP_cp} - h_{r_su_HP_cp})$$

"HP compressor work"

!"MP expansion valve"

$$h_{r_ex_MP_expv} = h_{r_ex_HP_expv}$$

"enthalpy of propane refrigerant exhaust from MP expansion valve"

!"MP evaporator"

"supply"

$$h_{r_su_MP_ev} = h_{r_ex_MP_expv}$$

"enthalpy of propane refrigerant supply to MP evaporator"

$$t_{ng_su_MP_ev} = t_{ng_ex_HP_ev}$$

"temperature of natural gas supply to MP evaporator" "(no heat loss)"

$$P_{ng_su_MP_ev} = P_{ng_ex_HP_ev}$$

"pressure of natural gas supply to MP evaporator" "(no head loss)"

"refrigerant side"

$$Q_{\dot{MP_ev}} = m_{\dot{r}_{MP_ev}} * (h_{r_ex_MP_ev} - h_{r_su_MP_ev})$$

"heat duty of MP evaporator"

$$h_{r_ex_MP_ev} = \text{ENTHALPY}(\text{fluid1}\$, P = P_{MP_ev}, t = t_{r_ex_MP_ev})$$

"enthalpy of propane refrigerant exhaust from MP evaporator"

$$t_{r_ex_MP_ev} = t_{MP_ev} + \Delta T_{at_oh}$$

"temperature of propane refrigerant exhaust from MP evaporator (including superheat)"

$$t_{MP_ev} = \text{TEMPERATURE}(\text{fluid1}\$, P = P_{MP_ev}, x = 1)$$

"temperature of propane refrigerant exhaust from MP evaporator (excluding superheat)"

"natural gas side"

$$Q_{\dot{MP_ev}} = m_{\dot{ng_ev}} * (h_{ng_su_MP_ev} - h_{ng_ex_MP_ev})$$

"heat duty of MP mr_evaporator"

$$P_{ng_ex_MP_ev} = P_{ng_su_MP_ev}$$

"enthalpy of propane refrigerant exhaust from MP evaporator" "(no head loss)"

$$h_{ng_su_MP_ev} = \text{ENTHALPY}(\text{fluid2}\$, t = t_{ng_su_MP_ev}, P = P_{ng_su_MP_ev})$$

"enthalpy of natural gas supply to MP evaporator"

"heat exchange effectiveness"

$$Q_{\text{dot_MP_ev}} = m_{\text{dot_ng_ev}} * c_{\text{p_ng_MP_ev}} * \epsilon_{\text{MP_ev}} * (t_{\text{ng_su_MP_ev}} - t_{\text{MP_ev}})$$

"heat duty of MP evaporator"

$$c_{\text{p_ng_MP_ev}} = \text{CP}(\text{fluid2}\$, t=t_{\text{ng_su_MP_ev}}, P=P_{\text{ng_su_MP_ev}})$$

"specific heat of natural gas supply to MP evaporator"

$$\epsilon_{\text{MP_ev}} = 1 - \exp(-\text{NTU}_{\text{MP_ev}})$$

"effectiveness of MP evaporator (semi-isothermal heat exchange)"

$$\text{NTU}_{\text{MP_ev}} = \text{AU}_{\text{MP_ev}} / (m_{\text{dot_ng_ev}} * c_{\text{p_ng_MP_ev}})$$

"number of transfer units of MP evaporator"

$$\text{AU}_{\text{MP_ev}} = \text{AU}_{\text{MP_ev_n}}$$

"heat transfer coefficient-area product of MP propane evaporator (modification needed to take effect of flow rate variations)"

$$t_{\text{ng_ex_MP_ev}} = \text{TEMPERATURE}(\text{fluid2}\$, h=h_{\text{ng_ex_MP_ev}}, P=P_{\text{ng_ex_MP_ev}})$$

"temperature of natural gas exhaust from MP evaporator"

"!MP mr_evaporator"

"supply"

$$h_{\text{r_su_MP_mr_ev}} = h_{\text{r_ex_MP_expv}}$$

"enthalpy of propane refrigerant supply to MP mr_evaporator"

"refrigerant side"

$$Q_{\text{dot_MP_mr_ev}} = m_{\text{dot_r_MP_mr_ev}} * (h_{\text{r_ex_MP_mr_ev}} - h_{\text{r_su_MP_mr_ev}})$$

"heat duty of MP mr_evaporator"

$$h_{\text{r_ex_MP_mr_ev}} = \text{ENTHALPY}(\text{fluid1}\$, P=P_{\text{MP_mr_ev}}, t=t_{\text{r_ex_MP_mr_ev}})$$

"enthalpy of propane refrigerant exhaust from MP mr_evaporator"

$$P_{\text{MP_mr_ev}} = P_{\text{MP_ev}}$$

"pressure of natural gas supply to MP evaporator"

$$t_{\text{r_ex_MP_mr_ev}} = t_{\text{MP_mr_ev}} + \text{DELTA}t_{\text{oh}}$$

"temperature of propane refrigerant exhaust from MP mr_evaporator (including superheat)"

$$t_{\text{MP_mr_ev}} = \text{TEMPERATURE}(\text{fluid1}\$, P=P_{\text{MP_mr_ev}}, x=1)$$

"temperature of propane refrigerant exhaust from MP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

$$h_{\text{N2_su_MP_ev}} = \text{enthalpy}(\text{nitrogen}, T=t_{\text{mr_su_MP_ev}}, p=p_{\text{mr_ev}})$$

"enthalpy of nitrogen component of mixed refrigerant supply to MP mr_evaporator"

$$h_{\text{CH4_su_MP_ev}} = \text{enthalpy}(\text{methane}, T=t_{\text{mr_su_MP_ev}}, p=p_{\text{mr_ev}})$$

"enthalpy of methane component of mixed refrigerant supply to MP mr_evaporator"

$$h_{\text{C2H6_su_MP_ev}} = \text{enthalpy}(\text{ethane}, T=t_{\text{mr_su_MP_ev}}, p=p_{\text{mr_ev}})$$

"enthalpy of ethane component of mixed refrigerant supply to MP mr_evaporator"

$h_{C3H8_su_MP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_su_MP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant supply to MP mr_evaporator"
 "Calculation of the exhaust enthalpies:"
 $h_{N2_ex_MP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of nitrogen component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{CH4_ex_MP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of methane component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{C2H6_ex_MP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of ethane component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{C3H8_ex_MP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant exhaust from MP mr_evaporator"
 "Calculate evaporator duty"
 $Q_{dot_N2_MP_ev} = M_{dot_N2} * (h_{N2_su_MP_ev} - h_{N2_ex_MP_ev})$
 "heat duty of nitrogen component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_CH4_MP_ev} = M_{dot_CH4} * (h_{CH4_su_MP_ev} - h_{CH4_ex_MP_ev})$
 "heat duty of methane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_C2H6_MP_ev} = M_{dot_C2H6} * (h_{C2H6_su_MP_ev} - h_{C2H6_ex_MP_ev})$
 "heat duty of ethane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_C3H8_MP_ev} = M_{dot_C3H8} * (h_{C3H8_su_MP_ev} - h_{C3H8_ex_MP_ev})$
 "heat duty of butane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_MP_mr_ev} =$
 $SUM(Q_{dot_N2_MP_ev}, Q_{dot_CH4_MP_ev}, Q_{dot_C2H6_MP_ev}, Q_{dot_C3H8_MP_ev})$
 "heat duty of MP mr_evaporator"

 "MP Compressor"
 "Calculate exhaust enthalpy (based on energy balance at inlet of MP compressor)"
 $h_{r_su_MP_cp} = (m_{dot_r_MP_ev} * h_{r_ex_MP_ev} + m_{dot_r_MP_mr_ev} * h_{r_ex_MP_mr_ev} + m_{dot_r_LP_cp} * h_{r_ex_LP_cp}) / m_{dot_r_MP_cp}$
 "mass balance at inlet of MP compressor"
 $m_{dot_r_MP_cp} = m_{dot_r_MP_ev} + m_{dot_r_MP_mr_ev} + m_{dot_r_LP_cp}$
 "mass flow rate of propane refrigerant through MP compressor"
 $s_{r_su_MP_cp} = \text{ENTROPY}(\text{fluid1}, h=h_{r_su_MP_cp}, P=P_{MP_ev})$
 "entropy of propane refrigerant supply to MP compressor"
 $h_{r_ex_MP_cp_s} = \text{ENTHALPY}(\text{fluid1}, P=P_{HP_ev}, s=s_{r_su_MP_cp})$
 "MP compressor exhaust enthalpy at isentropic condition"
 $\epsilon_{s_MP_cp} = (h_{r_ex_MP_cp_s} - h_{r_su_MP_cp}) / (h_{r_ex_MP_cp} - h_{r_su_MP_cp})$
 "isentropic efficiency of MP compressor"
 $W_{dot_MP_cp} = m_{dot_r_MP_cp} * (h_{r_ex_MP_cp} - h_{r_su_MP_cp})$
 "MP compressor work"

"!LP expansion valve"

$h_{r_ex_LP_expv}=h_{r_ex_MP_expv}$

"enthalpy of propane refrigerant exhaust from HHP expansion valve"

"!LP evaporator"

"supply"

$h_{r_su_LP_ev}=h_{r_ex_LP_expv}$

"enthalpy of propane refrigerant supply to LP evaporator"

$t_{ng_su_LP_ev}=t_{ng_ex_MP_ev}$

"temperature of natural gas supply to LP evaporator""(no heat loss)"

$P_{ng_su_LP_ev}=P_{ng_ex_MP_ev}$

"pressure of natural gas supply to LP evaporator""(no head loss)"

"refrigerant side"

$Q_{dot_LP_ev}=m_{dot_r_LP_ev}*(h_{r_ex_LP_ev}-h_{r_su_LP_ev})$

"heat duty of LP evaporator"

$h_{r_ex_LP_ev}=ENTHALPY(fluid1\$,P=P_{LP_ev},t=t_{r_ex_LP_ev})$

"enthalpy of propane refrigerant exhaust from LP evaporator"

$t_{r_ex_LP_ev}=t_{LP_ev}+DELTA_{t_oh}$

"temperature of propane refrigerant exhaust from LP evaporator (including superheat)"

$t_{LP_ev}=TEMPERATURE(fluid1\$,P=P_{LP_ev},x=1)$

"temperature of propane refrigerant exhaust from LP evaporator (excluding superheat)"

"natural gas side"

$Q_{dot_LP_ev}=m_{dot_ng_ev}*(h_{ng_su_LP_ev}-h_{ng_ex_LP_ev})$

"heat duty of LP mr_evaporator"

$P_{ng_ex_LP_ev}=P_{ng_su_LP_ev}$

"enthalpy of propane refrigerant exhaust from LP evaporator" "(no head loss)"

$h_{ng_su_LP_ev}=ENTHALPY(fluid2\$,t=t_{ng_su_LP_ev},P=P_{ng_su_LP_ev})$

"enthalpy of natural gas supply to LP evaporator"

"heat exchange effectiveness"

$Q_{dot_LP_ev}=m_{dot_ng_ev}*c_{p_ng_LP_ev}*epsilon_{LP_ev}*(t_{ng_su_LP_ev}-t_{LP_ev})$

"heat duty of LP evaporator"

$c_{p_ng_LP_ev}=CP(fluid2\$,t=t_{ng_su_LP_ev},P=P_{ng_su_LP_ev})$

"specific heat of natural gas supply to LP evaporator"

$epsilon_{LP_ev}=1-\exp(-NTU_{LP_ev})$

"effectiveness of LP evaporator (semi-isothermal heat exchange)"

$NTU_{LP_ev}=AU_{LP_ev}/(m_{dot_ng_ev}*c_{p_ng_LP_ev})$

"number of transfer units of LP evaporator"

AU_LP_ev=AU_LP_ev_n

"heat transfer coefficient-area product of LP propane evaporator (modification needed to take effect of flow rate variations)"

t_ng_ex_LP_ev=TEMPERATURE(fluid2\$,h=h_ng_ex_LP_ev,P=P_ng_ex_LP_ev)

"temperature of natural gas exhaust from LP evaporator"

"!LP mr_evaporator"

"supply"

h_r_su_LP_mr_ev=h_r_ex_LP_expv

"enthalpy of propane refrigerant supply to LP mr_evaporator"

"refrigerant side"

Q_dot_LP_mr_ev=m_dot_r_LP_mr_ev*(h_r_ex_LP_mr_ev-h_r_su_LP_mr_ev)

"heat duty of LP mr_evaporator"

h_r_ex_LP_mr_ev=ENTHALPY(fluid1\$,P=P_LP_mr_ev,t=t_r_ex_LP_mr_ev)

"enthalpy of propane refrigerant exhaust from LP mr_evaporator"

P_LP_mr_ev=P_LP_ev

"pressure of natural gas supply to LP evaporator"

t_r_ex_LP_mr_ev=t_LP_mr_ev+DELTA_{t_oh}

"temperature of propane refrigerant exhaust from LP mr_evaporator (including superheat)"

t_LP_mr_ev=TEMPERATURE(fluid1\$,P=P_LP_mr_ev,x=1)

"temperature of propane refrigerant exhaust from LP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

h_N2_su_LP_ev = enthalpy(nitrogen,T=t_mr_su_LP_ev,p=p_mr_ev)

"enthalpy of nitrogen component of mixed refrigerant supply to LP mr_evaporator"

h_CH4_su_LP_ev = enthalpy(methane,T=t_mr_su_LP_ev,p=p_mr_ev)

"enthalpy of methane component of mixed refrigerant supply to LP mr_evaporator"

h_C2H6_su_LP_ev = enthalpy(ethane,T=t_mr_su_LP_ev,p=p_mr_ev)

"enthalpy of ethane component of mixed refrigerant supply to LP mr_evaporator"

h_C3H8_su_LP_ev = enthalpy(propane,T=t_mr_su_LP_ev,p=p_mr_ev)

"enthalpy of butane component of mixed refrigerant supply to LP mr_evaporator"

"Calculation of the exhaust enthalpies:"

h_N2_ex_LP_ev = enthalpy(nitrogen,T=t_mr_ex_LP_ev,p=p_mr_ev)

"enthalpy of nitrogen component of mixed refrigerant exhaust from LP mr_evaporator"

h_CH4_ex_LP_ev = enthalpy(methane,T=t_mr_ex_LP_ev,p=p_mr_ev)

"enthalpy of methane component of mixed refrigerant exhaust from LP mr_evaporator"

h_C2H6_ex_LP_ev = enthalpy(ethane,T=t_mr_ex_LP_ev,p=p_mr_ev)

"enthalpy of ethane component of mixed refrigerant exhaust from LP mr_evaporator"

$h_{C3H8_ex_LP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_LP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant exhaust from LP mr_evaporator"
 "Calculate evaporator duty"
 $Q_{dot_N2_LP_ev} = M_{dot_N2} * (h_{N2_su_LP_ev} - h_{N2_ex_LP_ev})$
 "heat duty of nitrogen component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_CH4_LP_ev} = M_{dot_CH4} * (h_{CH4_su_LP_ev} - h_{CH4_ex_LP_ev})$
 "heat duty of methane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_C2H6_LP_ev} = M_{dot_C2H6} * (h_{C2H6_su_LP_ev} - h_{C2H6_ex_LP_ev})$
 "heat duty of ethane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_C3H8_LP_ev} = M_{dot_C3H8} * (h_{C3H8_su_LP_ev} - h_{C3H8_ex_LP_ev})$
 "heat duty of butane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_LP_mr_ev} =$
 $SUM(Q_{dot_N2_LP_ev}, Q_{dot_CH4_LP_ev}, Q_{dot_C2H6_LP_ev}, Q_{dot_C3H8_LP_ev})$
 "heat duty of LP mr_evaporator"

"!!LP Compressor"

"Calculate exhaust enthalpy (based on energy balance at inlet of LP compressor)"
 $h_{r_su_LP_cp} = (m_{dot_r_LP_ev} * h_{r_ex_LP_ev} + m_{dot_r_LP_mr_ev} * h_{r_ex_LP_mr_ev}) / m_{dot_r_LP_cp}$
 "mass balance at inlet of LP compressor"
 $m_{dot_r_LP_cp} = m_{dot_r_LP_ev} + m_{dot_r_LP_mr_ev}$
 "mass flow rate of propane refrigerant through LP compressor"
 $s_{r_su_LP_cp} = \text{ENTROPY}(\text{fluid1}\$, h=h_{r_su_LP_cp}, P=P_{LP_ev})$
 "entropy of propane refrigerant supply to LP compressor"
 $h_{r_ex_LP_cp_s} = \text{ENTHALPY}(\text{fluid1}\$, P=P_{MP_ev}, s=s_{r_su_LP_cp})$
 "LP compressor exhaust enthalpy at isentropic condition"
 $\epsilon_{s_LP_cp} = (h_{r_ex_LP_cp_s} - h_{r_su_LP_cp}) / (h_{r_ex_LP_cp} - h_{r_su_LP_cp})$
 "isentropic efficiency of LP compressor"
 $W_{dot_LP_cp} = m_{dot_r_LP_cp} * (h_{r_ex_LP_cp} - h_{r_su_LP_cp})$
 "LP compressor work"

"!!1.6. Performances"

"Calculate total evaporator duty (heat load) of propane refrigerant cycle"
 $Q_{dot_tot_ev} = (Q_{dot_HHP_ev} + Q_{dot_HP_ev} + Q_{dot_MP_ev} + Q_{dot_LP_ev}) + (Q_{dot_HH} P_{mr_ev} + Q_{dot_HP_mr_ev} + Q_{dot_MP_mr_ev} + Q_{dot_LP_mr_ev})$
 "Calculate total work of propane refrigerant compressor"
 $W_{dot_tot_cp} = W_{dot_HHP_cp} + W_{dot_HP_cp} + W_{dot_MP_cp} + W_{dot_LP_cp}$
 "Calculate the coefficient of performance of propane refrigerant compressor"
 $COP = Q_{dot_tot_ev} / W_{dot_tot_cp}$ "useful heat/network input"

"Propane compressor power (Name plate / design) value"

W_dot_tot_cp_design = 41600000 [W]

"!Property Plot data"

"Properties to be plotted are placed in arrays using square brackets to enclose the array index. In this way, the property values are displayed in the Array window after calculations are completed and they can then be directly plotted"

p[1]=p_cd

"pressure of condenser "

h[1]=h_r_ex_HHP_cp

"enthalpy of refrigerant exhaust from HHP compressor"

s[1]=ENTROPY(fluid1\$,h=h[1],P=P[1])

"entropy of refrigerant exhaust from HHP compressor"

t[1]=TEMPERATURE(fluid1\$,h=h[1],P=P[1])

"temperature of refrigerant exhaust from HHP compressor"

p[2]=p_cd

"pressure of condenser "

h[2]=h_r_ex_cd

"enthalpy of refrigerant exhaust from condenser"

s[2]=ENTROPY(fluid1\$,h=h[2],P=P[2])

"entropy of refrigerant exhaust from condenser"

t[2]=TEMPERATURE(fluid1\$,h=h[2],P=P[2])

"temperature of refrigerant exhaust from condenser"

p[3]=p_HHP_ev

"pressure of HHP evaporator"

h[3]=h_r_su_HHP_ev

"enthalpy of refrigerant supply to HHP evaporator"

s[3]=ENTROPY(fluid1\$,h=h[3],P=P[3])

"entropy of refrigerant supply to HHP evaporator"

t[3]=TEMPERATURE(fluid1\$,h=h[3],P=P[3])

"temperature of refrigerant supply to HHP evaporator"

p[4]=p_HHP_ev

"pressure of HHP evaporator"

h[4]=h_r_ex_HHP_ev

"enthalpy of refrigerant exhaust from HHP evaporator"

s[4]=ENTROPY(fluid1\$,h=h[4],P=P[4])

"entropy of refrigerant exhaust from HHP evaporator"

$t[4]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[4],P=P[4])$
 "temperature of refrigerant exhaust from HHP evaporator"
 $p[5]=p_{\text{HHP_ev}}$
 "pressure of HHP evaporator"
 $h[5]=h_{\text{r_ex_HP_cp}}$
 "enthalpy of refrigerant exhaust from HP compressor"
 $s[5]=\text{ENTROPY}(\text{fluid1}\$,h=h[5],P=P[5])$
 "entropy of refrigerant exhaust from HP compressor"
 $t[5]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[5],P=P[5])$
 "temperature of refrigerant exhaust from HP compressor"
 $p[6]=p_{\text{HHP_ev}}$
 "pressure of HHP evaporator"
 $h[6]=h_{\text{r_su_HHP_cp}}$
 "enthalpy of refrigerant supply to HHP compressor"
 $s[6]=\text{ENTROPY}(\text{fluid1}\$,h=h[6],P=P[6])$
 "entropy of refrigerant supply to HHP compressor"
 $t[6]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[6],P=P[6])$
 "temperature of refrigerant supply to HHP compressor"
 $p[7]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"
 $h[7]=h_{\text{r_su_HP_ev}}$
 "enthalpy of refrigerant supply to HP evaporator"
 $s[7]=\text{ENTROPY}(\text{fluid1}\$,h=h[7],P=P[7])$
 "entropy of refrigerant supply to HP evaporator"
 $t[7]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[7],P=P[7])$
 "temperature of refrigerant supply to HP evaporator"
 $p[8]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"
 $h[8]=h_{\text{r_ex_HP_ev}}$
 "enthalpy of refrigerant exhaust from HP evaporator"
 $s[8]=\text{ENTROPY}(\text{fluid1}\$,h=h[8],P=P[8])$
 "entropy of refrigerant exhaust from HP evaporator"
 $t[8]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[8],P=P[8])$
 "temperature of refrigerant exhaust from HP evaporator"
 $p[9]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"
 $h[9]=h_{\text{r_ex_MP_cp}}$
 "enthalpy of refrigerant exhaust from MP compressor"

$s[9]=\text{ENTROPY}(\text{fluid1}\$,h=h[9],P=P[9])$
 "entropy of refrigerant exhaust from MP compressor"
 $t[9]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[9],P=P[9])$
 "temperature of refrigerant exhaust from MP compressor"
 $p[10]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"
 $h[10]=h_{\text{r_su_HP_cp}}$
 "enthalpy of refrigerant supply to HP compressor"
 $s[10]=\text{ENTROPY}(\text{fluid1}\$,h=h[10],P=P[10])$
 "entropy of refrigerant supply to HP compressor"
 $t[10]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[10],P=P[10])$
 "temperature of refrigerant supply to HP compressor"
 $p[11]=p_{\text{MP_ev}}$
 "pressure of MP evaporator"
 $h[11]=h_{\text{r_su_MP_ev}}$
 "enthalpy of refrigerant supply to MP evaporator"
 $s[11]=\text{ENTROPY}(\text{fluid1}\$,h=h[11],P=P[11])$
 "entropy of refrigerant supply to MP evaporator"
 $t[11]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[11],P=P[11])$
 "temperature of refrigerant supply to MP evaporator"
 $p[12]=p_{\text{MP_ev}}$
 "pressure of MP evaporator"
 $h[12]=h_{\text{r_ex_MP_ev}}$
 "enthalpy of refrigerant exhaust from MP evaporator"
 $s[12]=\text{ENTROPY}(\text{fluid1}\$,h=h[12],P=P[12])$
 "entropy of refrigerant exhaust from MP evaporator"
 $t[12]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[12],P=P[12])$
 "temperature of refrigerant exhaust from MP evaporator"
 $p[13]=p_{\text{MP_ev}}$
 "pressure of MP evaporator"
 $h[13]=h_{\text{r_ex_LP_cp}}$
 "enthalpy of refrigerant exhaust from LP compressor"
 $s[13]=\text{ENTROPY}(\text{fluid1}\$,h=h[13],P=P[13])$
 "entropy of refrigerant exhaust from LP compressor"
 $t[13]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[13],P=P[13])$
 "temperature of refrigerant exhaust from LP compressor"
 $p[14]=p_{\text{MP_ev}}$
 "pressure of MP evaporator"

$h[14]=h_{r_su_MP_cp}$
 "enthalpy of refrigerant supply to MP compressor"
 $s[14]=ENTROPY(fluid1\$,h=h[14],P=P[14])$
 "entropy of refrigerant supply to MP compressor"
 $t[14]=TEMPERATURE(fluid1\$,h=h[14],P=P[14])$
 "temperature of refrigerant supply to MP compressor"
 $p[15]=p_{LP_ev}$
 "pressure of LP evaporator"
 $h[15]=h_{r_su_LP_ev}$
 "enthalpy of refrigerant supply to LP evaporator"
 $s[15]=ENTROPY(fluid1\$,h=h[15],P=P[15])$
 "entropy of refrigerant supply to LP evaporator"
 $t[15]=TEMPERATURE(fluid1\$,h=h[15],P=P[15])$
 "temperature of refrigerant supply to LP evaporator"
 $p[16]=p_{LP_ev}$
 "pressure of LP evaporator"
 $h[16]=h_{r_ex_LP_ev}$
 "enthalpy of refrigerant exhaust from LP evaporator"
 $s[16]=ENTROPY(fluid1\$,h=h[16],P=P[16])$
 "entropy of refrigerant exhaust from LP evaporator"
 $t[16]=TEMPERATURE(fluid1\$,h=h[16],P=P[16])$
 "temperature of refrigerant exhaust from LP evaporator"
 $p[17]=p_{LP_ev}$
 "pressure of LP evaporator"
 $h[17]=h_{r_su_LP_cp}$
 "enthalpy of refrigerant supply to LP compressor"
 $s[17]=ENTROPY(fluid1\$,h=h[17],P=P[17])$
 "entropy of refrigerant supply to LP compressor"
 $t[17]=TEMPERATURE(fluid1\$,h=h[17],P=P[17])$
 "temperature of refrigerant supply to LP compressor"
 $p[18]=p_{LP_mr_ev}$
 "pressure of LP mr_evaporator"
 $h[18]=h_{r_ex_LP_mr_ev}$
 "enthalpy of refrigerant exhaust from LP mr_evaporator"
 $s[18]=ENTROPY(fluid1\$,h=h[18],P=P[18])$
 "entropy of refrigerant exhaust from LP evaporator"
 $t[18]=TEMPERATURE(fluid1\$,h=h[18],P=P[18])$
 "temperature of refrigerant exhaust from LP evaporator"

```

p[19]=p_MP_mr_ev
"pressure of MP mr_evaporator"
h[19]=h_r_ex_MP_mr_ev
"enthalpy of refrigerant exhaust from MP evaporator"
s[19]=ENTROPY(fluid1$,h=h[19],P=P[19])
"entropy of refrigerant exhaust from MP evaporator"
t[19]=TEMPERATURE(fluid1$,h=h[19],P=P[19])
"temperature of refrigerant exhaust from MP evaporator"
p[20]=p_HP_mr_ev
"pressure of HP mr_evaporator"
h[20]=h_r_ex_HP_mr_ev
"enthalpy of refrigerant exhaust from HP mr_evaporator"
s[20]=ENTROPY(fluid1$,h=h[20],P=P[20])
"entropy of refrigerant exhaust from HP mr_evaporator"
t[20]=TEMPERATURE(fluid1$,h=h[20],P=P[20])
"temperature of refrigerant exhaust from HP mr_evaporator"
p[21]=p_HHP_mr_ev
"pressure of HHP mr_evaporator"
h[21]=h_r_ex_HHP_mr_ev
"enthalpy of refrigerant exhaust from HHP mr_evaporator"
s[21]=ENTROPY(fluid1$,h=h[21],P=P[21])
"entropy of refrigerant exhaust from HHP mr_evaporator"
t[21]=TEMPERATURE(fluid1$,h=h[21],P=P[21])
"temperature of refrigerant exhaust from HHP mr_evaporator"

"Enthalpy check for propane cycle"
duplicate i=1,21
  hh[i] = Enthalpy(fluid1$,T=T[i],s=s[i])
end

```

"!2. MIXED REFRIGERANT CYCLE"

"Assumption"

"1. $M_{\text{dot_tot}}$, $T_{\text{su_ev}}$, $p_{\text{su_ev}}$, $T_{\text{ex_ev}}$, $p_{\text{ex_ev}}$ are design values"

"Note: The main cryogenic heat exchanger is the evaporator of the mixed refrigerant cycle"

"Note: The propane cycle mr_evaporators make up the condenser of the mixed refrigerant cycle"

"!2.1. Inputs"

"Molar mass of mixed refrigerant"

$MM_{mix} = (X_{N2} * MM_{N2}) + (X_{ch4} * MM_{ch4}) + (X_{c2h6} * MM_{c2h6}) + (X_{C3h8} * MM_{C3H8})$ "molecular weight of mixed refrigerant"

"Mixed refrigerant Inputs"

$M_{dot_tot} = 274$ [kg/s]

"total mass flow rate of mixed refrigerant"

$T_{su_ev} = -31.6$ [C]

"temperature of mixed refrigerant supply to evaporator"

$p_{su_ev} = 44.65E5$ [Pa]

"pressure of mixed refrigerant supply to evaporator"

$T_{ex_ev} = -34.5$ [C]

"temperature of mixed refrigerant exhaust from evaporator"

$p_{ex_ev} = 3E5$ [Pa]

"pressure of mixed refrigerant exhaust from evaporator"

"water side inputs"

$M_{dot_w_int1} = (m_{dot_w_cd} / 3.8)$

"mass flow of cooling water supply to mixed refrigerant compressor intercooler 1"

$M_{dot_w_int2} = (m_{dot_w_cd} / 5.9)$

"mass flow of cooling water supply to mixed refrigerant compressor intercooler 2"

$M_{dot_w_aft} = (m_{dot_w_cd} / 5.5)$

"mass flow of cooling water supply to mixed refrigerant compressor aftercooler"

$t_{w_su_int1} = t_{w_ex}$

"temperature of cooling water supply to intercooler 1"

$t_{w_su_int2} = t_{w_ex}$

"temperature of cooling water supply to intercooler 2"

$t_{w_su_aft} = t_{w_ex}$

"temperature of cooling water supply to aftercooler"

$t_{w_ex_int1} = t_{w_su}$

"temperature of cooling water exhaust from intercooler 1"

$t_{w_ex_int2} = t_{w_su}$

"temperature of cooling water exhaust from intercooler 2"

$t_{w_ex_aft} = t_{w_su}$

"temperature of cooling water exhaust from aftercooler "

"Mixed refrigerant compressor power (Name plate / design) value"

$W_{dot_comp_design} = 86600000$ [W]

"!2.2. Outputs"

" Q_{dot_ev} "

"evaporator heat transfer (duty)"

"Q_dot_cd"

"condenser heat transfer (duty)"

"!2.3. Equations"

"!Mixed Refrigerant Evaporator (cryogenic heat exchanger)"

"Calculation of the supply enthalpies:"

$h_{n2_su_ev} = \text{enthalpy}(\text{nitrogen}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of nitrogen component of mixed refrigerant supply to evaporator"

$h_{ch4_su_ev} = \text{enthalpy}(\text{methane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of methane component of mixed refrigerant supply to evaporator"

$h_{c2h6_su_ev} = \text{enthalpy}(\text{ethane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of ethane component of mixed refrigerant supply to evaporator"

$h_{c3h8_su_ev} = \text{enthalpy}(\text{propane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of propane component of mixed refrigerant supply to evaporator"

"Calculation of the exhaust enthalpies:"

$h_{n2_ex_ev} = \text{enthalpy}(\text{nitrogen}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of nitrogen component of mixed refrigerant exhaust from evaporator"

$h_{ch4_ex_ev} = \text{enthalpy}(\text{methane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of methane component of mixed refrigerant exhaust from evaporator"

$h_{c2h6_ex_ev} = \text{enthalpy}(\text{ethane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of ethane component of mixed refrigerant exhaust from evaporator"

$h_{c3h8_ex_ev} = \text{enthalpy}(\text{propane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of propane component of mixed refrigerant exhaust from evaporator"

"Calculate evaporator (cryogenic heat exchanger) duty"

$Q_{dot_n2_ev} = M_{dot_n2} * (h_{n2_ex_ev} - h_{n2_su_ev})$

"heat duty of nitrogen component of mixed refrigerant evaporator"

$Q_{dot_ch4_ev} = M_{dot_ch4} * (h_{ch4_ex_ev} - h_{ch4_su_ev})$

"heat duty of methane component of mixed refrigerant evaporator"

$Q_{dot_c2h6_ev} = M_{dot_c2h6} * (h_{c2h6_ex_ev} - h_{c2h6_su_ev})$

"heat duty of ethane component of mixed refrigerant evaporator"

$Q_{dot_c3h8_ev} = M_{dot_c3h8} * (h_{c3h8_ex_ev} - h_{c3h8_su_ev})$

"heat duty of propane component of mixed refrigerant evaporator"

$Q_{dot_tot_ev_mix} = \text{SUM}(Q_{dot_n2_ev}, Q_{dot_ch4_ev}, Q_{dot_c2h6_ev}, Q_{dot_c3h8_ev})$

"heat duty of evaporator"

"!Mixed Refrigerant Condenser"

"Calculate mixed refrigerant condenser duty"

$Q_{dot_tot_cd_mix} =$

$\text{SUM}(Q_{dot_HHP_mr_ev}, Q_{dot_HP_mr_ev}, Q_{dot_MP_mr_ev}, Q_{dot_LP_mr_ev})$

"! Cooling water heat flow :"

$c_{w,p} = 4190$ [J/kg-K] "specific heat capacity of water"

$Q_{dot,w_int1} = M_{dot,w_int1} * c_{w,p} * (t_{w_ex_int1} - t_{w_su_int1})$

"heat duty of cooling water supply to intercooler 1"

$Q_{dot,w_int2} = M_{dot,w_int2} * c_{w,p} * (t_{w_ex_int2} - t_{w_su_int2})$

"heat duty of cooling water supply to intercooler 2"

$Q_{dot,w_aft} = M_{dot,w_aft} * c_{w,p} * (t_{w_ex_aft} - t_{w_su_aft})$

"heat duty of cooling water supply to aftercooler "

{ $Q_{dot_tot_w} = \text{SUM}(Q_{dot,w_int1}, Q_{dot,w_int2}, Q_{dot,w_aft})$ }

"!Compressor"

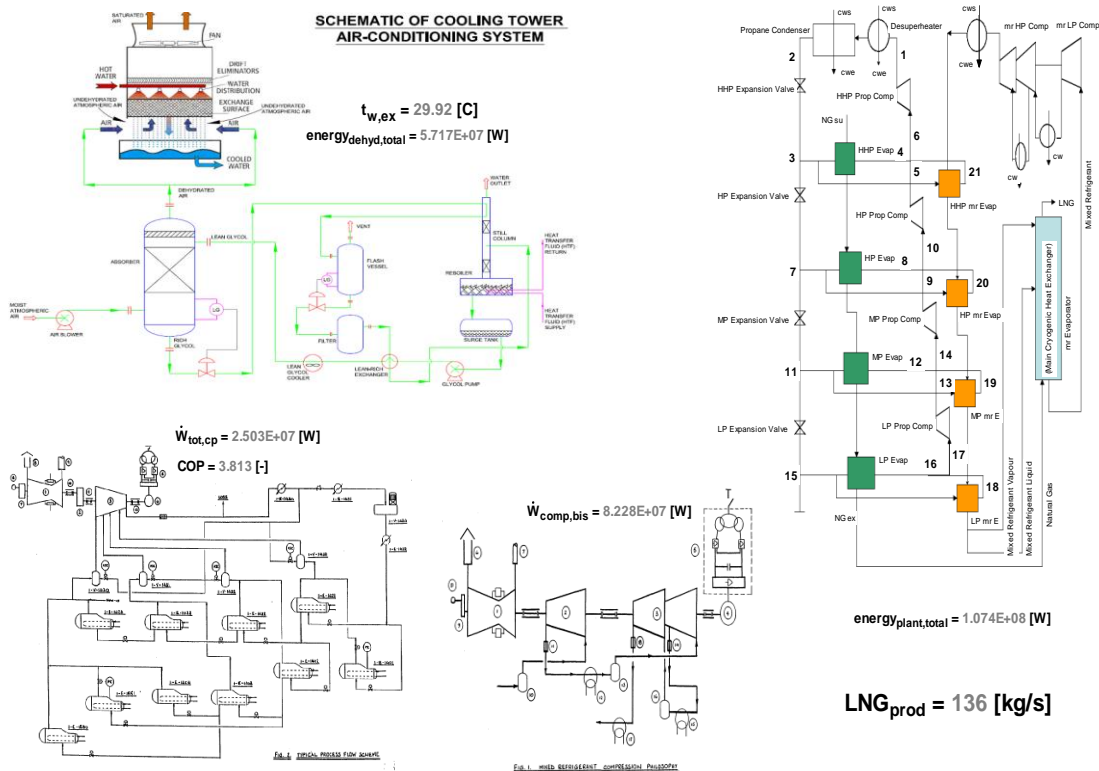
"Calculate compressor work from energy balance on the refrigeration circuit"

$W_{dot_comp} = Q_{dot_tot_cd_mix} - Q_{dot_tot_ev_mix}$

+ $Q_{dot,w_int1} + Q_{dot,w_int2} + Q_{dot,w_aft}$ "check energy balance"

"!TOTAL ENERGY CONSUMPTION BY GAS LIQUEFACTION PLANT"

$energy_plant_total = W_{dot_tot_cp} + W_{dot_com}$



Schematic of EES Model with Cooling Tower Air Conditioning

"!COOLING TOWER PERFORMANCE IMPROVEMENT MODEL (Without Air Conditioning)"

"ESTIMATION OF HEAT TRANSFER COEFFICIENT OF A COOLING TOWER (FOR TOWER SIZING)"

PROCEDURE DCCTSIZING

(M_dot_w,M_dot_a,t_w_su,t_w_ex,t_wb_su,C_0,C_1,C_2,C_3:AU_n,Q_dot_n,NTU_f)

\$COMMON P_atm, rho_a, c_p_a, c_p_g, rho_w, c_p_w, c_p_as, h_fg0

"Assumption"

"1. Initial guess of exhaust wet-bulb temperature"

"Calculate the supply moist air enthalpy"

$h_{a_su}=C_0+C_1*t_{wb_su}+C_2*t_{wb_su}^2+C_3*t_{wb_su}^3$

"Calculate the water side heat capacity flow rate"

$C_{dot_w}=c_{p_w}*M_{dot_w}$

"Calculate the air-water heat transfer"

$Q_{dot_n}=C_{dot_w}*(t_{w_su}-t_{w_ex})$

"1st guess value of the exhaust air wet-bulb temperature"

$t_{wb_ex}=t_{wb_su}+5[C]$

10:

"Calculate the exhaust moist air enthalpy"

$h_{a_ex}=C_0+C_1*t_{wb_ex}+C_2*t_{wb_ex}^2+C_3*t_{wb_ex}^3$

"Calculate the fictitious specific heat and the fictitious fluid heat capacity flow rate"

$c_{p_af}=(h_{a_ex}-h_{a_su})/(t_{wb_ex}-t_{wb_su})$

$C_{dot_af}=c_{p_af}*m_{dot_a}$

"Experimental wet-bulb temperature"

$t_{wb_exp}=t_{wb_ex}$

"Re-calculate the exhaust air wet-bulb temperature"

$t_{wb_ex}=t_{wb_su}+(Q_{dot_n}/C_{dot_af})$

"Convergence test"

$Err_Rel=abs((t_{wb_ex}-t_{wb_exp})/t_{wb_exp})$

IF(Err_Rel>0.001) THEN

GOTO 10

ENDIF

"Calculate the fictitious fluid heat transfer coefficient"

$DT_{lm}=((t_{w_su}-t_{wb_ex})-(t_{w_ex}-t_{wb_su}))/ln((t_{w_su}-t_{wb_ex})/(t_{w_ex}-t_{wb_su}))$

$AU_f=Q_{dot_n}/DT_{lm}$

"Calculate the number of transfer units"

$C_{dot_min}=\min(C_{dot_w}, C_{dot_af})$

$C_{dot_max}=\max(C_{dot_w}, C_{dot_af})$

NTU_f=AU_f/C_dot_min

"Calculate the actual heat transfer coefficient"

AU_n=AU_f*c_p_as/c_p_af

END

"DETERMINATION OF THE LEAVING WATER TEMPERATURE AND THE LEAVING AIR WET-BULB TEMPERATURE (TOWER SIMULATION)"

PROCEDURE DCCTSIM

(M_dot_w_su,M_dot_a,t_w_su,t_w_ref,t_a_su,RH_su,AU_n,M_dot_a_n,M_dot_w_n,C_0,C_1,C_2,C_3:Q_dot,t_w_ex,t_wb_ex)

\$COMMON P_atm, rho_a, c_p_a, c_p_g, rho_w, c_p_w, c_p_as, h_fg0

"Assumptions"

"1. Maximum air flow per cooling tower cell for actual condition is same as design condition"

"2. Initial guess of exhaust wet-bulb temperature"

"3. Hypothesis: saturated air at CT exhaust"

RH_ex=1

"Calculate supply air wetbulb temperature"

t_wb_su=WETBULB(AIRH2O,T=t_a_su,P=P_atm,R=RH_su)

"Calculate the supply moist air enthalpy"

h_a_su=C_0+C_1*t_wb_su+C_2*t_wb_su^2+C_3*t_wb_su^3

"Calculate supply air humidity ratio"

w_su=HumRat(AIRH2O,h=h_a_su,R=RH_su,P=P_atm)

"Calculate the water side heat capacity flow rate"

C_dot_w=c_p_w*M_dot_w_su

"1st guess value of the exhaust air wet-bulb temperature"

t_wb_ex=t_wb_su+5[C]

10:

"Calculate the exhaust moist air enthalpy"

h_a_ex=C_0+C_1*t_wb_ex+C_2*t_wb_ex^2+C_3*t_wb_ex^3

"Calculate the fictitious specific heat and the fictitious fluid heat capacity flow rate"

c_p_af=(h_a_ex-h_a_su)/(t_wb_ex-t_wb_su)

C_dot_af=c_p_af*m_dot_a

"Calculate the ratio of the minimum heat capacity flow rate to the maximum one"

C_dot_min =min (C_dot_w, C_dot_af)

C_dot_max =max (C_dot_w, C_dot_af)

C_dot_ratio=C_dot_min/C_dot_max

"Calculate the actual heat transfer coefficient"

m=0.43

n=1.03

```

AU=AU_n*(M_dot_w_su/M_dot_w_n)^m*(M_dot_a/M_dot_a_n)^n " according to
J.Lebrun -(the given values for the exponents m &
n are best fit values)"
"Calculate the fictitious heat transfer coefficient and the number of transfer unit"
AU_f=AU*c_p_af/c_p_as
NTU=AU_f/C_dot_min
"Calculate the effectiveness of the heat exchanger"
IF(C_dot_ratio<0.995) THEN
Eff=(1-EXP(-NTU*(1-C_dot_ratio)))/(1-C_dot_ratio*EXP(-NTU*(1-C_dot_ratio)))
ELSE
Eff=NTU/(1+NTU)
ENDIF
"Calculate the air-water heat transfer"
Q_dot=Eff*C_dot_min*(t_w_su-t_wb_su)
"Experimental wet-bulb temperature"
t_wb_exp=t_wb_ex
"Re-calculate the exhaust air wet-bulb temperature"
t_wb_ex=t_wb_su+(Q_dot/C_dot_af)
"Convengence test"
Err_Rel=abs((t_wb_ex-t_wb_exp)/t_wb_exp)
IF(Err_Rel>0.001) THEN
GOTO 10
ENDIF
"Calculate the exhaust water temperature"
t_w_ex=t_w_su-Q_dot/C_dot_w
"Calculate exhaust air temperature"
t_a_ex=Temperature(AirH2O,h=h_a_ex,R=1,P=P_atm)
"Calculate exhaust air humidity ratio"
w_ex=HumRat(AirH2O,h=h_a_ex,R=1,P=P_atm)
"Calculate water evaporation rate"
M_dot_w_consumed=M_dot_a*(w_ex-w_su)
"Calculate exhaust water flow rate"
M_dot_w_ex=M_dot_w_su-M_dot_w_consumed
"Calculate the exhaust water temperature"
t_w_ex=((M_dot_w_su*c_p_w*(t_w_su-t_w_ref)-Q_dot)/(M_dot_w_ex*c_p_w))+t_w_ref
END

"DETERMINATION OF TEMPERATURE RISE OF LEAVING WATER IN COOLING WATER
CIRCULATING PUMP"

```

PROCEDURE PUMPDELTA_T

```
(power_pump,pump_efficiency,m_dot_w_pump,c_p_w:t_delta_pump)
$COMMON P_atm, rho_a, c_p_a, c_p_g, rho_w, c_p_w, c_p_as, h_fg0
"Calculate the cooling water temperature rise in circulation pump"
t_delta_pump=((power_pump*(1-pump_efficiency))/(c_p_w*m_dot_w_pump))
END
```

!"SENSITIVITY ANALYSIS - Cooling Water Temperature vs. Liquefied Natural Gas Production"

"DETERMINATION OF SENSITIVITY OF PRODUCTION TO COOLING WATER SUPPLY TEMPERATURE"

PROCEDURE SENSITIVITY (X:LNG_prod)

"Calculate Liquefied Natural Gas production variation with cooling water supply temperature"

```
IF (X <= 32) THEN
LNG_prod=(-0.53*X)+151.84
ELSE
LNG_prod=(-3.05*X)+232.8
ENDIF
END
```

"DETERMINATION OF UNCERTAINTY OF THE CALCULATED COLD WATER TEMPERATURE"

"Assumptions"

"1. The uncertainty of the air flow rate is taken as +/- 50 kg/s while that of the ambient air relative humidity is taken as 0.05"

"2. The uncertainty of the calculated cold water temperature is based on the air flow rate and the relative humidity uncertainties only"

U=UncertaintyOf(t_w_ex)

!"SIZING"

"Design data (based on 8-cell tower)"

```
M_dot_w_n=7777.776 [kg/s]
M_dot_a_n= convert(m^3/h,m^3/s) * 15408000 [m^3/h] * rho_a
t_w_su_n=40.1 [C]
t_w_ex_n=32 [C]
t_wb_su_n=28 [C]
C_0=9362.5 [J/kg]
C_1=1786.1 [J/kg-K]
C_2=11.35 [J/kg-K^2]
```

C_3=0.98855 [J/kg-K³]

"Identification of parameters"

CALL DCCTSIING

(M_dot_w_n,M_dot_a_n,t_w_su_n,t_w_ex_n,t_wb_su_n,C_0,C_1,C_2,C_3:AU_n,Q_dot_n,NTU_f)

"!SIMULATION"

"Actual data (based on 9-cell tower)"

M_dot_w_su=8305.556 [kg/s]

M_dot_a = convert(m³/h,m³/s) * 17334000 [m³/h] * rho_a

t_w_su=39.7 [C]

t_w_ref=0 [C]

t_a_su=31.9 [C]

RH_su=0.632 [-]

"Identification of parameters"

CALL DCCTSIM

(M_dot_w_su,M_dot_a,t_w_su,t_w_ref,t_a_su,RH_su,AU_n,M_dot_a_n,M_dot_w_n,C_0,C_1,C_2,C_3:Q_dot,t_w_ex,t_wb_ex)

"!COOLING WATER CIRCULATION PUMP"

"Calculation data"

power_pump=2200000 [W]

"cooling water circulation pump power"

pump_efficiency=0.87 [-]

"cooling water circulation pump efficiency"

m_dot_w_pump=M_dot_w_su*0.5

"mass flow rate of cooling water through pump"

"Identification of parameters"

CALL PUMPDELAT (power_pump,pump_efficiency,m_dot_w_pump,c_p_w:t_delta_pump)

"!Fluid properties"

P_atm=100600 [Pa]

c_p_a=1005 [J/kg-K]

c_p_g=1820 [J/kg-K]

c_p_w=4190 [J/kg-K]

c_p_as=1025 [J/kg-K]

h_fg0=2.501E6 [J/kg]

rho_a=1.11 [kg/m³]

rho_w=1000 [kg/m³]

"!SENSITIVITY"

"Calculation data"

X=t_w_ex

"temperature of cooling water exhaust from cooling tower"

"Identification of parameters"

CALL SENSITIVITY (X:LNG_prod)

"!ENERGY ANALYSIS FOR THE GAS LIQUEFACTION PLANT"

"!1. PROPANE REFRIGERANT CYCLE"

"!1.1. Inputs"

"!1.1.1 Natural Gas Inputs"

"Supply conditions"

t_ng_su_ev=39 [C]

"temperature of natural gas to propane evaporators"

P_ng_su_ev=53.5E5 [Pa]

"pressure of natural gas to propane evaporators"

M_dot_ng_ev=142 [kg/s]

"mass flow rate of natural gas through propane evaporators"

"!1.1.2 Cooling Water Inputs"

"Cooling water temperature"

m_dot_w_desup=600 [kg/s]

"mass flow rate of cooling water through propane desuperheater"

t_w_su_desup=t_w_ex

"temperature of cooling water supply to propane desuperheater"

t_w_ex_desup=t_w_su

"temperature of cooling water exhaust from propane desuperheater"

t_w_su_cd=t_w_ex

"temperature of cooling water supply to propane condenser"

t_w_ex_cd=t_w_su

"temperature of cooling water supply from propane condenser"

{m_dot_w_cd=3600 [kg/s]}

"!1.1.3 Mixed Refrigerant Inputs"

"Mixed Refrigerant (mr) common Inputs"

M_dot_mr_ev = 274 [kg/s]

"mass flow rate of mixed refrigerant through propane mr_evaporators"

p_mr_ev = 48E5 [Pa]

"pressure of mixed refrigerant to propane mr_evaporators"

"Molar fractions of the components of mixed refrigerant"

$$X_{N2} = 0.084$$

"mol fraction of nitrogen in mixed refrigerant"

$$X_{CH4} = 0.383$$

"mol fraction of propane in mixed refrigerant"

$$X_{C2H6} = 0.411$$

"mol fraction of ethane in mixed refrigerant"

$$X_{C3H8} = 0.122$$

"mol fraction of butane in mixed refrigerant"

"Molar masses of the components of mixed refrigerant"

$$MM_{N2} = \text{molarmass}(\text{nitrogen})$$

"molecular weight of nitrogen"

$$MM_{CH4} = \text{molarmass}(\text{methane})$$

"molecular weight of propane"

$$MM_{C2H6} = \text{molarmass}(\text{ethane})$$

"molecular weight of ethane"

$$MM_{C3H8} = \text{molarmass}(\text{propane})$$

"molecular weight of butane"

"Mass flow rates of the components of mixed refrigerant"

$$M_{\text{dot}}_{N2} = M_{\text{dot}}_{\text{mr_ev}} * X_{N2}$$

"mass flow rate of nitrogen in mixed refrigerant"

$$M_{\text{dot}}_{CH4} = M_{\text{dot}}_{\text{mr_ev}} * X_{CH4}$$

"mass flow rate of propane in mixed refrigerant"

$$M_{\text{dot}}_{C2H6} = M_{\text{dot}}_{\text{mr_ev}} * X_{C2H6}$$

"mass flow rate of ethane in mixed refrigerant"

$$M_{\text{dot}}_{C3H8} = M_{\text{dot}}_{\text{mr_ev}} * X_{C3H8}$$

"mass flow rate of butane in mixed refrigerant"

$$M_{\text{dot}}_{\text{mr_ev_bis}} = \text{SUM}(M_{\text{dot}}_{N2}, M_{\text{dot}}_{CH4}, M_{\text{dot}}_{C2H6}, M_{\text{dot}}_{C3H8})$$

"total mass flow rate of mixed refrigerant"

"HHP mr_evaporator inputs"

$$c_{p_mr_HHP_ev} = 2992 \text{ [J/kg-K]}$$

"specific heat capacity of mixed refrigerant in HHP propane mr_evaporator"

$$t_{\text{mr_su_HHP_ev}} = 35 \text{ [C]}$$

"temperature of mixed refrigerant supply to HHP propane mr_evaporator"

$$t_{\text{mr_ex_HHP_ev}} = 11.5 \text{ [C]}$$

"temperature of mixed refrigerant exhaust from HHP propane mr_evaporator"

"HP mr_evaporator inputs"

$$c_{p_mr_HP_ev} = 3077 \text{ [J/kg-K]}$$

"specific heat capacity of mixed refrigerant in HP propane mr_evaporator"

$t_{mr_su_HP_ev}=11.5$ [C]
 "temperature of mixed refrigerant supply to HP propane mr_evaporator"
 $t_{mr_ex_HP_ev}=-4.5$ [C]
 "temperature of mixed refrigerant exhaust from HP propane mr_evaporator"
 "MP mr_evaporator inputs"
 $c_{p_mr_MP_ev}=3010$ [J/kg-K]
 "specific heat capacity of mixed refrigerant in MP propane mr_evaporator"
 $t_{mr_su_MP_ev}=-4.5$ [C]
 "temperature of mixed refrigerant supply to MP propane mr_evaporator"
 $t_{mr_ex_MP_ev}=-18.8$ [C]
 "temperature of mixed refrigerant exhaust from MP propane mr_evaporator"
 "LP mr_evaporator inputs"
 $c_{p_mr_LP_ev}=2963$ [J/kg-K]
 "specific heat capacity of mixed refrigerant in LP propane mr_evaporator"
 $t_{mr_su_LP_ev}=-18.8$ [C]
 "temperature of mixed refrigerant supply to LP propane mr_evaporator"
 $t_{mr_ex_LP_ev}=-31.6$ [C]
 "temperature of mixed refrigerant exhaust from LP propane mr_evaporator"

"!1.2. Outputs"

"propane refrigerant compressor mechanical power"
 "propane refrigeration cycle coefficient of performance (COP)"

"!1.3. Parameters"

$fluid1\$='propane'$
 $fluid2\$='methane'$
 "Compressor"
 $epsilon_s_HHP_cp=0.76$
 "isentropic efficiency of High High Pressure stage of propane refrigerant compressor"
 $epsilon_s_HP_cp=0.74$
 "isentropic efficiency of High Pressure stage of propane refrigerant compressor"
 $epsilon_s_MP_cp=0.77$
 "isentropic efficiency of Medium Pressure stage of propane refrigerant compressor"
 $epsilon_s_LP_cp=0.82$
 "isentropic efficiency of Low Pressure stage of propane refrigerant compressor"
 "Heat exchangers (evaporators and condenser)"
 $AU_HHP_ev_n=918080$ [W/K]
 "heat transfer coefficient-area product of High High Pressure propane evaporator"

AU_HP_ev_n=918136 [W/K]
 "heat transfer coefficient-area product of High Pressure evaporator"
 AU_MP_ev_n=816888 [W/K]
 "heat transfer coefficient-area product of Medium Pressure evaporator"
 AU_LP_ev_n=762420 [W/K]
 "heat transfer coefficient-area product of Low Pressure evaporator"
 AU_mr_HHP_ev=2088394 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_HP_ev=4388696 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_MP_ev=3390400 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_mr_LP_ev=2736708 [W/K]
 "heat transfer coefficient-area product of High High Pressure mr_evaporator"
 AU_desup=1022200 [W/K]
 "heat transfer coefficient-area product of propane desuperheater"
 AU_cd=35613420 [W/K]
 "heat transfer coefficient-area product of condenser"
 DELTAt_sc=8.5
 "subcooling (at outlet of condenser)"
 DELTAt_oh=5
 "superheating (at outlet of evaporators)"
 "Control (boundary conditions)"
 t_ng_ex_HHP_ev=21 [C]
 "temperature of natural gas exhaust from High High Pressure propane evaporator"
 t_ng_ex_HP_ev=-4.5 [C]
 "temperature of natural gas exhaust from High Pressure propane evaporator"
 t_ng_ex_MP_ev=-18.8 [C]
 "temperature of natural gas exhaust from Medium Pressure propane evaporator"
 t_ng_ex_LP_ev=-29.7 [C]
 "temperature of natural gas exhaust from Low Pressure propane evaporator"

"Note: x is the propane quality (vapour mass fraction) in the condenser. x=1 denotes saturated vapour state and x=0 denotes saturated liquid state"

"!1.4. Equations"

"!Desuperheater"

"supply"

$T_{r_su_desup}=t[1]$
 "temperature of propane refrigerant supply to desuperheater"
 $h_{r_su_desup}=\text{ENTHALPY}(\text{fluid1}\$,T=T_{r_su_desup},P=P_{cd})$
 "enthalpy of propane refrigerant supply to desuperheater"
 "refrigerant side"
 $T_{r_ex_desup}=\text{TEMPERATURE}(\text{fluid1}\$,P=P_{cd},x=1)$
 "temperature of propane refrigerant exhaust from desuperheater"
 $h_{r_ex_desup}=\text{ENTHALPY}(\text{fluid1}\$,T=T_{r_ex_desup},x=1)$
 "enthalpy of propane refrigerant exhaust from desuperheater"
 $Q_{dot_desup}=m_{dot_r_HHP_cp}*(h_{r_su_desup}-h_{r_ex_desup})$
 "heat duty of desuperheater"

"!Condenser"
 "supply"
 $h_{r_su_cd}=h_{r_ex_desup}$
 "enthalpy of propane refrigerant supply to condenser"
 $T_{r_su_cd}=T_{r_ex_desup}$
 "temperature of propane refrigerant supply to condenser"
 "refrigerant side"
 $Q_{dot_cd}=m_{dot_r_HHP_cp}*(h_{r_su_cd}-h_{r_ex_cd})$
 "heat duty of condenser"
 $T_{r_ex_cd}=T_{cd}-\text{DELTA}t_{sc}$
 "temperature of propane refrigerant exhaust from condenser (including subcooling)"
 $T_{cd}=\text{TEMPERATURE}(\text{fluid1}\$,P=P_{cd},x=0)$
 "temperature of propane refrigerant exhaust from condenser (excluding subcooling)"
 $h_{r_ex_cd}=\text{ENTHALPY}(\text{fluid1}\$,P=P_{cd},T=T_{r_ex_cd})$
 "enthalpy of propane refrigerant exhaust from condenser" {need to fix subcooling}
 "water side"
 $Q_{dot_cd}=m_{dot_w_cd}*c_{p_w}*(t_{w_ex_cd}-t_{w_su_cd})$
 "heat duty of condenser"
 "heat exchange effectiveness"
 $Q_{dot_cd}=m_{dot_w_cd}*c_{p_w}*\epsilon_{cd}*(t_{cd}-t_{w_su_cd})$
 "heat duty of condenser"
 $\epsilon_{cd}=1-\exp(-\text{NTU}_{cd})$
 "effectiveness of condenser"
 $\text{NTU}_{cd}=\text{AU}_{cd}/(m_{dot_w_cd}*c_{p_w})$
 "number of transfer units of condenser"

"!HHP Expansion valve"

$h_{r_ex_HHP_expv}=h_{r_ex_cd}$

"enthalpy of propane refrigerant exhaust from HHP expansion valve"

"!HHP evaporator"

"supply"

$h_{r_su_HHP_ev}=h_{r_ex_HHP_expv}$

"enthalpy of propane refrigerant supply to HHP evaporator"

$t_{ng_su_HHP_ev}=t_{ng_su_ev}$

"temperature of natural gas supply to HHP evaporator"

$P_{ng_su_HHP_ev}=P_{ng_su_ev}$

"pressure of natural gas supply to HHP evaporator"

"refrigerant side"

$Q_{dot_HHP_ev}=m_{dot_r_HHP_ev}*(h_{r_ex_HHP_ev}-h_{r_su_HHP_ev})$

"heat duty of HHP evaporator"

$h_{r_ex_HHP_ev}=ENTHALPY(fluid1\$,P=P_{HHP_ev},t=t_{r_ex_HHP_ev})$

"enthalpy of propane refrigerant exhaust from HHP evaporator"

$t_{r_ex_HHP_ev}=t_{HHP_ev}+DELTA t_{oh}$

"temperature of propane refrigerant exhaust from HHP evaporator (including superheat)"

$t_{HHP_ev}=TEMPERATURE(fluid1\$,P=P_{HHP_ev},x=1)$

"temperature of propane refrigerant exhaust from HHP evaporator (excluding superheat)"

"natural gas side"

$Q_{dot_HHP_ev}=m_{dot_ng_ev}*(h_{ng_su_HHP_ev}-h_{ng_ex_HHP_ev})$

"heat duty of HHP evaporator"

$P_{ng_ex_HHP_ev}=P_{ng_su_HHP_ev}$

"pressure of natural gas exhaust from HHP evaporator (no head loss)"

$h_{ng_su_HHP_ev}=ENTHALPY(fluid2\$,t=t_{ng_su_HHP_ev},P=P_{ng_su_HHP_ev})$

"enthalpy of natural gas supply to HHP evaporator"

"heat exchange effectiveness"

$Q_{dot_HHP_ev}=m_{dot_ng_ev}*c_{p_ng_HHP_ev}*\epsilon_{HHP_ev}*(t_{ng_su_HHP_ev}-t_{HHP_ev})$ "heat duty of HHP evaporator"

$c_{p_ng_HHP_ev}=CP(fluid2\$,t=t_{ng_su_HHP_ev},P=P_{ng_su_HHP_ev})$

"specific heat of natural gas supply to HHP evaporator"

$\epsilon_{HHP_ev}=1-\exp(-NTU_{HHP_ev})$

"effectiveness of HHP evaporator (semi-isothermal heat exchange)"

$NTU_{HHP_ev}=AU_{HHP_ev}/(m_{dot_ng_ev}*c_{p_ng_HHP_ev})$

"number of transfer units of HHP evaporator"

AU_HHP_ev=AU_HHP_ev_n

"heat transfer coefficient-area product of HHP propane evaporator (modification needed to take effect of flow rate variations)"

t_ng_ex_HHP_ev=TEMPERATURE(fluid2\$,h=h_ng_ex_HHP_ev,P=P_ng_ex_HHP_ev)

"temperature of natural gas exhaust from HHP evaporator"

"!HHP mr_evaporator"

"supply"

h_r_su_HHP_mr_ev=h_r_ex_HHP_expv

"enthalpy of propane refrigerant supply to HHP mr_evaporator"

"refrigerant side"

Q_dot_HHP_mr_ev=m_dot_r_HHP_mr_ev*(h_r_ex_HHP_mr_ev-h_r_su_HHP_mr_ev)

"heat duty of HHP mr_evaporator"

h_r_ex_HHP_mr_ev=ENTHALPY(fluid1\$,P=P_HHP_mr_ev,t=t_r_ex_HHP_mr_ev)

"enthalpy of propane refrigerant exhaust from HHP mr_evaporator"

P_HHP_mr_ev=P_HHP_ev-1.8E5 [Pa]

"pressure of natural gas supply to HHP evaporator"

t_r_ex_HHP_mr_ev=t_HHP_mr_ev+DELTA_t_oh

"temperature of propane refrigerant exhaust from HHP mr_evaporator (including superheat)"

t_HHP_mr_ev=TEMPERATURE(fluid1\$,P=P_HHP_mr_ev,x=1)

"temperature of propane refrigerant exhaust from HHP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

h_N2_su_HHP_ev = enthalpy(nitrogen,T=t_mr_su_HHP_ev,p=p_mr_ev)

"enthalpy of nitrogen component of mixed refrigerant supply to HHP mr_evaporator"

h_CH4_su_HHP_ev = enthalpy(methane,T=t_mr_su_HHP_ev,p=p_mr_ev)

"enthalpy of methane component of mixed refrigerant supply to HHP mr_evaporator"

h_C2H6_su_HHP_ev = enthalpy(ethane,T=t_mr_su_HHP_ev,p=p_mr_ev)

"enthalpy of ethane component of mixed refrigerant supply to HHP mr_evaporator"

h_C3H8_su_HHP_ev = enthalpy(propane,T=t_mr_su_HHP_ev,p=p_mr_ev)

"enthalpy of butane component of mixed refrigerant supply to HHP mr_evaporator"

"Calculation of the exhaust enthalpies:"

h_N2_ex_HHP_ev = enthalpy(nitrogen,T=t_mr_ex_HHP_ev,p=p_mr_ev)

"enthalpy of nitrogen component of mixed refrigerant exhaust from HHP mr_evaporator"

h_CH4_ex_HHP_ev = enthalpy(methane,T=t_mr_ex_HHP_ev,p=p_mr_ev)

"enthalpy of methane component of mixed refrigerant exhaust from HHP mr_evaporator"

h_C2H6_ex_HHP_ev = enthalpy(ethane,T=t_mr_ex_HHP_ev,p=p_mr_ev)

"enthalpy of ethane component of mixed refrigerant exhaust from HHP mr_evaporator"

$h_{C3H8_ex_HHP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_HHP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant exhaust from HHP mr_evaporator"
 "Calculate evaporator duty"
 $Q_{dot_N2_HHP_ev} = M_{dot_N2} * (h_{N2_su_HHP_ev} - h_{N2_ex_HHP_ev})$
 "heat duty of nitrogen component of mixed refrigerant HHP mr_evaporator"
 $Q_{dot_CH4_HHP_ev} = M_{dot_CH4} * (h_{CH4_su_HHP_ev} - h_{CH4_ex_HHP_ev})$
 "heat duty of methane component of mixed refrigerant HHP mr_evaporator"
 $Q_{dot_C2H6_HHP_ev} = M_{dot_C2H6} * (h_{C2H6_su_HHP_ev} - h_{C2H6_ex_HHP_ev})$
 "heat duty of ethane component of mixed refrigerant HHP mr_evaporator"
 $Q_{dot_C3H8_HHP_ev} = M_{dot_C3H8} * (h_{C3H8_su_HHP_ev} - h_{C3H8_ex_HHP_ev})$
 "heat duty of butane component of mixed refrigerant HHP mr_evaporator"
 $Q_{dot_HHP_mr_ev} =$
 $SUM(Q_{dot_N2_HHP_ev}, Q_{dot_CH4_HHP_ev}, Q_{dot_C2H6_HHP_ev}, Q_{dot_C3H8_HHP_ev})$
 "heat duty of HHP mr_evaporator"

"!HHP Compressor"

"Calculate exhaust enthalpy (based on energy balance at inlet of HHP compressor)"
 $h_{r_su_HHP_cp} = (m_{dot_r_HHP_ev} * h_{r_ex_HHP_ev} + m_{dot_r_HHP_mr_ev} * h_{r_ex_HHP_mr_ev} + m_{dot_r_HP_cp} * h_{r_ex_HP_cp}) / m_{dot_r_HHP_cp}$
 "mass balance at inlet of HHP compressor"
 $m_{dot_r_HHP_cp} = m_{dot_r_HHP_ev} + m_{dot_r_HHP_mr_ev} + m_{dot_r_HP_cp}$
 "mass flow rate of propane refrigerant through HHP compressor"
 $s_{r_su_HHP_cp} = \text{ENTROPY}(\text{fluid1}, h=h_{r_su_HHP_cp}, P=P_{HHP_ev})$
 "entropy of propane refrigerant supply to HHP compressor"
 $h_{r_ex_HHP_cp_s} = \text{ENTHALPY}(\text{fluid1}, P=P_{cd}, s=s_{r_su_HHP_cp})$
 "HHP compressor exhaust enthalpy at isentropic condition"
 $\epsilon_{s_HHP_cp} = (h_{r_ex_HHP_cp_s} - h_{r_su_HHP_cp}) / (h_{r_ex_HHP_cp} - h_{r_su_HHP_cp})$ "isentropic efficiency of HHP compressor"
 $W_{dot_HHP_cp} = m_{dot_r_HHP_cp} * (h_{r_ex_HHP_cp} - h_{r_su_HHP_cp})$
 "HHP compressor work"

"!HP expansion valve"

$h_{r_ex_HP_expv} = h_{r_ex_HHP_expv}$
 "enthalpy of propane refrigerant exhaust from HP expansion valve"

"!HP evaporator"

"supply"
 $h_{r_su_HP_ev} = h_{r_ex_HP_expv}$
 "enthalpy of propane refrigerant supply to HP evaporator"

$t_{ng_su_HP_ev}=t_{ng_ex_HHP_ev}$
 "temperature of natural gas supply to HP evaporator"(no heat loss)"
 $P_{ng_su_HP_ev}=P_{ng_ex_HHP_ev}$
 "pressure of natural gas supply to HP evaporator"(no head loss)"
 "refrigerant side"
 $Q_{dot_HP_ev}=m_{dot_r_HP_ev}(h_{r_ex_HP_ev}-h_{r_su_HP_ev})$
 "heat duty of HP evaporator"
 $h_{r_ex_HP_ev}=ENTHALPY(fluid1\$,P=P_{HP_ev},t=t_{r_ex_HP_ev})$
 "enthalpy of propane refrigerant exhaust from HP evaporator"
 $t_{r_ex_HP_ev}=t_{HP_ev}+DELTA t_{oh}$
 "temperature of propane refrigerant exhaust from HP evaporator (including superheat)"
 $t_{HP_ev}=TEMPERATURE(fluid1\$,P=P_{HP_ev},x=1)$
 "temperature of propane refrigerant exhaust from HP evaporator (excluding superheat)"
 "natural gas side"
 $Q_{dot_HP_ev}=m_{dot_ng_ev}(h_{ng_su_HP_ev}-h_{ng_ex_HP_ev})$
 "heat duty of HP mr_evaporator"
 $P_{ng_ex_HP_ev}=P_{ng_su_HP_ev}$
 "pressure of natural gas exhaust from HP evaporator (no head loss)"
 $h_{ng_su_HP_ev}=ENTHALPY(fluid2\$,t=t_{ng_su_HP_ev},P=P_{ng_su_HP_ev})$
 "enthalpy of natural gas supply to HP evaporator"
 "heat exchange effectiveness"
 $Q_{dot_HP_ev}=m_{dot_ng_ev}c_{p_ng_HP_ev}\epsilon_{HP_ev}(t_{ng_su_HP_ev}-t_{HP_ev})$
 "heat duty of HP evaporator"
 $c_{p_ng_HP_ev}=CP(fluid2\$,t=t_{ng_su_HP_ev},P=P_{ng_su_HP_ev})$
 "specific heat of natural gas supply to HP evaporator"
 $\epsilon_{HP_ev}=1-\exp(-NTU_{HP_ev})$
 "effectiveness of HP evaporator (semi-isothermal heat exchange)"
 $NTU_{HP_ev}=AU_{HP_ev}/(m_{dot_ng_ev}c_{p_ng_HP_ev})$
 "number of transfer units of HP evaporator"
 $AU_{HP_ev}=AU_{HP_ev_n}$
 "heat transfer coefficient-area product of HP propane evaporator (modification needed to take effect of flow rate variations)"
 $t_{ng_ex_HP_ev}=TEMPERATURE(fluid2\$,h=h_{ng_ex_HP_ev},P=P_{ng_ex_HP_ev})$
 "temperature of natural gas exhaust from HP evaporator"

 "!HP mr_evaporator"
 "supply"
 $h_{r_su_HP_mr_ev}=h_{r_ex_HP_expv}$
 "enthalpy of propane refrigerant supply to HP mr_evaporator"

"refrigerant side"

$$Q_dot_HP_mr_ev = m_dot_r_HP_mr_ev * (h_r_ex_HP_mr_ev - h_r_su_HP_mr_ev)$$

"heat duty of HP mr_evaporator"

$$h_r_ex_HP_mr_ev = ENTHALPY(fluid1$, P=P_HP_mr_ev, t=t_r_ex_HP_mr_ev)$$

"enthalpy of propane refrigerant exhaust from HP mr_evaporator"

$$P_HP_mr_ev = P_HP_ev$$

"pressure of natural gas supply to HP evaporator"

$$t_r_ex_HP_mr_ev = t_HP_mr_ev + \Delta T_{oh}$$

"temperature of propane refrigerant exhaust from HP mr_evaporator (including superheat)"

$$t_HP_mr_ev = TEMPERATURE(fluid1$, P=P_HP_mr_ev, x=1)$$

"temperature of propane refrigerant exhaust from HP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

$$h_N2_su_HP_ev = \text{enthalpy}(\text{nitrogen}, T=t_mr_su_HP_ev, p=p_mr_ev)$$

"enthalpy of nitrogen component of mixed refrigerant supply to HP mr_evaporator"

$$h_CH4_su_HP_ev = \text{enthalpy}(\text{methane}, T=t_mr_su_HP_ev, p=p_mr_ev)$$

"enthalpy of methane component of mixed refrigerant supply to HP mr_evaporator"

$$h_C2H6_su_HP_ev = \text{enthalpy}(\text{ethane}, T=t_mr_su_HP_ev, p=p_mr_ev)$$

"enthalpy of ethane component of mixed refrigerant supply to HP mr_evaporator"

$$h_C3H8_su_HP_ev = \text{enthalpy}(\text{propane}, T=t_mr_su_HP_ev, p=p_mr_ev)$$

"enthalpy of butane component of mixed refrigerant supply to HP mr_evaporator"

"Calculation of the exhaust enthalpies:"

$$h_N2_ex_HP_ev = \text{enthalpy}(\text{nitrogen}, T=t_mr_ex_HP_ev, p=p_mr_ev)$$

"enthalpy of nitrogen component of mixed refrigerant exhaust from HP mr_evaporator"

$$h_CH4_ex_HP_ev = \text{enthalpy}(\text{methane}, T=t_mr_ex_HP_ev, p=p_mr_ev)$$

"enthalpy of methane component of mixed refrigerant exhaust from HP mr_evaporator"

$$h_C2H6_ex_HP_ev = \text{enthalpy}(\text{ethane}, T=t_mr_ex_HP_ev, p=p_mr_ev)$$

"enthalpy of ethane component of mixed refrigerant exhaust from HP mr_evaporator"

$$h_C3H8_ex_HP_ev = \text{enthalpy}(\text{propane}, T=t_mr_ex_HP_ev, p=p_mr_ev)$$

"enthalpy of butane component of mixed refrigerant exhaust from HP mr_evaporator"

"Calculate evaporator duty"

$$Q_dot_N2_HP_ev = M_dot_N2 * (h_N2_su_HP_ev - h_N2_ex_HP_ev)$$

"heat duty of nitrogen component of mixed refrigerant HP mr_evaporator"

$$Q_dot_CH4_HP_ev = M_dot_CH4 * (h_CH4_su_HP_ev - h_CH4_ex_HP_ev)$$

"heat duty of methane component of mixed refrigerant HP mr_evaporator"

$$Q_dot_C2H6_HP_ev = M_dot_C2H6 * (h_C2H6_su_HP_ev - h_C2H6_ex_HP_ev)$$

"heat duty of ethane component of mixed refrigerant HP mr_evaporator"

$$Q_dot_C3H8_HP_ev = M_dot_C3H8 * (h_C3H8_su_HP_ev - h_C3H8_ex_HP_ev)$$

"heat duty of butane component of mixed refrigerant HP mr_evaporator"

$Q_{\text{dot_HP_mr_ev}} =$
 $\text{SUM}(Q_{\text{dot_N2_HP_ev}}, Q_{\text{dot_CH4_HP_ev}}, Q_{\text{dot_C2H6_HP_ev}}, Q_{\text{dot_C3H8_HP_ev}})$
 "heat duty of HHP mr_evaporator"

!"HP Compressor"

"Calculate exhaust enthalpy (based on energy balance at inlet of HP compressor)"

$h_{\text{r_su_HP_cp}} = (m_{\text{dot_r_HP_ev}} * h_{\text{r_ex_HP_ev}}$
 $+ m_{\text{dot_r_HP_mr_ev}} * h_{\text{r_ex_HP_mr_ev}} + m_{\text{dot_r_MP_cp}} * h_{\text{r_ex_MP_cp}}) / m_{\text{dot_r_HP_c}}$
 p

"mass balance at inlet of HP compressor"

$m_{\text{dot_r_HP_cp}} = m_{\text{dot_r_HP_ev}} + m_{\text{dot_r_HP_mr_ev}} + m_{\text{dot_r_MP_cp}}$

"mass flow rate of propane refrigerant through HP compressor"

$s_{\text{r_su_HP_cp}} = \text{ENTROPY}(\text{fluid1}, h = h_{\text{r_su_HP_cp}}, P = P_{\text{HP_ev}})$

"entropy of propane refrigerant supply to HP compressor"

$h_{\text{r_ex_HP_cp_s}} = \text{ENTHALPY}(\text{fluid1}, P = P_{\text{HHP_ev}}, s = s_{\text{r_su_HP_cp}})$

"HP compressor exhaust enthalpy at isentropic condition"

$\epsilon_{\text{s_HP_cp}} = (h_{\text{r_ex_HP_cp_s}} - h_{\text{r_su_HP_cp}}) / (h_{\text{r_ex_HP_cp}} - h_{\text{r_su_HP_cp}})$

"isentropic efficiency of HP compressor"

$W_{\text{dot_HP_cp}} = m_{\text{dot_r_HP_cp}} * (h_{\text{r_ex_HP_cp}} - h_{\text{r_su_HP_cp}})$

"HP compressor work"

!"MP expansion valve"

$h_{\text{r_ex_MP_expv}} = h_{\text{r_ex_HP_expv}}$

"enthalpy of propane refrigerant exhaust from MP expansion valve"

!"MP evaporator"

"supply"

$h_{\text{r_su_MP_ev}} = h_{\text{r_ex_MP_expv}}$

"enthalpy of propane refrigerant supply to MP evaporator"

$t_{\text{ng_su_MP_ev}} = t_{\text{ng_ex_HP_ev}}$

"temperature of natural gas supply to MP evaporator" (no heat loss)"

$P_{\text{ng_su_MP_ev}} = P_{\text{ng_ex_HP_ev}}$

"pressure of natural gas supply to MP evaporator" (no head loss)"

"refrigerant side"

$Q_{\text{dot_MP_ev}} = m_{\text{dot_r_MP_ev}} * (h_{\text{r_ex_MP_ev}} - h_{\text{r_su_MP_ev}})$

"heat duty of MP evaporator"

$h_{\text{r_ex_MP_ev}} = \text{ENTHALPY}(\text{fluid1}, P = P_{\text{MP_ev}}, t = t_{\text{r_ex_MP_ev}})$

"enthalpy of propane refrigerant exhaust from MP evaporator"

$t_{r_ex_MP_ev}=t_{MP_ev}+\Delta T_{at_oh}$
 "temperature of propane refrigerant exhaust from MP evaporator (including superheat)"
 $t_{MP_ev}=\text{TEMPERATURE}(\text{fluid1}\$,P=P_{MP_ev},x=1)$
 "temperature of propane refrigerant exhaust from MP evaporator (excluding superheat)"
 "natural gas side"
 $Q_{dot_MP_ev}=m_{dot_ng_ev}*(h_{ng_su_MP_ev}-h_{ng_ex_MP_ev})$
 "heat duty of MP mr_evaporator"
 $P_{ng_ex_MP_ev}=P_{ng_su_MP_ev}$
 "enthalpy of propane refrigerant exhaust from MP evaporator" "(no head loss)"
 $h_{ng_su_MP_ev}=\text{ENTHALPY}(\text{fluid2}\$,t=t_{ng_su_MP_ev},P=P_{ng_su_MP_ev})$
 "enthalpy of natural gas supply to MP evaporator"
 "heat exchange effectiveness"
 $Q_{dot_MP_ev}=m_{dot_ng_ev}*c_{p_ng_MP_ev}*\epsilon_{MP_ev}*(t_{ng_su_MP_ev}-t_{MP_ev})$
 "heat duty of MP evaporator"
 $c_{p_ng_MP_ev}=\text{CP}(\text{fluid2}\$,t=t_{ng_su_MP_ev},P=P_{ng_su_MP_ev})$
 "specific heat of natural gas supply to MP evaporator"
 $\epsilon_{MP_ev}=1-\exp(-NTU_{MP_ev})$
 "effectiveness of MP evaporator (semi-isothermal heat exchange)"
 $NTU_{MP_ev}=AU_{MP_ev}/(m_{dot_ng_ev}*c_{p_ng_MP_ev})$
 "number of transfer units of MP evaporator"
 $AU_{MP_ev}=AU_{MP_ev_n}$
 "heat transfer coefficient-area product of MP propane evaporator (modification needed to take effect of flow rate variations)"
 $t_{ng_ex_MP_ev}=\text{TEMPERATURE}(\text{fluid2}\$,h=h_{ng_ex_MP_ev},P=P_{ng_ex_MP_ev})$
 "temperature of natural gas exhaust from MP evaporator"

 "MP mr_evaporator"
 "supply"
 $h_{r_su_MP_mr_ev}=h_{r_ex_MP_expv}$
 "enthalpy of propane refrigerant supply to MP mr_evaporator"
 "refrigerant side"
 $Q_{dot_MP_mr_ev}=m_{dot_r_MP_mr_ev}*(h_{r_ex_MP_mr_ev}-h_{r_su_MP_mr_ev})$
 "heat duty of MP mr_evaporator"
 $h_{r_ex_MP_mr_ev}=\text{ENTHALPY}(\text{fluid1}\$,P=P_{MP_mr_ev},t=t_{r_ex_MP_mr_ev})$
 "enthalpy of propane refrigerant exhaust from MP mr_evaporator"
 $P_{MP_mr_ev}=P_{MP_ev}$
 "pressure of natural gas supply to MP evaporator"
 $t_{r_ex_MP_mr_ev}=t_{MP_mr_ev}+\Delta T_{at_oh}$
 "temperature of propane refrigerant exhaust from MP mr_evaporator (including superheat)"

$t_{MP_mr_ev} = \text{TEMPERATURE}(\text{fluid1}\$, P=P_{MP_mr_ev}, x=1)$
 "temperature of propane refrigerant exhaust from MP mr_evaporator (excluding superheat)"
 "mixed refrigerant side"
 "Calculation of the supply enthalpies:"
 $h_{N2_su_MP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_su_MP_ev}, p=p_{mr_ev})$
 "enthalpy of nitrogen component of mixed refrigerant supply to MP mr_evaporator"
 $h_{CH4_su_MP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_su_MP_ev}, p=p_{mr_ev})$
 "enthalpy of methane component of mixed refrigerant supply to MP mr_evaporator"
 $h_{C2H6_su_MP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_su_MP_ev}, p=p_{mr_ev})$
 "enthalpy of ethane component of mixed refrigerant supply to MP mr_evaporator"
 $h_{C3H8_su_MP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_su_MP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant supply to MP mr_evaporator"
 "Calculation of the exhaust enthalpies:"
 $h_{N2_ex_MP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of nitrogen component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{CH4_ex_MP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of methane component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{C2H6_ex_MP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of ethane component of mixed refrigerant exhaust from MP mr_evaporator"
 $h_{C3H8_ex_MP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_MP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant exhaust from MP mr_evaporator"
 "Calculate evaporator duty"
 $Q_{dot_N2_MP_ev} = M_{dot_N2} * (h_{N2_su_MP_ev} - h_{N2_ex_MP_ev})$
 "heat duty of nitrogen component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_CH4_MP_ev} = M_{dot_CH4} * (h_{CH4_su_MP_ev} - h_{CH4_ex_MP_ev})$
 "heat duty of methane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_C2H6_MP_ev} = M_{dot_C2H6} * (h_{C2H6_su_MP_ev} - h_{C2H6_ex_MP_ev})$
 "heat duty of ethane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_C3H8_MP_ev} = M_{dot_C3H8} * (h_{C3H8_su_MP_ev} - h_{C3H8_ex_MP_ev})$
 "heat duty of butane component of mixed refrigerant MP mr_evaporator"
 $Q_{dot_MP_mr_ev} =$
 $\text{SUM}(Q_{dot_N2_MP_ev}, Q_{dot_CH4_MP_ev}, Q_{dot_C2H6_MP_ev}, Q_{dot_C3H8_MP_ev})$
 "heat duty of MP mr_evaporator"

 "**!MP Compressor**"
 "Calculate exhaust enthalpy (based on energy balance at inlet of MP compressor)"
 $h_{r_su_MP_cp} = (m_{dot_r_MP_ev} * h_{r_ex_MP_ev} + m_{dot_r_MP_mr_ev} * h_{r_ex_MP_mr_ev} + m_{dot_r_LP_cp} * h_{r_ex_LP_cp}) / m_{dot_r_MP_cp}$
 "mass balance at inlet of MP compressor"

$$m_dot_r_MP_cp=m_dot_r_MP_ev+m_dot_r_MP_mr_ev+m_dot_r_LP_cp$$

"mass flow rate of propane refrigerant through MP compressor"

$$s_r_su_MP_cp=ENTROPY(fluid1$,h=h_r_su_MP_cp,P=P_MP_ev)$$

"entropy of propane refrigerant supply to MP compressor"

$$h_r_ex_MP_cp_s=ENTHALPY(fluid1$,P=P_HP_ev,s=s_r_su_MP_cp)$$

"MP compressor exhaust enthalpy at isentropic condition"

$$\epsilon_s_MP_cp=(h_r_ex_MP_cp_s-h_r_su_MP_cp)/(h_r_ex_MP_cp-h_r_su_MP_cp)$$

"isentropic efficiency of MP compressor"

$$W_dot_MP_cp=m_dot_r_MP_cp*(h_r_ex_MP_cp-h_r_su_MP_cp)$$

"MP compressor work"

"!LP expansion valve"

$$h_r_ex_LP_expv=h_r_ex_MP_expv$$

"enthalpy of propane refrigerant exhaust from HHP expansion valve"

"!LP evaporator"

"supply"

$$h_r_su_LP_ev=h_r_ex_LP_expv$$

"enthalpy of propane refrigerant supply to LP evaporator"

$$t_ng_su_LP_ev=t_ng_ex_MP_ev$$

"temperature of natural gas supply to LP evaporator"(no heat loss)"

$$P_ng_su_LP_ev=P_ng_ex_MP_ev$$

"pressure of natural gas supply to LP evaporator"(no head loss)"

"refrigerant side"

$$Q_dot_LP_ev=m_dot_r_LP_ev*(h_r_ex_LP_ev-h_r_su_LP_ev)$$

"heat duty of LP evaporator"

$$h_r_ex_LP_ev=ENTHALPY(fluid1$,P=P_LP_ev,t=t_r_ex_LP_ev)$$

"enthalpy of propane refrigerant exhaust from LP evaporator"

$$t_r_ex_LP_ev=t_LP_ev+DELTA t_{oh}$$

"temperature of propane refrigerant exhaust from LP evaporator (including superheat)"

$$t_LP_ev=TEMPERATURE(fluid1$,P=P_LP_ev,x=1)$$

"temperature of propane refrigerant exhaust from LP evaporator (excluding superheat)"

"natural gas side"

$$Q_dot_LP_ev=m_dot_ng_ev*(h_ng_su_LP_ev-h_ng_ex_LP_ev)$$

"heat duty of LP mr_evaporator"

$$P_ng_ex_LP_ev=P_ng_su_LP_ev$$

"enthalpy of propane refrigerant exhaust from LP evaporator" "(no head loss)"

$$h_ng_su_LP_ev=ENTHALPY(fluid2$,t=t_ng_su_LP_ev,P=P_ng_su_LP_ev)$$

"enthalpy of natural gas supply to LP evaporator"

"heat exchange effectiveness"

$$Q_dot_LP_ev=m_dot_ng_ev*c_p_ng_LP_ev*epsilon_LP_ev*(t_ng_su_LP_ev-t_LP_ev)$$

"heat duty of LP evaporator"

$$c_p_ng_LP_ev=CP(fluid2$,t=t_ng_su_LP_ev,P=P_ng_su_LP_ev)$$

"specific heat of natural gas supply to LP evaporator"

$$epsilon_LP_ev=1-exp(-NTU_LP_ev)$$

"effectiveness of LP evaporator (semi-isothermal heat exchange)"

$$NTU_LP_ev=AU_LP_ev/(m_dot_ng_ev*c_p_ng_LP_ev)$$

"number of transfer units of LP evaporator"

$$AU_LP_ev=AU_LP_ev_n$$

"heat transfer coefficient-area product of LP propane evaporator (modification needed to take effect of flow rate variations)"

$$t_ng_ex_LP_ev=TEMPERATURE(fluid2$,h=h_ng_ex_LP_ev,P=P_ng_ex_LP_ev)$$

"temperature of natural gas exhaust from LP evaporator"

"!LP mr_evaporator"

"supply"

$$h_r_su_LP_mr_ev=h_r_ex_LP_expv$$

"enthalpy of propane refrigerant supply to LP mr_evaporator"

"refrigerant side"

$$Q_dot_LP_mr_ev=m_dot_r_LP_mr_ev*(h_r_ex_LP_mr_ev-h_r_su_LP_mr_ev)$$

"heat duty of LP mr_evaporator"

$$h_r_ex_LP_mr_ev=ENTHALPY(fluid1$,P=P_LP_mr_ev,t=t_r_ex_LP_mr_ev)$$

"enthalpy of propane refrigerant exhaust from LP mr_evaporator"

$$P_LP_mr_ev=P_LP_ev$$

"pressure of natural gas supply to LP evaporator"

$$t_r_ex_LP_mr_ev=t_LP_mr_ev+DELTA t_oh$$

"temperature of propane refrigerant exhaust from LP mr_evaporator (including superheat)"

$$t_LP_mr_ev=TEMPERATURE(fluid1$,P=P_LP_mr_ev,x=1)$$

"temperature of propane refrigerant exhaust from LP mr_evaporator (excluding superheat)"

"mixed refrigerant side"

"Calculation of the supply enthalpies:"

$$h_N2_su_LP_ev = enthalpy(nitrogen,T=t_mr_su_LP_ev,p=p_mr_ev)$$

"enthalpy of nitrogen component of mixed refrigerant supply to LP mr_evaporator"

$$h_CH4_su_LP_ev = enthalpy(methane,T=t_mr_su_LP_ev,p=p_mr_ev)$$

"enthalpy of methane component of mixed refrigerant supply to LP mr_evaporator"

$$h_C2H6_su_LP_ev = enthalpy(ethane,T=t_mr_su_LP_ev,p=p_mr_ev)$$

"enthalpy of ethane component of mixed refrigerant supply to LP mr_evaporator"

$h_{C3H8_su_LP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_su_LP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant supply to LP mr_evaporator"
 "Calculation of the exhaust enthalpies:"
 $h_{N2_ex_LP_ev} = \text{enthalpy}(\text{nitrogen}, T=t_{mr_ex_LP_ev}, p=p_{mr_ev})$
 "enthalpy of nitrogen component of mixed refrigerant exhaust from LP mr_evaporator"
 $h_{CH4_ex_LP_ev} = \text{enthalpy}(\text{methane}, T=t_{mr_ex_LP_ev}, p=p_{mr_ev})$
 "enthalpy of methane component of mixed refrigerant exhaust from LP mr_evaporator"
 $h_{C2H6_ex_LP_ev} = \text{enthalpy}(\text{ethane}, T=t_{mr_ex_LP_ev}, p=p_{mr_ev})$
 "enthalpy of ethane component of mixed refrigerant exhaust from LP mr_evaporator"
 $h_{C3H8_ex_LP_ev} = \text{enthalpy}(\text{propane}, T=t_{mr_ex_LP_ev}, p=p_{mr_ev})$
 "enthalpy of butane component of mixed refrigerant exhaust from LP mr_evaporator"
 "Calculate evaporator duty"
 $Q_{dot_N2_LP_ev} = M_{dot_N2} * (h_{N2_su_LP_ev} - h_{N2_ex_LP_ev})$
 "heat duty of nitrogen component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_CH4_LP_ev} = M_{dot_CH4} * (h_{CH4_su_LP_ev} - h_{CH4_ex_LP_ev})$
 "heat duty of methane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_C2H6_LP_ev} = M_{dot_C2H6} * (h_{C2H6_su_LP_ev} - h_{C2H6_ex_LP_ev})$
 "heat duty of ethane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_C3H8_LP_ev} = M_{dot_C3H8} * (h_{C3H8_su_LP_ev} - h_{C3H8_ex_LP_ev})$
 "heat duty of butane component of mixed refrigerant LP mr_evaporator"
 $Q_{dot_LP_mr_ev} =$
 $SUM(Q_{dot_N2_LP_ev}, Q_{dot_CH4_LP_ev}, Q_{dot_C2H6_LP_ev}, Q_{dot_C3H8_LP_ev})$
 "heat duty of LP mr_evaporator"

 "!!LP Compressor"
 "Calculate exhaust enthalpy (based on energy balance at inlet of LP compressor)"
 $h_{r_su_LP_cp} = (m_{dot_r_LP_ev} * h_{r_ex_LP_ev} + m_{dot_r_LP_mr_ev} * h_{r_ex_LP_mr_ev}) / m_{dot_r_LP_cp}$
 "mass balance at inlet of LP compressor"
 $m_{dot_r_LP_cp} = m_{dot_r_LP_ev} + m_{dot_r_LP_mr_ev}$
 "mass flow rate of propane refrigerant through LP compressor"
 $s_{r_su_LP_cp} = \text{ENTROPY}(\text{fluid1}, h=h_{r_su_LP_cp}, P=P_{LP_ev})$
 "entropy of propane refrigerant supply to LP compressor"
 $h_{r_ex_LP_cp_s} = \text{ENTHALPY}(\text{fluid1}, P=P_{MP_ev}, s=s_{r_su_LP_cp})$
 "LP compressor exhaust enthalpy at isentropic condition"
 $\epsilon_{s_LP_cp} = (h_{r_ex_LP_cp_s} - h_{r_su_LP_cp}) / (h_{r_ex_LP_cp} - h_{r_su_LP_cp})$
 "isentropic efficiency of LP compressor"
 $W_{dot_LP_cp} = m_{dot_r_LP_cp} * (h_{r_ex_LP_cp} - h_{r_su_LP_cp})$
 "LP compressor work"

!1.6. Performances

"Calculate total evaporator duty (heat load) of propane refrigerant cycle"

$$Q_{\text{dot_tot_ev}} = (Q_{\text{dot_HHP_ev}} + Q_{\text{dot_HP_ev}} + Q_{\text{dot_MP_ev}} + Q_{\text{dot_LP_ev}}) + (Q_{\text{dot_HHP_mr_ev}} + Q_{\text{dot_HP_mr_ev}} + Q_{\text{dot_MP_mr_ev}} + Q_{\text{dot_LP_mr_ev}})$$

"Calculate total work of propane refrigerant compressor"

$$W_{\text{dot_tot_cp}} = W_{\text{dot_HHP_cp}} + W_{\text{dot_HP_cp}} + W_{\text{dot_MP_cp}} + W_{\text{dot_LP_cp}}$$

"Calculate the coefficient of performance of propane refrigerant compressor"

$$\text{COP} = Q_{\text{dot_tot_ev}} / W_{\text{dot_tot_cp}} \quad \text{"useful heat/network input"}$$

"Propane compressor power (Name plate / design) value"

$$W_{\text{dot_tot_cp_design}} = 41600000 \text{ [W]}$$

!Property Plot data

"Properties to be plotted are placed in arrays using square brackets to enclose the array index. In this way, the property values are displayed in the Array window after calculations are completed and they can then be directly plotted"

$$p[1] = p_{\text{cd}}$$

"pressure of condenser "

$$h[1] = h_{\text{r_ex_HHP_cp}}$$

"enthalpy of refrigerant exhaust from HHP compressor"

$$s[1] = \text{ENTROPY}(\text{fluid1}\$, h=h[1], P=P[1])$$

"entropy of refrigerant exhaust from HHP compressor"

$$t[1] = \text{TEMPERATURE}(\text{fluid1}\$, h=h[1], P=P[1])$$

"temperature of refrigerant exhaust from HHP compressor"

$$p[2] = p_{\text{cd}}$$

"pressure of condenser "

$$h[2] = h_{\text{r_ex_cd}}$$

"enthalpy of refrigerant exhaust from condenser"

$$s[2] = \text{ENTROPY}(\text{fluid1}\$, h=h[2], P=P[2])$$

"entropy of refrigerant exhaust from condenser"

$$t[2] = \text{TEMPERATURE}(\text{fluid1}\$, h=h[2], P=P[2])$$

"temperature of refrigerant exhaust from condenser"

$$p[3] = p_{\text{HHP_ev}}$$

"pressure of HHP evaporator"

$$h[3] = h_{\text{r_su_HHP_ev}}$$

"enthalpy of refrigerant supply to HHP evaporator"

$s[3]=\text{ENTROPY}(\text{fluid1}\$,h=h[3],P=P[3])$
 "entropy of refrigerant supply to HHP evaporator"
 $t[3]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[3],P=P[3])$
 "temperature of refrigerant supply to HHP evaporator"
 $p[4]=p_{\text{HHP_ev}}$
 "pressure of HHP evaporator"
 $h[4]=h_{\text{r_ex_HHP_ev}}$
 "enthalpy of refrigerant exhaust from HHP evaporator"
 $s[4]=\text{ENTROPY}(\text{fluid1}\$,h=h[4],P=P[4])$
 "entropy of refrigerant exhaust from HHP evaporator"
 $t[4]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[4],P=P[4])$
 "temperature of refrigerant exhaust from HHP evaporator"
 $p[5]=p_{\text{HHP_ev}}$
 "pressure of HHP evaporator"
 $h[5]=h_{\text{r_ex_HP_cp}}$
 "enthalpy of refrigerant exhaust from HP compressor"
 $s[5]=\text{ENTROPY}(\text{fluid1}\$,h=h[5],P=P[5])$
 "entropy of refrigerant exhaust from HP compressor"
 $t[5]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[5],P=P[5])$
 "temperature of refrigerant exhaust from HP compressor"
 $p[6]=p_{\text{HHP_ev}}$
 "pressure of HHP evaporator"
 $h[6]=h_{\text{r_su_HHP_cp}}$
 "enthalpy of refrigerant supply to HHP compressor"
 $s[6]=\text{ENTROPY}(\text{fluid1}\$,h=h[6],P=P[6])$
 "entropy of refrigerant supply to HHP compressor"
 $t[6]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[6],P=P[6])$
 "temperature of refrigerant supply to HHP compressor"
 $p[7]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"
 $h[7]=h_{\text{r_su_HP_ev}}$
 "enthalpy of refrigerant supply to HP evaporator"
 $s[7]=\text{ENTROPY}(\text{fluid1}\$,h=h[7],P=P[7])$
 "entropy of refrigerant supply to HP evaporator"
 $t[7]=\text{TEMPERATURE}(\text{fluid1}\$,h=h[7],P=P[7])$
 "temperature of refrigerant supply to HP evaporator"
 $p[8]=p_{\text{HP_ev}}$
 "pressure of HP evaporator"

$h[8]=h_{r_ex_HP_ev}$
 "enthalpy of refrigerant exhaust from HP evaporator"
 $s[8]=ENTROPY(fluid1\$,h=h[8],P=P[8])$
 "entropy of refrigerant exhaust from HP evaporator"
 $t[8]=TEMPERATURE(fluid1\$,h=h[8],P=P[8])$
 "temperature of refrigerant exhaust from HP evaporator"
 $p[9]=p_{HP_ev}$
 "pressure of HP evaporator"
 $h[9]=h_{r_ex_MP_cp}$
 "enthalpy of refrigerant exhaust from MP compressor"
 $s[9]=ENTROPY(fluid1\$,h=h[9],P=P[9])$
 "entropy of refrigerant exhaust from MP compressor"
 $t[9]=TEMPERATURE(fluid1\$,h=h[9],P=P[9])$
 "temperature of refrigerant exhaust from MP compressor"
 $p[10]=p_{HP_ev}$
 "pressure of HP evaporator"
 $h[10]=h_{r_su_HP_cp}$
 "enthalpy of refrigerant supply to HP compressor"
 $s[10]=ENTROPY(fluid1\$,h=h[10],P=P[10])$
 "entropy of refrigerant supply to HP compressor"
 $t[10]=TEMPERATURE(fluid1\$,h=h[10],P=P[10])$
 "temperature of refrigerant supply to HP compressor"
 $p[11]=p_{MP_ev}$
 "pressure of MP evaporator"
 $h[11]=h_{r_su_MP_ev}$
 "enthalpy of refrigerant supply to MP evaporator"
 $s[11]=ENTROPY(fluid1\$,h=h[11],P=P[11])$
 "entropy of refrigerant supply to MP evaporator"
 $t[11]=TEMPERATURE(fluid1\$,h=h[11],P=P[11])$
 "temperature of refrigerant supply to MP evaporator"
 $p[12]=p_{MP_ev}$
 "pressure of MP evaporator"
 $h[12]=h_{r_ex_MP_ev}$
 "enthalpy of refrigerant exhaust from MP evaporator"
 $s[12]=ENTROPY(fluid1\$,h=h[12],P=P[12])$
 "entropy of refrigerant exhaust from MP evaporator"
 $t[12]=TEMPERATURE(fluid1\$,h=h[12],P=P[12])$
 "temperature of refrigerant exhaust from MP evaporator"

$p[13]=p_MP_ev$
 "pressure of MP evaporator"
 $h[13]=h_r_ex_LP_cp$
 "enthalpy of refrigerant exhaust from LP compressor"
 $s[13]=ENTROPY(fluid1\$,h=h[13],P=P[13])$
 "entropy of refrigerant exhaust from LP compressor"
 $t[13]=TEMPERATURE(fluid1\$,h=h[13],P=P[13])$
 "temperature of refrigerant exhaust from LP compressor"
 $p[14]=p_MP_ev$
 "pressure of MP evaporator"
 $h[14]=h_r_su_MP_cp$
 "enthalpy of refrigerant supply to MP compressor"
 $s[14]=ENTROPY(fluid1\$,h=h[14],P=P[14])$
 "entropy of refrigerant supply to MP compressor"
 $t[14]=TEMPERATURE(fluid1\$,h=h[14],P=P[14])$
 "temperature of refrigerant supply to MP compressor"
 $p[15]=p_LP_ev$
 "pressure of LP evaporator"
 $h[15]=h_r_su_LP_ev$
 "enthalpy of refrigerant supply to LP evaporator"
 $s[15]=ENTROPY(fluid1\$,h=h[15],P=P[15])$
 "entropy of refrigerant supply to LP evaporator"
 $t[15]=TEMPERATURE(fluid1\$,h=h[15],P=P[15])$
 "temperature of refrigerant supply to LP evaporator"
 $p[16]=p_LP_ev$
 "pressure of LP evaporator"
 $h[16]=h_r_ex_LP_ev$
 "enthalpy of refrigerant exhaust from LP evaporator"
 $s[16]=ENTROPY(fluid1\$,h=h[16],P=P[16])$
 "entropy of refrigerant exhaust from LP evaporator"
 $t[16]=TEMPERATURE(fluid1\$,h=h[16],P=P[16])$
 "temperature of refrigerant exhaust from LP evaporator"
 $p[17]=p_LP_ev$
 "pressure of LP evaporator"
 $h[17]=h_r_su_LP_cp$
 "enthalpy of refrigerant supply to LP compressor"
 $s[17]=ENTROPY(fluid1\$,h=h[17],P=P[17])$
 "entropy of refrigerant supply to LP compressor"


```

t[17]=TEMPERATURE(fluid1$,h=h[17],P=P[17])
"temperature of refrigerant supply to LP compressor"
p[18]=p_LP_mr_ev
"pressure of LP mr_evaporator"
h[18]=h_r_ex_LP_mr_ev
"enthalpy of refrigerant exhaust from LP mr_evaporator"
s[18]=ENTROPY(fluid1$,h=h[18],P=P[18])
"entropy of refrigerant exhaust from LP evaporator"
t[18]=TEMPERATURE(fluid1$,h=h[18],P=P[18])
"temperature of refrigerant exhaust from LP evaporator"
p[19]=p_MP_mr_ev
"pressure of MP mr_evaporator"
h[19]=h_r_ex_MP_mr_ev
"enthalpy of refrigerant exhaust from MP evaporator"
s[19]=ENTROPY(fluid1$,h=h[19],P=P[19])
"entropy of refrigerant exhaust from MP evaporator"
t[19]=TEMPERATURE(fluid1$,h=h[19],P=P[19])
"temperature of refrigerant exhaust from MP evaporator"
p[20]=p_HP_mr_ev
"pressure of HP mr_evaporator"
h[20]=h_r_ex_HP_mr_ev
"enthalpy of refrigerant exhaust from HP mr_evaporator"
s[20]=ENTROPY(fluid1$,h=h[20],P=P[20])
"entropy of refrigerant exhaust from HP mr_evaporator"
t[20]=TEMPERATURE(fluid1$,h=h[20],P=P[20])
"temperature of refrigerant exhaust from HP mr_evaporator"
p[21]=p_HHP_mr_ev
"pressure of HHP mr_evaporator"
h[21]=h_r_ex_HHP_mr_ev
"enthalpy of refrigerant exhaust from HHP mr_evaporator"
s[21]=ENTROPY(fluid1$,h=h[21],P=P[21])
"entropy of refrigerant exhaust from HHP mr_evaporator"
t[21]=TEMPERATURE(fluid1$,h=h[21],P=P[21])
"temperature of refrigerant exhaust from HHP mr_evaporator"

"Enthalpy check for propane cycle"
duplicate i=1,21
  hh[i] = Enthalpy(fluid1$,T=T[i],s=s[i])
end

```

"!2. MIXED REFRIGERANT CYCLE"

"Note: The main cryogenic heat exchanger is the evaporator of the mixed refrigerant cycle"

"Note: The propane cycle mr_evaporators make up the condenser of the mixed refrigerant cycle"

"!2.1. Inputs"

"Molar mass of mixed refrigerant"

$MM_{mix} = (X_{N2} * MM_{N2}) + (X_{ch4} * MM_{ch4}) + (X_{c2h6} * MM_{c2h6}) + (X_{C3h8} * MM_{C3H8})$ "molecular weight of mixed refrigerant"

"Mixed refrigerant Inputs"

$M_{dot_tot} = 274$ [kg/s]

"total mass flow rate of mixed refrigerant"

$T_{su_ev} = -31.6$ [C]

"temperature of mixed refrigerant supply to evaporator"

$p_{su_ev} = 44.65E5$ [Pa]

"pressure of mixed refrigerant supply to evaporator"

$T_{ex_ev} = -34.5$ [C]

"temperature of mixed refrigerant exhaust from evaporator"

$p_{ex_ev} = 3E5$ [Pa]

"pressure of mixed refrigerant exhaust from evaporator"

"water side inputs"

$M_{dot_w_int1} = (m_{dot_w_cd} / 3.8)$

"mass flow of cooling water supply to mixed refrigerant compressor intercooler 1"

$M_{dot_w_int2} = (m_{dot_w_cd} / 5.9)$

"mass flow of cooling water supply to mixed refrigerant compressor intercooler 2"

$M_{dot_w_aft} = (m_{dot_w_cd} / 5.5)$

"mass flow of cooling water supply to mixed refrigerant compressor aftercooler"

$t_{w_su_int1} = t_{w_ex}$

"temperature of cooling water supply to intercooler 1"

$t_{w_su_int2} = t_{w_ex}$

"temperature of cooling water supply to intercooler 2"

$t_{w_su_aft} = t_{w_ex}$

"temperature of cooling water supply to aftercooler"

$t_{w_ex_int1} = t_{w_su}$

"temperature of cooling water exhaust from intercooler 1"

$t_{w_ex_int2}=t_{w_su}$

"temperature of cooling water exhaust from intercooler 2"

$t_{w_ex_aft}=t_{w_su}$

"temperature of cooling water exhaust from aftercooler "

"Mixed refrigerant compressor power (Name plate / design) value"

$W_{dot_comp_design} = 86600000$ [W]

!"2.2. Outputs"

" Q_{dot_ev} "

"evaporator heat transfer (duty)"

" Q_{dot_cd} "

"condenser heat transfer (duty)"

!"2.3. Equations"

!"Mixed Refrigerant Evaporator (cryogenic heat exchanger)"

"Calculation of the supply enthalpies:"

$h_{n2_su_ev} = \text{enthalpy}(\text{nitrogen}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of nitrogen component of mixed refrigerant supply to evaporator"

$h_{ch4_su_ev} = \text{enthalpy}(\text{methane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of methane component of mixed refrigerant supply to evaporator"

$h_{c2h6_su_ev} = \text{enthalpy}(\text{ethane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of ethane component of mixed refrigerant supply to evaporator"

$h_{c3h8_su_ev} = \text{enthalpy}(\text{propane}, T=T_{su_ev}, p=p_{su_ev})$

"enthalpy of propane component of mixed refrigerant supply to evaporator"

"Calculation of the exhaust enthalpies:"

$h_{n2_ex_ev} = \text{enthalpy}(\text{nitrogen}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of nitrogen component of mixed refrigerant exhaust from evaporator"

$h_{ch4_ex_ev} = \text{enthalpy}(\text{methane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of methane component of mixed refrigerant exhaust from evaporator"

$h_{c2h6_ex_ev} = \text{enthalpy}(\text{ethane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of ethane component of mixed refrigerant exhaust from evaporator"

$h_{c3h8_ex_ev} = \text{enthalpy}(\text{propane}, T=T_{ex_ev}, p=p_{ex_ev})$

"enthalpy of propane component of mixed refrigerant exhaust from evaporator"

"Calculate evaporator (cryogenic heat exchanger) duty"

$Q_{dot_n2_ev} = M_{dot_n2} * (h_{n2_ex_ev} - h_{n2_su_ev})$

"heat duty of nitrogen component of mixed refrigerant evaporator"

$Q_{dot_ch4_ev} = M_{dot_ch4} * (h_{ch4_ex_ev} - h_{ch4_su_ev})$

"heat duty of methane component of mixed refrigerant evaporator"

$$Q_{\text{dot_c2h6_ev}} = M_{\text{dot_c2h6}} * (h_{\text{c2h6_ex_ev}} - h_{\text{c2h6_su_ev}})$$

"heat duty of ethane component of mixed refrigerant evaporator"

$$Q_{\text{dot_c3h8_ev}} = M_{\text{dot_c3h8}} * (h_{\text{c3h8_ex_ev}} - h_{\text{c3h8_su_ev}})$$

"heat duty of propane component of mixed refrigerant evaporator"

$$Q_{\text{dot_tot_ev_mix}} = \text{SUM}(Q_{\text{dot_n2_ev}}, Q_{\text{dot_ch4_ev}}, Q_{\text{dot_c2h6_ev}}, Q_{\text{dot_c3h8_ev}})$$

"heat duty of evaporator"

"!Mixed Refrigerant Condenser"

"Calculate mixed refrigerant condenser duty"

$$Q_{\text{dot_tot_cd_mix}} =$$

$$\text{SUM}(Q_{\text{dot_HHP_mr_ev}}, Q_{\text{dot_HP_mr_ev}}, Q_{\text{dot_MP_mr_ev}}, Q_{\text{dot_LP_mr_ev}})$$

"! Cooling water heat flow :"

$$c_{\text{w_p}} = 4190 \text{ [J/kg-K]} \text{ "specific heat capacity of water"}$$

$$Q_{\text{dot_w_int1}} = M_{\text{dot_w_int1}} * c_{\text{w_p}} * (t_{\text{w_ex_int1}} - t_{\text{w_su_int1}})$$

"heat duty of cooling water supply to intercooler 1"

$$Q_{\text{dot_w_int2}} = M_{\text{dot_w_int2}} * c_{\text{w_p}} * (t_{\text{w_ex_int2}} - t_{\text{w_su_int2}})$$

"heat duty of cooling water supply to intercooler 2"

$$Q_{\text{dot_w_aft}} = M_{\text{dot_w_aft}} * c_{\text{w_p}} * (t_{\text{w_ex_aft}} - t_{\text{w_su_aft}})$$

"heat duty of cooling water supply to aftercooler "

$$\{Q_{\text{dot_tot_w}} = \text{SUM}(Q_{\text{dot_w_int1}}, Q_{\text{dot_w_int2}}, Q_{\text{dot_w_aft}})\}$$

"!Compressor"

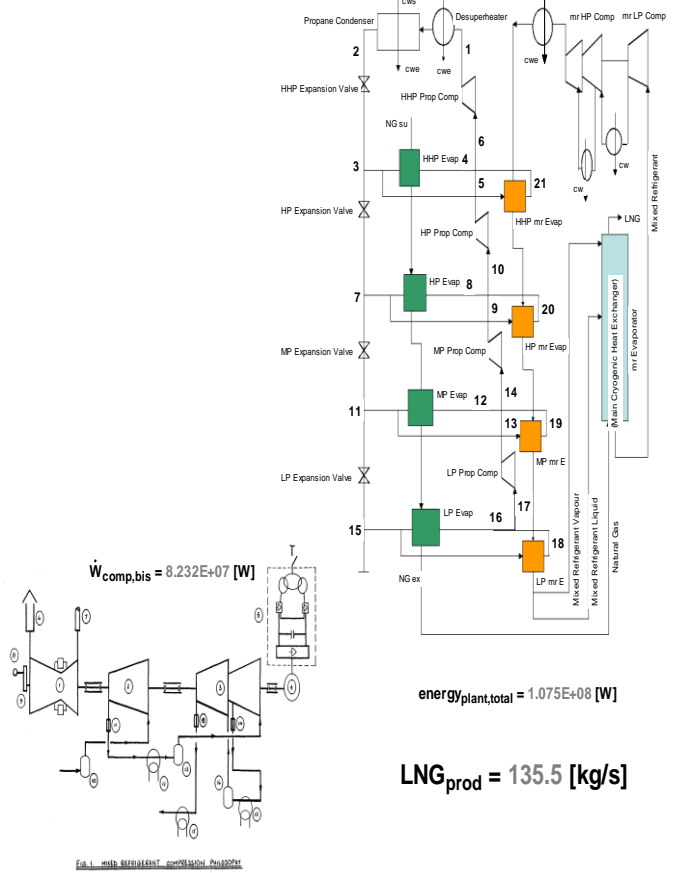
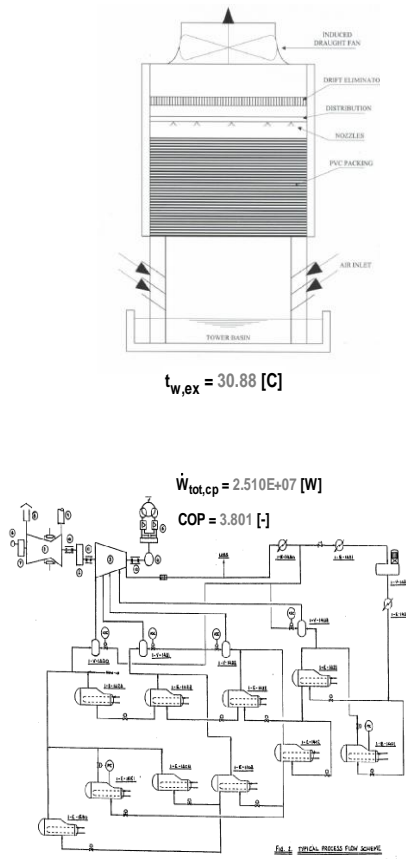
"Calculate compressor work from energy balance on the refrigeration circuit"

$$W_{\text{dot_comp}} = Q_{\text{dot_tot_cd_mix}} - Q_{\text{dot_tot_ev_mix}}$$

$$+ Q_{\text{dot_w_int1}} + Q_{\text{dot_w_int2}} + Q_{\text{dot_w_aft}} \text{ "check energy balance"}$$

"!TOTAL ENERGY CONSUMPTION BY GAS LIQUEFACTION PLANT"

$$\text{energy_plant_total} = W_{\text{dot_tot_cp}} + W_{\text{dot_comp}}$$



Schematic of EES Model **without** Cooling Tower Air Conditioning

Appendix 3: Model Verification Data Sheet

	Actual supply (hot) water temp	Model exhaust (cold) water temp	Actual exhaust (cold) water temp	Model supply air wet-bulb temp	Actual supply air dry-bulb temp	Model exhaust air dry-bulb temp	Model exhaust air wet-bulb temp	Model supply air density	Design supply air density	Model cooling tower duty	Model make-up water flow rate	Actual make-up water flow rate	Actual supply air relative humidity	Actual supply water flow rate	Model supply air humidity ratio	Model exhaust air humidity ratio	Model supply air enthalpy	Model air flow rate
1	temp	temp	temp	temp	temp	temp	temp	kg/m ³	kg/m ³	kJ/h	kg/h	kg/h		kg/h	kg/kg	kg/kg	kJ/kg	m ³ /h
3	4.05E+01	3.01E+01	3.24E+01	2.44E+01	2.78E+01	3.74E+01	3.74E+01	1.13E+00	1.06E+00	1.33E+09	4.43E+05	2.48E+05	7.58E+01	2.94E+07	1.80E-02	4.22E-02	7.37E+01	1.98E+07
4	4.06E+01	3.11E+01	3.29E+01	2.60E+01	2.95E+01	3.77E+01	3.77E+01	1.12E+00	1.06E+00	1.24E+09	4.16E+05	2.29E+05	7.58E+01	2.98E+07	2.00E-02	4.28E-02	8.06E+01	1.96E+07
5	4.06E+01	3.00E+01	3.28E+01	2.42E+01	2.76E+01	3.74E+01	3.74E+01	1.13E+00	1.06E+00	1.34E+09	4.45E+05	1.85E+05	7.58E+01	2.91E+07	1.78E-02	4.21E-02	7.32E+01	1.98E+07
6	3.97E+01	2.91E+01	3.20E+01	2.25E+01	2.58E+01	3.67E+01	3.67E+01	1.14E+00	1.06E+00	1.38E+09	4.51E+05	3.18E+05	7.58E+01	2.96E+07	1.60E-02	4.04E-02	6.66E+01	2.00E+07
7	3.84E+01	2.85E+01	3.10E+01	2.17E+01	2.36E+01	3.57E+01	3.57E+01	1.15E+00	1.06E+00	1.31E+09	4.18E+05	2.32E+05	8.48E+01	3.02E+07	1.57E-02	3.80E-02	6.36E+01	2.01E+07
8	3.90E+01	2.91E+01	3.14E+01	2.30E+01	2.56E+01	3.61E+01	3.61E+01	1.14E+00	1.06E+00	1.27E+09	4.11E+05	2.87E+05	8.10E+01	2.92E+07	1.69E-02	3.91E-02	6.85E+01	1.99E+07
9	3.90E+01	2.89E+01	3.08E+01	2.23E+01	2.61E+01	3.62E+01	3.62E+01	1.14E+00	1.06E+00	1.33E+09	4.38E+05	2.09E+05	7.20E+01	3.01E+07	1.55E-02	3.91E-02	6.56E+01	2.00E+07
10	4.02E+01	3.07E+01	3.27E+01	2.55E+01	2.99E+01	3.74E+01	3.74E+01	1.12E+00	1.06E+00	1.23E+09	4.19E+05	2.47E+05	7.08E+01	2.98E+07	1.90E-02	4.20E-02	7.84E+01	1.96E+07
11	4.04E+01	3.10E+01	3.27E+01	2.61E+01	2.84E+01	3.76E+01	3.76E+01	1.12E+00	1.06E+00	1.20E+09	3.96E+05	1.97E+05	8.40E+01	2.91E+07	2.07E-02	4.24E-02	8.13E+01	1.96E+07
12	3.93E+01	2.95E+01	3.21E+01	2.36E+01	2.61E+01	3.65E+01	3.65E+01	1.14E+00	1.06E+00	1.27E+09	4.13E+05	2.74E+05	8.12E+01	2.96E+07	1.75E-02	3.99E-02	7.08E+01	1.99E+07
13	3.51E+01	2.65E+01	2.86E+01	1.94E+01	2.99E+01	3.28E+01	3.28E+01	1.14E+00	1.06E+00	1.11E+09	4.08E+05	2.16E+05	3.76E+01	2.93E+07	9.96E-03	3.21E-02	5.53E+01	1.99E+07
14	3.90E+01	3.01E+01	3.20E+01	2.51E+01	2.72E+01	3.63E+01	3.63E+01	1.13E+00	1.06E+00	1.13E+09	3.69E+05	2.32E+05	8.46E+01	2.94E+07	1.94E-02	3.96E-02	7.68E+01	1.98E+07
15	3.87E+01	2.82E+01	3.14E+01	2.19E+01	2.38E+01	3.57E+01	3.57E+01	1.15E+00	1.06E+00	1.30E+09	4.13E+05	2.47E+05	8.51E+01	2.84E+07	1.59E-02	3.81E-02	6.42E+01	2.01E+07
16	3.95E+01	3.02E+01	3.24E+01	2.49E+01	2.88E+01	3.68E+01	3.68E+01	1.13E+00	1.06E+00	1.20E+09	4.03E+05	1.11E+05	7.34E+01	2.93E+07	1.85E-02	4.06E-02	7.61E+01	1.97E+07
17	3.90E+01	2.98E+01	3.21E+01	2.44E+01	2.75E+01	3.63E+01	3.63E+01	1.13E+00	1.06E+00	1.18E+09	3.89E+05	1.78E+05	7.81E+01	2.94E+07	1.82E-02	3.94E-02	7.39E+01	1.98E+07
18	3.95E+01	3.04E+01	3.26E+01	2.52E+01	2.97E+01	3.69E+01	3.69E+01	1.12E+00	1.06E+00	1.18E+09	4.04E+05	2.11E+05	7.03E+01	2.98E+07	1.87E-02	4.08E-02	7.73E+01	1.96E+07
19	3.84E+01	2.93E+01	3.14E+01	2.37E+01	2.76E+01	3.58E+01	3.58E+01	1.13E+00	1.06E+00	1.18E+09	3.95E+05	2.34E+05	7.21E+01	2.97E+07	1.69E-02	3.84E-02	7.09E+01	1.98E+07
20	3.95E+01	2.97E+01	3.23E+01	2.40E+01	2.65E+01	3.67E+01	3.67E+01	1.14E+00	1.06E+00	1.27E+09	4.13E+05	2.33E+05	8.15E+01	2.95E+07	1.79E-02	4.03E-02	7.21E+01	1.99E+07
21	4.00E+01	3.08E+01	3.27E+01	2.57E+01	2.93E+01	3.73E+01	3.73E+01	1.12E+00	1.06E+00	1.20E+09	4.04E+05	2.25E+05	7.55E+01	2.97E+07	1.96E-02	4.17E-02	7.93E+01	1.96E+07
22	4.00E+01	3.03E+01	3.27E+01	2.50E+01	2.79E+01	3.72E+01	3.72E+01	1.13E+00	1.06E+00	1.25E+09	4.13E+05	2.23E+05	7.90E+01	2.95E+07	1.89E-02	4.15E-02	7.62E+01	1.97E+07
23	3.84E+01	2.94E+01	3.12E+01	2.44E+01	2.54E+01	3.57E+01	3.57E+01	1.14E+00	1.06E+00	1.11E+09	3.52E+05	2.69E+05	9.20E+01	2.85E+07	1.91E-02	3.81E-02	7.39E+01	1.99E+07
79	3.97E+01	3.09E+01	3.22E+01	2.60E+01	3.19E+01	3.71E+01	3.71E+01	1.11E+00	1.06E+00	1.15E+09	4.05E+05	4.23E+04	6.32E+01	2.99E+07	1.91E-02	4.14E-02	8.07E+01	1.95E+07
170	4.02E+01	2.96E+01	3.25E+01	2.43E+01	2.69E+01	3.69E+01	3.69E+01	1.13E+00	1.06E+00	1.27E+09	4.15E+05	1.69E+05	8.11E+01	2.76E+07	1.83E-02	4.09E-02	7.35E+01	1.98E+07

Appendix 4: Simulation Model Results

- *Results for Model with Air Conditioning*

A=2 [-]
 acc_gravity=9.81 [m/s²]
 AU_cd=3.561E+07 [W/K]
 AU_desup=1.022E+06 [W/K]
 AU_HHP_ev=918080 [W/K]
 AU_HHP_ev_n=918080 [W/K]
 AU_HP_ev=918136 [W/K]
 AU_HP_ev_n=918136 [W/K]
 AU_LP_ev=762420 [W/K]
 AU_LP_ev_n=762420 [W/K]
 AU_MP_ev=816888 [W/K]
 AU_MP_ev_n=816888 [W/K]
 AU_mr_HHP_ev=2.088E+06 [W/K]
 AU_mr_HP_ev=4.389E+06 [W/K]
 AU_mr_LP_ev=2.737E+06 [W/K]
 AU_mr_MP_ev=3.390E+06 [W/K]
 AU_n=1.491E+07 [W/K]
 AU_rb=3.091E+06 [W/K]
 B=761000 [-]
 COP=3.813 [-]
 C_0=9363 [J/kg]
 C_1=1786 [J/kg-K]
 C_2=11.35 [J/kg-K²]
 C_3=0.9886 [J/kg-K³]
 c_p_a=1005 [J/kg-K]
 c_p_as=1025 [J/kg-K]
 c_p_g=2184 [J/kg-K]
 c_p_h=2907 [J/kg-K]
 c_p_mr_HHP_ev=2992 [J/kg-K]
 c_p_mr_HP_ev=3077 [J/kg-K]
 c_p_mr_LP_ev=2963 [J/kg-K]
 c_p_mr_MP_ev=3010 [J/kg-K]
 c_p_ng_HHP_ev=2581 [J/kg-K]
 c_p_ng_HP_ev=2611 [J/kg-K]
 c_p_ng_LP_ev=2856 [J/kg-K]
 c_p_ng_MP_ev=2726 [J/kg-K]
 c_p_s=1820 [J/kg-K]
 c_p_w=4190 [J/kg-K]
 c_w_p=4190 [J/kg-K]
 DELTAt_oh=5 [C]
 DELTAt_sc=8.5 [C]
 DP_a_absorber=200 [Pa]
 eff=0.75 [-]
 eff_motor=0.95 [-]
 eff_pump=0.88 [-]
 energy_dehyd_total=5.717E+07 [W]
 energy_plant_total=1.074E+08 [W]
 epsilon_cd=0.9606 [-]
 epsilon_HHP_ev=0.9183 [-]
 epsilon_HP_ev=0.9159 [-]
 epsilon_LP_ev=0.8474 [-]
 epsilon_MP_ev=0.8788 [-]

epsilon_s_HHP_cp=0.76 [-]
epsilon_s_HP_cp=0.74 [-]
epsilon_s_LP_cp=0.82 [-]
epsilon_s_MP_cp=0.77 [-]
fluid1\$='propane'
fluid2\$='methane'
gamma=0.63 [-]
Head_pump=18 [m]
h_c2h6_ex_ev=-106197 [J/kg]
h_C2H6_ex_HHP_ev=-385289 [J/kg]
h_C2H6_ex_HP_ev=-439652 [J/kg]
h_C2H6_ex_LP_ev=-519084 [J/kg]
h_C2H6_ex_MP_ev=-482890 [J/kg]
h_c2h6_su_ev=-519222 [J/kg]
h_C2H6_su_HHP_ev=-135867 [J/kg]
h_C2H6_su_HP_ev=-385289 [J/kg]
h_C2H6_su_LP_ev=-482890 [J/kg]
h_C2H6_su_MP_ev=-439652 [J/kg]
h_c3h8_ex_ev=117405 [J/kg]
h_C3H8_ex_HHP_ev=231213 [J/kg]
h_C3H8_ex_HP_ev=191589 [J/kg]
h_C3H8_ex_LP_ev=128035 [J/kg]
h_C3H8_ex_MP_ev=157545 [J/kg]
h_c3h8_su_ev=127731 [J/kg]
h_C3H8_su_HHP_ev=292926 [J/kg]
h_C3H8_su_HP_ev=231213 [J/kg]
h_C3H8_su_LP_ev=157545 [J/kg]
h_C3H8_su_MP_ev=191589 [J/kg]
h_ch4_ex_ev=-133319 [J/kg]
h_CH4_ex_HHP_ev=-82903 [J/kg]
h_CH4_ex_HP_ev=-124746 [J/kg]
h_CH4_ex_LP_ev=-199467 [J/kg]
h_CH4_ex_MP_ev=-163326 [J/kg]
h_ch4_su_ev=-193419 [J/kg]
h_CH4_su_HHP_ev=-22699 [J/kg]
h_CH4_su_HP_ev=-82903 [J/kg]
h_CH4_su_LP_ev=-163326 [J/kg]
h_CH4_su_MP_ev=-124746 [J/kg]
h_n2_ex_ev=246598 [J/kg]
h_N2_ex_HHP_ev=284115 [J/kg]
h_N2_ex_HP_ev=265945 [J/kg]
h_N2_ex_LP_ev=234502 [J/kg]
h_N2_ex_MP_ev=249477 [J/kg]
h_n2_su_ev=235596 [J/kg]
h_N2_su_HHP_ev=310431 [J/kg]
h_N2_su_HP_ev=284115 [J/kg]
h_N2_su_LP_ev=249477 [J/kg]
h_N2_su_MP_ev=265945 [J/kg]
h_ng_ex_HHP_ev=-64293 [J/kg]
h_ng_ex_HP_ev=-132107 [J/kg]
h_ng_ex_LP_ev=-203884 [J/kg]
h_ng_ex_MP_ev=-171936 [J/kg]
h_ng_su_HHP_ev=-17603 [J/kg]
h_ng_su_HP_ev=-64293 [J/kg]
h_ng_su_LP_ev=-171936 [J/kg]
h_ng_su_MP_ev=-132107 [J/kg]
h_r_ex_cd=283771 [J/kg]
h_r_ex_desup=614409 [J/kg]

h_r_ex_HHP_cp=653398 [J/kg]
h_r_ex_HHP_cp_s=645316 [J/kg]
h_r_ex_HHP_ev=605058 [J/kg]
h_r_ex_HHP_expv=283771 [J/kg]
h_r_ex_HHP_mr_ev=595185 [J/kg]
h_r_ex_HP_cp=637045 [J/kg]
h_r_ex_HP_cp_s=623571 [J/kg]
h_r_ex_HP_ev=575027 [J/kg]
h_r_ex_HP_expv=283771 [J/kg]
h_r_ex_HP_mr_ev=575027 [J/kg]
h_r_ex_LP_cp=568386 [J/kg]
h_r_ex_LP_cp_s=564265 [J/kg]
h_r_ex_LP_ev=545494 [J/kg]
h_r_ex_LP_expv=283771 [J/kg]
h_r_ex_LP_mr_ev=545494 [J/kg]
h_r_ex_MP_cp=592126 [J/kg]
h_r_ex_MP_cp_s=585486 [J/kg]
h_r_ex_MP_ev=558654 [J/kg]
h_r_ex_MP_expv=283771 [J/kg]
h_r_ex_MP_mr_ev=558654 [J/kg]
h_r_su_cd=614409 [J/kg]
h_r_su_desup=653398 [J/kg]
h_r_su_HHP_cp=619725 [J/kg]
h_r_su_HHP_ev=283771 [J/kg]
h_r_su_HHP_mr_ev=283771 [J/kg]
h_r_su_HP_cp=585224 [J/kg]
h_r_su_HP_ev=283771 [J/kg]
h_r_su_HP_mr_ev=283771 [J/kg]
h_r_su_LP_cp=545494 [J/kg]
h_r_su_LP_ev=283771 [J/kg]
h_r_su_LP_mr_ev=283771 [J/kg]
h_r_su_MP_cp=563255 [J/kg]
h_r_su_MP_ev=283771 [J/kg]
h_r_su_MP_mr_ev=283771 [J/kg]
LNG_prod=136 [kg/s]
MM_C2H6=30.07 [kg/kmol]
MM_C3H8=44.1 [kg/kmol]
MM_CH4=16.04 [kg/kmol]
MM_mix=26.24 [kg/kmol]
MM_N2=28.01 [kg/kmol]
MW_s=18.02 [kg/kmol]
M_dot_a=5345 [kg/s]
M_dot_a_dehyd=1069 [kg/s]
M_dot_a_n=4751 [kg/s]
M_dot_C2H6=112.6 [kg/s]
M_dot_C3H8=33.43 [kg/s]
M_dot_CH4=104.9 [kg/s]
m_dot_g=216.5 [kg/s]
M_dot_g_hr=779228 [kg/h]
m_dot_h=373 [kg/s]
M_dot_mr_ev=274 [kg/s]
M_dot_mr_ev_bis=274 [kg/s]
M_dot_N2=23.02 [kg/s]
M_dot_ng_ev=142 [kg/s]
m_dot_r_HHP_cp=325.9 [kg/s]
m_dot_r_HHP_ev=20.64 [kg/s]
m_dot_r_HHP_mr_ev=119.1 [kg/s]
m_dot_r_HP_cp=186.2 [kg/s]

m_dot_r_HP_ev=33.06 [kg/s]
m_dot_r_HP_mr_ev=42.08 [kg/s]
m_dot_r_LP_cp=52.48 [kg/s]
m_dot_r_LP_ev=17.33 [kg/s]
m_dot_r_LP_mr_ev=35.15 [kg/s]
m_dot_r_MP_cp=111 [kg/s]
m_dot_r_MP_ev=20.57 [kg/s]
m_dot_r_MP_mr_ev=37.96 [kg/s]
M_dot_tot=274 [kg/s]
M_dot_w_aft=477.8 [kg/s]
m_dot_w_cd=2628 [kg/s]
m_dot_w_desup=600 [kg/s]
M_dot_w_int1=691.6 [kg/s]
M_dot_w_int2=445.4 [kg/s]
M_dot_w_n=7778 [kg/s]
m_dot_w_pump=4153 [kg/s]
M_dot_w_su=8306 [kg/s]
M_ratio=9.8 [liter/kg]
NTU_cd=3.234 [-]
NTU_f=3.063 [-]
NTU_HHP_ev=2.505 [-]
NTU_HP_ev=2.476 [-]
NTU_LP_ev=1.88 [-]
NTU_MP_ev=2.111 [-]
power_pump=2.200E+06 [W]
pump_efficiency=0.87 [-]
P_atm=100600 [Pa]
P_cd=1.373E+06 [Pa]
p_ex_ev=300000 [Pa]
P_HHP_ev=821362 [Pa]
P_HHP_mr_ev=641362 [Pa]
P_HP_ev=376599 [Pa]
P_HP_mr_ev=376599 [Pa]
P_LP_ev=154946 [Pa]
P_LP_mr_ev=154946 [Pa]
P_MP_ev=234722 [Pa]
P_MP_mr_ev=234722 [Pa]
p_mr_ev=4.800E+06 [Pa]
P_ng_ex_HHP_ev=5.350E+06 [Pa]
P_ng_ex_HP_ev=5.350E+06 [Pa]
P_ng_ex_LP_ev=5.350E+06 [Pa]
P_ng_ex_MP_ev=5.350E+06 [Pa]
P_ng_su_ev=5.350E+06 [Pa]
P_ng_su_HHP_ev=5.350E+06 [Pa]
P_ng_su_HP_ev=5.350E+06 [Pa]
P_ng_su_LP_ev=5.350E+06 [Pa]
P_ng_su_MP_ev=5.350E+06 [Pa]
p_su_ev=4.465E+06 [Pa]
Q_dot=3.567E+08 [W]
Q_dot_c2h6_ev=4.651E+07 [W]
Q_dot_C2H6_HHP_ev=2.809E+07 [W]
Q_dot_C2H6_HP_ev=6.122E+06 [W]
Q_dot_C2H6_LP_ev=4.076E+06 [W]
Q_dot_C2H6_MP_ev=4.869E+06 [W]
Q_dot_c3h8_ev=-345180 [W]
Q_dot_C3H8_HHP_ev=2.063E+06 [W]
Q_dot_C3H8_HP_ev=1.325E+06 [W]
Q_dot_C3H8_LP_ev=986473 [W]

Q_dot_C3H8_MP_ev=1.138E+06 [W]
Q_dot_cd=1.077E+08 [W]
Q_dot_ch4_ev=6.307E+06 [W]
Q_dot_CH4_HHP_ev=6.318E+06 [W]
Q_dot_CH4_HP_ev=4.391E+06 [W]
Q_dot_CH4_LP_ev=3.793E+06 [W]
Q_dot_CH4_MP_ev=4.049E+06 [W]
Q_dot_cooler=8.556E+06 [W]
Q_dot_desup=1.270E+07 [W]
Q_dot_exchanger=7.044E+07 [W]
Q_dot_HHP_ev=6.630E+06 [W]
Q_dot_HHP_mr_ev=3.707E+07 [W]
Q_dot_HP_ev=9.630E+06 [W]
Q_dot_HP_mr_ev=1.225E+07 [W]
Q_dot_LP_ev=4.537E+06 [W]
Q_dot_LP_mr_ev=9.200E+06 [W]
Q_dot_MP_ev=5.656E+06 [W]
Q_dot_MP_mr_ev=1.043E+07 [W]
Q_dot_n=2.640E+08 [W]
Q_dot_n2_ev=253220 [W]
Q_dot_N2_HHP_ev=605691 [W]
Q_dot_N2_HP_ev=418194 [W]
Q_dot_N2_LP_ev=344668 [W]
Q_dot_N2_MP_ev=379028 [W]
Q_dot_rb=4.832E+07 [W]
Q_dot_tot_cd_mix=6.897E+07 [W]
Q_dot_tot_ev=9.542E+07 [W]
Q_dot_tot_ev_mix=5.273E+07 [W]
Q_dot_w_aft=1.959E+07 [W]
Q_dot_w_int1=2.835E+07 [W]
Q_dot_w_int2=1.826E+07 [W]
Q_flux_rb=24000 [W/m^2]
Q_rb=250000 [J/liter]
R=8314 [J/kmol-K]
rho_a=1.11 [kg/m^3]
rho_a_std=1.225 [kg/m^3]
rho_g=1.12 [kg/liter]
rho_h=681 [kg/m^3]
rho_w=1000 [kg/m^3]
RH_dehyd=0.1005 [-]
RH_mix=0.5278 [-]
RH_undehyd=0.632 [-]
s_r_su_HHP_cp=2437 [J/kg-K]
s_r_su_HP_cp=2450 [J/kg-K]
s_r_su_LP_cp=2454 [J/kg-K]
s_r_su_MP_cp=2450 [J/kg-K]
tube_OD_rb=0.01905 [m]
tube_TK_rb=0.00165 [m]
t_a=31.9 [C]
t_app_cooler=5 [C]
t_a_dehyd=31.9 [C]
t_a_mix=31.9 [C]
t_a_su=31.9 [C]
t_a_undehyd=31.9 [C]
T_cd=40.1 [C]
t_delta_pump=0.01644 [C]
T_ex_ev=-34.5 [C]
t_ex_lean_g=55 [C]

t_ex_lean_g_cooler=36.9 [C]
t_ex_rich_g=184 [C]
t_g_ex_rb=204 [C]
t_g_su_rb=184 [C]
t_HHP_ev=19.3 [C]
t_HHP_mr_ev=10.26 [C]
t_HP_ev=-7.352 [C]
t_HP_mr_ev=-7.352 [C]
t_h_ex_rb=190.4 [C]
t_h_su_rb=235 [C]
t_LP_ev=-32 [C]
t_LP_mr_ev=-32 [C]
t_MP_ev=-21.13 [C]
t_MP_mr_ev=-21.13 [C]
t_mr_ex_HHP_ev=11.5 [C]
t_mr_ex_HP_ev=-4.5 [C]
t_mr_ex_LP_ev=-31.6 [C]
t_mr_ex_MP_ev=-18.8 [C]
t_mr_su_HHP_ev=35 [C]
t_mr_su_HP_ev=11.5 [C]
t_mr_su_LP_ev=-18.8 [C]
t_mr_su_MP_ev=-4.5 [C]
t_ng_ex_HHP_ev=21 [C]
t_ng_ex_HP_ev=-4.5 [C]
t_ng_ex_LP_ev=-29.7 [C]
t_ng_ex_MP_ev=-18.8 [C]
t_ng_su_ev=39 [C]
t_ng_su_HHP_ev=39 [C]
t_ng_su_HP_ev=21 [C]
t_ng_su_LP_ev=-18.8 [C]
t_ng_su_MP_ev=-4.5 [C]
T_r_ex_cd=31.6 [C]
T_r_ex_desup=40.1 [C]
t_r_ex_HHP_ev=24.3 [C]
t_r_ex_HHP_mr_ev=15.26 [C]
t_r_ex_HP_ev=-2.352 [C]
t_r_ex_HP_mr_ev=-2.352 [C]
t_r_ex_LP_ev=-27 [C]
t_r_ex_LP_mr_ev=-27 [C]
t_r_ex_MP_ev=-16.13 [C]
t_r_ex_MP_mr_ev=-16.13 [C]
T_r_su_cd=40.1 [C]
T_r_su_desup=57.61 [C]
T_std=288.2 [K]
T_su_ev=-31.6 [C]
t_su_lean_g=204 [C]
t_su_lean_g_cooler=55 [C]
t_su_rich_g=35 [C]
t_wb_ex=36.78 [C]
t_wb_mix=24.1 [C]
t_wb_su_n=28 [C]
t_w_ex=29.92 [C]
t_w_ex_aft=39.7 [C]
t_w_ex_cd=39.7 [C]
t_w_ex_desup=39.7 [C]
t_w_ex_int1=39.7 [C]
t_w_ex_int2=39.7 [C]
t_w_ex_n=32 [C]

t_w_ref=0 [C]
 t_w_su=39.7 [C]
 t_w_su_aft=29.92 [C]
 t_w_su_cd=29.92 [C]
 t_w_su_desup=29.92 [C]
 t_w_su_int1=29.92 [C]
 t_w_su_int2=29.92 [C]
 t_w_su_n=40.1 [C]
 U=0
 vel_h=2 [m/s]
 wt_g=98.8 [-]
 W_dot_blower=256800 [W]
 W_dot_comp=8.244E+07 [W]
 W_dot_comp_design=8.660E+07 [W]
 W_dot_HHP_cp=1.097E+07 [W]
 W_dot_HP_cp=9.647E+06 [W]
 W_dot_LP_cp=1.201E+06 [W]
 W_dot_MP_cp=3.205E+06 [W]
 W_dot_pump=45719 [W]
 W_dot_tot_cp=2.503E+07 [W]
 W_dot_tot_cp_design=4.160E+07 [W]
 w_s_su=0.02588 [kg/m³]
 X=29.92 [-]
 X_C2H6=0.411 [-]
 X_C3H8=0.122 [-]
 X_CH4=0.383 [-]
 X_dehyd=0.2 [-]
 X_N2=0.084 [-]

- **Results for Model without Air Conditioning**

AU_cd=3.561E+07 [W/K]
 AU_desup=1.022E+06 [W/K]
 AU_HHP_ev=918080 [W/K]
 AU_HHP_ev_n=918080 [W/K]
 AU_HP_ev=918136 [W/K]
 AU_HP_ev_n=918136 [W/K]
 AU_LP_ev=762420 [W/K]
 AU_LP_ev_n=762420 [W/K]
 AU_MP_ev=816888 [W/K]
 AU_MP_ev_n=816888 [W/K]
 AU_mr_HHP_ev=2.088E+06 [W/K]
 AU_mr_HP_ev=4.389E+06 [W/K]
 AU_mr_LP_ev=2.737E+06 [W/K]
 AU_mr_MP_ev=3.390E+06 [W/K]
 AU_n=1.491E+07 [W/K]
 COP=3.801 [-]
 C_0=9363 [J/kg]
 C_1=1786 [J/kg-K]
 C_2=11.35 [J/kg-K²]
 C_3=0.9886 [J/kg-K³]
 c_p_a=1005 [J/kg-K]
 c_p_as=1025 [J/kg-K]
 c_p_g=1820 [J/kg-K]
 c_p_mr_HHP_ev=2992 [J/kg-K]
 c_p_mr_HP_ev=3077 [J/kg-K]
 c_p_mr_LP_ev=2963 [J/kg-K]

c_p_mr_MP_ev=3010 [J/kg-K]
c_p_ng_HHP_ev=2581 [J/kg-K]
c_p_ng_HP_ev=2611 [J/kg-K]
c_p_ng_LP_ev=2856 [J/kg-K]
c_p_ng_MP_ev=2726 [J/kg-K]
c_p_w=4190 [J/kg-K]
c_w_p=4190 [J/kg-K]
DELTA_{t_oh}=5 [C]
DELTA_{t_sc}=8.5 [C]
energy_plant_total=1.075E+08 [W]
epsilon_cd=0.9457 [-]
epsilon_HHP_ev=0.9183 [-]
epsilon_HP_ev=0.9159 [-]
epsilon_LP_ev=0.8474 [-]
epsilon_MP_ev=0.8788 [-]
epsilon_s_HHP_cp=0.76 [-]
epsilon_s_HP_cp=0.74 [-]
epsilon_s_LP_cp=0.82 [-]
epsilon_s_MP_cp=0.77 [-]
fluid1\$='propane'
fluid2\$='methane'
h_c2h6_ex_ev=-106197 [J/kg]
h_C2H6_ex_HHP_ev=-385289 [J/kg]
h_C2H6_ex_HP_ev=-439652 [J/kg]
h_C2H6_ex_LP_ev=-519084 [J/kg]
h_C2H6_ex_MP_ev=-482890 [J/kg]
h_c2h6_su_ev=-519222 [J/kg]
h_C2H6_su_HHP_ev=-135867 [J/kg]
h_C2H6_su_HP_ev=-385289 [J/kg]
h_C2H6_su_LP_ev=-482890 [J/kg]
h_C2H6_su_MP_ev=-439652 [J/kg]
h_c3h8_ex_ev=117405 [J/kg]
h_C3H8_ex_HHP_ev=231213 [J/kg]
h_C3H8_ex_HP_ev=191589 [J/kg]
h_C3H8_ex_LP_ev=128035 [J/kg]
h_C3H8_ex_MP_ev=157545 [J/kg]
h_c3h8_su_ev=127731 [J/kg]
h_C3H8_su_HHP_ev=292926 [J/kg]
h_C3H8_su_HP_ev=231213 [J/kg]
h_C3H8_su_LP_ev=157545 [J/kg]
h_C3H8_su_MP_ev=191589 [J/kg]
h_ch4_ex_ev=-133319 [J/kg]
h_CH4_ex_HHP_ev=-82903 [J/kg]
h_CH4_ex_HP_ev=-124746 [J/kg]
h_CH4_ex_LP_ev=-199467 [J/kg]
h_CH4_ex_MP_ev=-163326 [J/kg]
h_ch4_su_ev=-193419 [J/kg]
h_CH4_su_HHP_ev=-22699 [J/kg]
h_CH4_su_HP_ev=-82903 [J/kg]
h_CH4_su_LP_ev=-163326 [J/kg]
h_CH4_su_MP_ev=-124746 [J/kg]
h_fg0=2.501E+06 [J/kg]
h_n2_ex_ev=246598 [J/kg]
h_N2_ex_HHP_ev=284115 [J/kg]
h_N2_ex_HP_ev=265945 [J/kg]
h_N2_ex_LP_ev=234502 [J/kg]
h_N2_ex_MP_ev=249477 [J/kg]
h_n2_su_ev=235596 [J/kg]

h_N2_su_HHP_ev=310431 [J/kg]
h_N2_su_HP_ev=284115 [J/kg]
h_N2_su_LP_ev=249477 [J/kg]
h_N2_su_MP_ev=265945 [J/kg]
h_ng_ex_HHP_ev=-64293 [J/kg]
h_ng_ex_HP_ev=-132107 [J/kg]
h_ng_ex_LP_ev=-203884 [J/kg]
h_ng_ex_MP_ev=-171936 [J/kg]
h_ng_su_HHP_ev=-17603 [J/kg]
h_ng_su_HP_ev=-64293 [J/kg]
h_ng_su_LP_ev=-171936 [J/kg]
h_ng_su_MP_ev=-132107 [J/kg]
h_r_ex_cd=284064 [J/kg]
h_r_ex_desup=614495 [J/kg]
h_r_ex_HHP_cp=653558 [J/kg]
h_r_ex_HHP_cp_s=645438 [J/kg]
h_r_ex_HHP_ev=605058 [J/kg]
h_r_ex_HHP_expv=284064 [J/kg]
h_r_ex_HHP_mr_ev=595185 [J/kg]
h_r_ex_HP_cp=637045 [J/kg]
h_r_ex_HP_cp_s=623572 [J/kg]
h_r_ex_HP_ev=575027 [J/kg]
h_r_ex_HP_expv=284064 [J/kg]
h_r_ex_HP_mr_ev=575027 [J/kg]
h_r_ex_LP_cp=568386 [J/kg]
h_r_ex_LP_cp_s=564265 [J/kg]
h_r_ex_LP_ev=545494 [J/kg]
h_r_ex_LP_expv=284064 [J/kg]
h_r_ex_LP_mr_ev=545494 [J/kg]
h_r_ex_MP_cp=592127 [J/kg]
h_r_ex_MP_cp_s=585486 [J/kg]
h_r_ex_MP_ev=558654 [J/kg]
h_r_ex_MP_expv=284064 [J/kg]
h_r_ex_MP_mr_ev=558654 [J/kg]
h_r_su_cd=614495 [J/kg]
h_r_su_desup=653558 [J/kg]
h_r_su_HHP_cp=619727 [J/kg]
h_r_su_HHP_ev=284064 [J/kg]
h_r_su_HHP_mr_ev=284064 [J/kg]
h_r_su_HP_cp=585225 [J/kg]
h_r_su_HP_ev=284064 [J/kg]
h_r_su_HP_mr_ev=284064 [J/kg]
h_r_su_LP_cp=545494 [J/kg]
h_r_su_LP_ev=284064 [J/kg]
h_r_su_LP_mr_ev=284064 [J/kg]
h_r_su_MP_cp=563255 [J/kg]
h_r_su_MP_ev=284064 [J/kg]
h_r_su_MP_mr_ev=284064 [J/kg]
LNG_prod=135.5 [kg/s]
MM_C2H6=30.07 [kg/kmol]
MM_C3H8=44.1 [kg/kmol]
MM_CH4=16.04 [kg/kmol]
MM_mix=26.24 [kg/kmol]
MM_N2=28.01 [kg/kmol]
M_dot_a=5345 [kg/s]
M_dot_a_n=4751 [kg/s]
M_dot_C2H6=112.6 [kg/s]
M_dot_C3H8=33.43 [kg/s]

M_dot_CH4=104.9 [kg/s]
M_dot_mr_ev=274 [kg/s]
M_dot_mr_ev_bis=274 [kg/s]
M_dot_N2=23.02 [kg/s]
M_dot_ng_ev=142 [kg/s]
m_dot_r_HHP_cp=326.2 [kg/s]
m_dot_r_HHP_ev=20.65 [kg/s]
m_dot_r_HHP_mr_ev=119.2 [kg/s]
m_dot_r_HP_cp=186.4 [kg/s]
m_dot_r_HP_ev=33.1 [kg/s]
m_dot_r_HP_mr_ev=42.12 [kg/s]
m_dot_r_LP_cp=52.54 [kg/s]
m_dot_r_LP_ev=17.35 [kg/s]
m_dot_r_LP_mr_ev=35.19 [kg/s]
m_dot_r_MP_cp=111.1 [kg/s]
m_dot_r_MP_ev=20.6 [kg/s]
m_dot_r_MP_mr_ev=38 [kg/s]
M_dot_tot=274 [kg/s]
M_dot_w_aft=530.4 [kg/s]
m_dot_w_cd=2917 [kg/s]
m_dot_w_desup=600 [kg/s]
M_dot_w_int1=767.7 [kg/s]
M_dot_w_int2=494.5 [kg/s]
M_dot_w_n=7778 [kg/s]
m_dot_w_pump=4153 [kg/s]
M_dot_w_su=8306 [kg/s]
NTU_cd=2.913 [-]
NTU_f=3.063 [-]
NTU_HHP_ev=2.505 [-]
NTU_HP_ev=2.476 [-]
NTU_LP_ev=1.88 [-]
NTU_MP_ev=2.111 [-]
power_pump=2.200E+06 [W]
pump_efficiency=0.87 [-]
P_atm=100600 [Pa]
P_cd=1.376E+06 [Pa]
p_ex_ev=300000 [Pa]
P_HHP_ev=821362 [Pa]
P_HHP_mr_ev=641362 [Pa]
P_HP_ev=376599 [Pa]
P_HP_mr_ev=376599 [Pa]
P_LP_ev=154946 [Pa]
P_LP_mr_ev=154946 [Pa]
P_MP_ev=234722 [Pa]
P_MP_mr_ev=234722 [Pa]
p_mr_ev=4.800E+06 [Pa]
P_ng_ex_HHP_ev=5.350E+06 [Pa]
P_ng_ex_HP_ev=5.350E+06 [Pa]
P_ng_ex_LP_ev=5.350E+06 [Pa]
P_ng_ex_MP_ev=5.350E+06 [Pa]
P_ng_su_ev=5.350E+06 [Pa]
P_ng_su_HHP_ev=5.350E+06 [Pa]
P_ng_su_HP_ev=5.350E+06 [Pa]
P_ng_su_LP_ev=5.350E+06 [Pa]
P_ng_su_MP_ev=5.350E+06 [Pa]
p_su_ev=4.465E+06 [Pa]
Q_dot=3.217E+08 [W]
Q_dot_c2h6_ev=4.651E+07 [W]

Q_dot_C2H6_HHP_ev=2.809E+07 [W]
 Q_dot_C2H6_HP_ev=6.122E+06 [W]
 Q_dot_C2H6_LP_ev=4.076E+06 [W]
 Q_dot_C2H6_MP_ev=4.869E+06 [W]
 Q_dot_c3h8_ev=-345180 [W]
 Q_dot_C3H8_HHP_ev=2.063E+06 [W]
 Q_dot_C3H8_HP_ev=1.325E+06 [W]
 Q_dot_C3H8_LP_ev=986473 [W]
 Q_dot_C3H8_MP_ev=1.138E+06 [W]
 Q_dot_cd=1.077E+08 [W]
 Q_dot_ch4_ev=6.307E+06 [W]
 Q_dot_CH4_HHP_ev=6.318E+06 [W]
 Q_dot_CH4_HP_ev=4.391E+06 [W]
 Q_dot_CH4_LP_ev=3.793E+06 [W]
 Q_dot_CH4_MP_ev=4.049E+06 [W]
 Q_dot_desup=1.274E+07 [W]
 Q_dot_HHP_ev=6.630E+06 [W]
 Q_dot_HHP_mr_ev=3.707E+07 [W]
 Q_dot_HP_ev=9.630E+06 [W]
 Q_dot_HP_mr_ev=1.225E+07 [W]
 Q_dot_LP_ev=4.537E+06 [W]
 Q_dot_LP_mr_ev=9.200E+06 [W]
 Q_dot_MP_ev=5.656E+06 [W]
 Q_dot_MP_mr_ev=1.043E+07 [W]
 Q_dot_n=2.640E+08 [W]
 Q_dot_n2_ev=253220 [W]
 Q_dot_N2_HHP_ev=605691 [W]
 Q_dot_N2_HP_ev=418194 [W]
 Q_dot_N2_LP_ev=344668 [W]
 Q_dot_N2_MP_ev=379028 [W]
 Q_dot_tot_cd_mix=6.897E+07 [W]
 Q_dot_tot_ev=9.542E+07 [W]
 Q_dot_tot_ev_mix=5.273E+07 [W]
 Q_dot_w_aft=1.960E+07 [W]
 Q_dot_w_int1=2.836E+07 [W]
 Q_dot_w_int2=1.827E+07 [W]
 rho_a=1.11 [kg/m^3]
 rho_w=1000 [kg/m^3]
 RH_su=0.632 [-]
 s_r_su_HHP_cp=2437 [J/kg-K]
 s_r_su_HP_cp=2450 [J/kg-K]
 s_r_su_LP_cp=2454 [J/kg-K]
 s_r_su_MP_cp=2450 [J/kg-K]
 t_a_su=31.9 [C]
 T_cd=40.21 [C]
 t_delta_pump=0.01644 [C]
 T_ex_ev=-34.5 [C]
 t_HHP_ev=19.3 [C]
 t_HHP_mr_ev=10.26 [C]
 t_HP_ev=-7.352 [C]
 t_HP_mr_ev=-7.352 [C]
 t_LP_ev=-32 [C]
 t_LP_mr_ev=-32 [C]
 t_MP_ev=-21.13 [C]
 t_MP_mr_ev=-21.13 [C]
 t_mr_ex_HHP_ev=11.5 [C]
 t_mr_ex_HP_ev=-4.5 [C]
 t_mr_ex_LP_ev=-31.6 [C]

t_mr_ex_MP_ev=-18.8 [C]
t_mr_su_HHP_ev=35 [C]
t_mr_su_HP_ev=11.5 [C]
t_mr_su_LP_ev=-18.8 [C]
t_mr_su_MP_ev=-4.5 [C]
t_ng_ex_HHP_ev=21 [C]
t_ng_ex_HP_ev=-4.5 [C]
t_ng_ex_LP_ev=-29.7 [C]
t_ng_ex_MP_ev=-18.8 [C]
t_ng_su_ev=39 [C]
t_ng_su_HHP_ev=39 [C]
t_ng_su_HP_ev=21 [C]
t_ng_su_LP_ev=-18.8 [C]
t_ng_su_MP_ev=-4.5 [C]
T_r_ex_cd=31.71 [C]
T_r_ex_desup=40.21 [C]
t_r_ex_HHP_ev=24.3 [C]
t_r_ex_HHP_mr_ev=15.26 [C]
t_r_ex_HP_ev=-2.352 [C]
t_r_ex_HP_mr_ev=-2.352 [C]
t_r_ex_LP_ev=-27 [C]
t_r_ex_LP_mr_ev=-27 [C]
t_r_ex_MP_ev=-16.13 [C]
t_r_ex_MP_mr_ev=-16.13 [C]
T_r_su_cd=40.21 [C]
T_r_su_desup=57.74 [C]
T_su_ev=-31.6 [C]
t_wb_ex=37.02 [C]
t_wb_su_n=28 [C]
t_w_ex=30.88 [C]
t_w_ex_aft=39.7 [C]
t_w_ex_cd=39.7 [C]
t_w_ex_desup=39.7 [C]
t_w_ex_int1=39.7 [C]
t_w_ex_int2=39.7 [C]
t_w_ex_n=32 [C]
t_w_ref=0 [C]
t_w_su=39.7 [C]
t_w_su_aft=30.88 [C]
t_w_su_cd=30.88 [C]
t_w_su_desup=30.88 [C]
t_w_su_int1=30.88 [C]
t_w_su_int2=30.88 [C]
t_w_su_n=40.1 [C]
U=0
W_dot_comp=8.247E+07 [W]
W_dot_comp_design=8.660E+07 [W]
W_dot_HHP_cp=1.103E+07 [W]
W_dot_HP_cp=9.657E+06 [W]
W_dot_LP_cp=1.203E+06 [W]
W_dot_MP_cp=3.209E+06 [W]
W_dot_tot_cp=2.510E+07 [W]
W_dot_tot_cp_design=4.160E+07 [W]
X=30.88 [-]
X_C2H6=0.411 [-]
X_C3H8=0.122 [-]
X_CH4=0.383 [-]
X_N2=0.084 [-]

Appendix 5: Air Conditioning Unit Sizing Calculation Sheet**Calculation of Glycol
(TEG) Circulation Rate**

Lean TEG concentration (wt%)	98.8	<i>Maximum TEG purity for a regenerator temperature of 204 °C and atmospheric pressure</i>
Density of TEG (kg/m ³)	1120	
Dehydrated air fraction - x	0.2	<i>Specified</i>
Total mass flow rate of air to cooling tower (kg/s)	5094	<i>Given</i>
Mass flow rate of air through dehydration absorber (kg/s)	1018.8	<i>Given</i>
Density of air (kg/m ³)	1.11	<i>"</i>
Volume flow rate of air through dehydration absorber (m ³ /s)	917.8	<i>Calculated</i>
Air pressure (bara)	1.006	<i>Given</i>
Air temperature - Inlet Dry bulb (°C)	30	<i>"</i>
Air inlet wet bulb temperature (°C)	28	<i>"</i>
Relative Humidity of inlet air (%)	86	<i>"</i>
Relative Humidity of outlet air (%)	10	<i>Specified</i>
Dewpoint of absorber air at inlet conditions (°C)	27.3	<i>Calculated - using online calculator - at 30 °C & 28 °C inlet air dry bulb & wet bulb temperatures respectively and RH of 86%</i>
Moisture content of inlet air saturated at given inlet pressure & temperature - W (kg/m ³)	0.025878	
Dewpoint of absorber inlet air at standard condition (°C)	12.7	<i>Calculated - using online calculator - at 15 °C inlet air temp and RH of 86%</i>
Moisture content of outlet air at given/specified outlet pressure, temperature & relative humidity (kg/m ³)	0.00301	
Dewpoint of absorber air at outlet condition (°C)	-5	<i>Calculated - using online calculator - at 30 °C inlet air temp and RH of 10%</i>
Dewpoint of absorber outlet air at standard condition (°C)	-16.4	<i>Calculated - using online calculator - at 15 °C inlet air temp and RH of 10%</i>
MW air (kg/kmol)	28.97	<i>Given</i>
Universal gas law constant R (J/kmolK)	8314	<i>"</i>
MW water vapour (kg/kmol)	18.02	<i>"</i>
Standard pressure (Pa or N/m ²)	101325	<i>"</i>
Standard temperature (K)	288.15	<i>"</i>
Density of air at standard conditions (kg/m ³)	1.225	
Molar volume of air (ideal gas) at standard T & P (m ³ /kmol)	23.64	
Standard volume flow rate of air (std m ³ /s)	831.5	
Theoretical number of absorber tray N	3	
Tray efficiency (%)	25	

Actual number of absorber tray	12	Used Equation 3.33 - Actual number of tray = theoretical tray / tray efficiency
Mol fraction water in lean TEG - X_o	0.092	Used Equation 3.26
Activity coefficient for water concentration in lean TEG at 30 °C and 98.8 wt% TEG	0.63	From Figure 3.9 (for a TEG concentration of 98.8 wt% and inlet air temperature of 30° C)
Water content of gas in equilibrium with incoming lean TEG - W_o (kg/10 ⁶ std m ³)	1498.5	Used Equation 3.25
Absorber efficiency - E	0.938	Used Equation 3.23
Absorber factor - A	2	
Equilibrium constant for water in a TEG-water system - K	0.021423	From Equation 3.24
Molar flow rate of air - V_n (kmol/h)	126603	
Molar flow rate of TEG entering absorber top tray - L_o (kmol/h)	5424.5	Used $L_o = A.K.V_n$
Molecular weight of lean TEG (kg/kmol)	137.87	Used Equation 3.27
Mass flow rate of lean TEG entering absorber per hour (kg TEG/h)	747863	Used the ideal gas relationship $m = n.MW$
TEG circulation volume rate (liter/h)	667735	
Mass of water absorbed per day (kg/d)	1642837	Water absorbed from the air flowing through the absorber in one day
Mass of water absorbed per hour (kg/h)	68452	
Circulation ratio (liter TEG/ kg H ₂ O absorbed)	9.8	TEG circulation rate per kg of water absorbed

Absorber Design Calculations

Volume flow rate of air (m ³ /s)	917.8	
Air density (kg/m ³)	1.11	
TEG density (kg/m ³)	1120	
Number of theoretical trays	3	
Overall tray efficiency (%)	25	
Number of actual absorber trays - N	12	
Spacing between trays (mm)	610	
Allowable gas velocity for bubble-caps tray (m/s)	1.75	Used Equation 3.28
Allowable gas velocity for structured packing (m/s)	2.85	Used Equation 3.32
Absorber diameter for bubble-caps tray (m)	25.87	Used Equation 3.29
Absorber diameter for structured packing tray (m)	20.26	Used Equation 3.29
Absorber height for bubble-caps tray (m)	7.32	
Height of packing Equivalent to a Theoretical Plate (HETP)	1.5	Reference GPSA Manual
Absorber height for structured	18	

packing (m)

Total height for structured
packing (m) 18.61

*Added spacing to allow for vapour disengagement at top of
absorber and glycol surge volume at bottom of absorber*

**Regenerator Design
Calculation**

Lean TEG (wt%) 98.8

Circulation ratio (liters TEG/kg
H₂O) 9.8

TEG density (kg/liter) 1.12

Rich TEG (wt%) 90.5

Reboiler Design Calculations

TEG inlet temperature (°C) 184

TEG outlet temperature (°C) 204

TEG flow rate (kg/s) 207.7

TEG specific heat capacity
(J/mol K) 327.6

HTF inlet temperature (°C) 235

HTF flow rate (kg/s) 373

HTF specific heat capacity
(kJ/kg K) 2.907

HTF film coefficient of heat
transfer (W/m² K) 1722

HTF thermal conductivity
(W/m K) 0.114

HTF density (kg/m³) 681

HTF velocity (m/s) 1.8

Allowable pressure drop
through tubes (bar) 1.05

Tube outside diameter (mm) 19.05

Tube wall thickness (mm) 1.65

Tube inside diameter (mm) 15.75

Heat flux (kW/m²) 24

Reboiler duty (kJ/liter) 250

Heat transfer rate (kJ/s or kW) 46370.5

HTF outlet temperature (K) 465

Heat transfer surface area
(m²) 1932

LMTD (K) 17

Overall heat transfer
coefficient U (kW/m² K) 1.398

Cross-sectional flow area per
tube (m²) 0.000195

Total flow cross-sectional area
of HTF in reboiler tube side
(m²) 0.304291

Number of tubes required 1562

Length of tube (m) 20.67

Bundle diameter (m) 0.45

Initial selection for design purpose

Still Column Design
Calculations

TEG flow rate (kg/h) 747863

TEG density (kg/m ³)	1120
TEG flow rate (m ³ /h)	667.73
Empirical constant - A	210
Still column diameter (mm)	5427

Lean-Rich Glycol (Interchange) Exchanger Design Calculation

Approach (°C)	20
Rich TEG inlet temperature (°C)	35
Rich TEG outlet temperature (°C)	184
Lean TEG inlet temperature (°C)	204
Lean TEG outlet temperature (°C)	55
TEG specific heat capacity (kJ/kg K)	2.184
TEG flow rate (kg/s)	207.7
Exchanger duty (kW)	67602

Lean Glycol Cooler Design Calculation

Ambient air inlet temperature (°C)	30
Approach (°C)	5
Lean TEG inlet temperature (°C)	55
Lean TEG outlet temperature (°C)	35
Cooler duty (kW)	9074.1

Surge Vessel Design Calculation

Working temperature (°C)	240	
Working pressure (barg)	1	
Design temperature range - min / max (°C)	20 / 320	<i>NB: 320 °C is max bulk temp of Shell Themia B oil used as HTF in Reboiler</i>
Design pressure (barg)	10	
Retention time (mins)	20	
TEG mass flow rate (kg/s)	207.7	
TEG density (kg/m ³)	1120	
TEG volumetric flow rate (m ³ /s)	0.185	
Surge vessel volume (m ³)	222.58	

Air Blower Calculation

Air pressure drop across absorber (Pa)	200
Blower efficiency	0.75
Volume flow rate of air through dehydration absorber (m ³ /s)	917.8
Blower (fan) power (W)	244756.8

Glycol Pump Calculation

Pump head (m)	18
Pump efficiency	0.88
Motor efficiency	0.95
TEG volumetric flow rate (m ³ /s)	0.185
Density of TEG (kg/m ³)	1120
Acceleration due to gravity (m/s ²)	9.81
Pump power (W)	43878.79

Appendix 6: Cooling Tower Characteristic Calculation Sheet

• **Characteristics With Air Conditioning**

Determination of Actual value of L/G (with air conditioning)

	Actual 1
Design L/G	1.63
Design Fan Power	1360
Design Water Flow Rate	28000
Actual Fan Power	1530
Actual Water Flow Rate	29800
Actual L/G	1.67

Determination of Actual value of KaV/L (with air conditioning)

	Actual 1
Hot water temperature T1	40.5
Cold water temperature T2	30.6
Cooling range of tower (total temperature change)	9.9
Wet-bulb temperature of supply air Twb	25.1
Initial (supply) air enthalpy at supply wet-bulb temperature h1 (kJ/kg)	76.7
Specific heat of water c (kJ/kg K)	4.19
Actual L/G	1.67
Enthalpy of exit air h2 from tower	145.89
Temperature of water in first segment of tower height: $T_2 + 0.1 (T_1 - T_2)$	31.6
Enthalpy of saturated boundary film (vapour) at first segment water temperature (kJ/kg)	108.35
Enthalpy of exit air from first segment	83.62
Enthalpy difference (driving potential) in first segment	24.73
Reciprocal of enthalpy difference in first segment: 1/dh	0.0404
Temperature of water in second segment of tower height: $T_2 + 0.4 (T_1 - T_2)$	34.6
Enthalpy of saturated boundary film (vapour) at second segment temperature (kJ/kg)	126.4
Enthalpy of exit air from second segment	104.38
Enthalpy difference (driving potential) in second segment	22.02
Reciprocal of enthalpy difference in second segment: 1/dh	0.0454
Temperature of water in third segment of tower height: $T_1 - 0.4 (T_1 - T_2)$	36.5
Enthalpy of saturated boundary film (vapour) at third segment temperature (kJ/kg)	139.2
Enthalpy of exit air from third segment	118.21
Enthalpy difference (driving potential) in third segment	20.99
Reciprocal of enthalpy difference in third segment: 1/dh	0.0477
Temperature of water in fourth segment of tower height: $T_1 - 0.4 (T_1 - T_2)$	39.5
Enthalpy of saturated boundary film (vapour) at fourth segment temperature (kJ/kg)	161.92
Enthalpy of exit air from fourth segment	138.97
Enthalpy difference (driving potential) in fourth segment	22.95
Reciprocal of enthalpy difference in fourth segment: 1/dh	0.0436
Sum of enthalpy reciprocals	0.1771
Actual KaV/L	1.84

- **Characteristics Without Air Conditioning**

Determination of Actual value of L/G (without air conditioning)	Actual 1
Design L/G	1.63
Design Fan Power	1360
Design Water Flow Rate	28000
Actual Fan Power	1530
Actual Water Flow Rate	29800
Actual L/G	1.67
Determination of Actual value of KaV/L (without air conditioning)	Actual 1
Hot water temperature T1	40.5
Cold water temperature T2	31.7
Temperature change (cooling range)	8.8
Wet-bulb temperature of supply air Twb	27.2
Initial (supply) air enthalpy at supply wet-bulb temperature h1 (kJ/kg)	85.95
Specific heat of water c (kJ/kg K)	4.19
Actual L/G	1.67
Enthalpy of exit air h2 from tower	147.45
Temperature of water in first segment of tower height: $T_2 + 0.1 (T_1 - T_2)$	32.6
Enthalpy of saturated boundary film (vapour) at first segment temperature (kJ/kg)	114.09
Enthalpy of exit air from first segment	92.10
Enthalpy difference (driving potential) in first segment	21.99
Reciprocal of enthalpy difference in first segment: $1/dh$	0.0455
Temperature of water in second segment of tower height: $T_2 + 0.4 (T_1 - T_2)$	35.2
Enthalpy of saturated boundary film (vapour) at second segment temperature (kJ/kg)	130.32
Enthalpy of exit air from second segment	110.55
Enthalpy difference (driving potential) in second segment	19.77
Reciprocal of enthalpy difference in second segment: $1/dh$	0.0506
Temperature of water in third segment of tower height: $T_1 - 0.4 (T_1 - T_2)$	37.0
Enthalpy of saturated boundary film (vapour) at third segment temperature (kJ/kg)	142.77
Enthalpy of exit air from third segment	122.85
Enthalpy difference (driving potential) in third segment	19.92
Reciprocal of enthalpy difference in third segment: $1/dh$	0.0502
Temperature of water in fourth segment of tower height: $T_1 - 0.4 (T_1 - T_2)$	39.6
Enthalpy of saturated boundary film (vapour) at fourth segment temperature (kJ/kg)	162.73
Enthalpy of exit air from fourth segment	141.30
Enthalpy difference (driving potential) in fourth segment	21.43
Reciprocal of enthalpy difference in fourth segment: $1/dh$	0.0467
Sum of enthalpy reciprocals	0.1929
Actual KaV/L	1.78

Appendix 7: Physical Properties of Glycols

Physical Properties	Monoethylene Glycol MEG	Diethylene Glycol DEG	Triethylene Glycol TEG	Tetraethylene Glycol TETRA EG
Formula	$C_2H_6O_2$	$C_4H_{10}O_4$	$C_6H_{14}O_4$	$C_8H_{18}O_5$
CAS Number	107-21-1	111-46-6	112-27-6	112-60-7
Molecular Weight, g/mol	62	106.12	150	194.2
Boiling Point @ 760 mm Hg, °C (°F)	197 (387)	245 (473)	288 (550)	329 (625) Decomposes
Vapor Pressure at 20°C (68°F) mm Hg	0.06	0.002	<0.01	<0.01
Density, (g/cc) @ 20°C (68°F)	1.115	1.118	1.125	1.124
Density, (g/cc) @ 60°C (140°F) 1.096	1.085	1.087	1.093	1.096
Pounds Per Gallon @ 25°C (77°F)	9.26	9.27	9.35	9.37
Freezing Point, °C (°F)	-13.4 (7.9)	-9.0 (16)	-4.3 (24)	-4 (25)
Pour Point, °C (°F)	<-59 (<-75)	-54 (-65)	-58 (-73)	-41 (-42)
Viscosity, cP @ 25°C (68°F)	16.9	35.7	49.0	58.3
Viscosity, cP @ 60°C (140°F)	5.2	7.3	10.3	11.4
Surface Tension, dynes/cm @ 25°C (77°F)	48	44.8	45.5	44.0
Refractive Index @ 20°C (68°F)	1.430	1.447	1.455	1.459
Specific Heat @ 25°C (77°F) Btu/lb/°F	0.58	0.55	0.52	0.52
Flash Point, °C (°F)	116 (241) ⁽²⁾	154 (310) ⁽²⁾	177 (350) ⁽²⁾	202 (395) ⁽²⁾
Dipole Moment in Debyes	2.28	2.69	2.99	3.25
Coefficient of Expansion x 104 (0-60°C)	6.5	6.6	7.2	7.3
Thermal Conductivity, Btu hr-1 ft-1 °F-1 25°C (77°F)	0.1490	0.1175	0.1133	0.1106
Thermal Conductivity, Btu hr-1 ft-1 °F-1 25°C (77°F)	0.1490	0.1175	0.1133	0.1106
Heat of Formation, Btu/lb @ 25°C (77°F)	-93	-131	-171	-212
Heat of Vaporization, Btu/lb @ 25°C (77°F)	369	234.4	179	139

Appendix 8: Derivation of the Cooling Tower Characteristic (Merkel) Equation

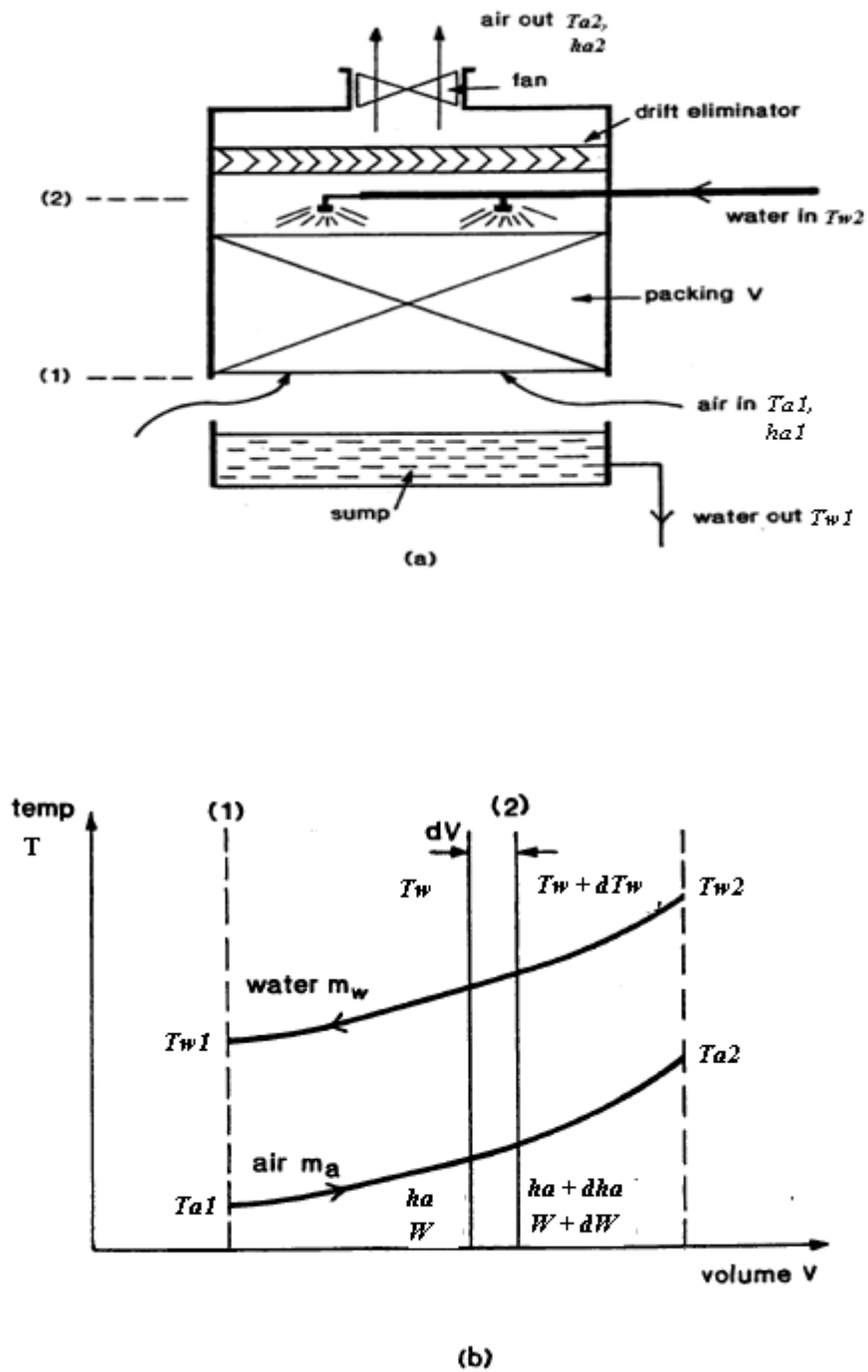


Figure 1

Figure 1(a) shows a typical arrangement of an induced-draught, counterflow cooling tower. Air is circulated upward through the tower by a fan. The hot water is admitted into the upper part of the tower through distribution nozzles and falls downward in counterflow to the air. The packing (or fill) retards the rate of water fall and increases the water surface exposed to the air. Drift eliminators at the top of the tower minimise the drift or carry-over of liquid water in the exit air.

Figure 1(b) depicts schematically the variation in air and water temperatures through a cooling tower of volume V . If a differential volume of the tower packing, dV , is considered, two processes takes place (Figure 2):

- Heat transfer (convection)
- Mass transfer (evaporation)

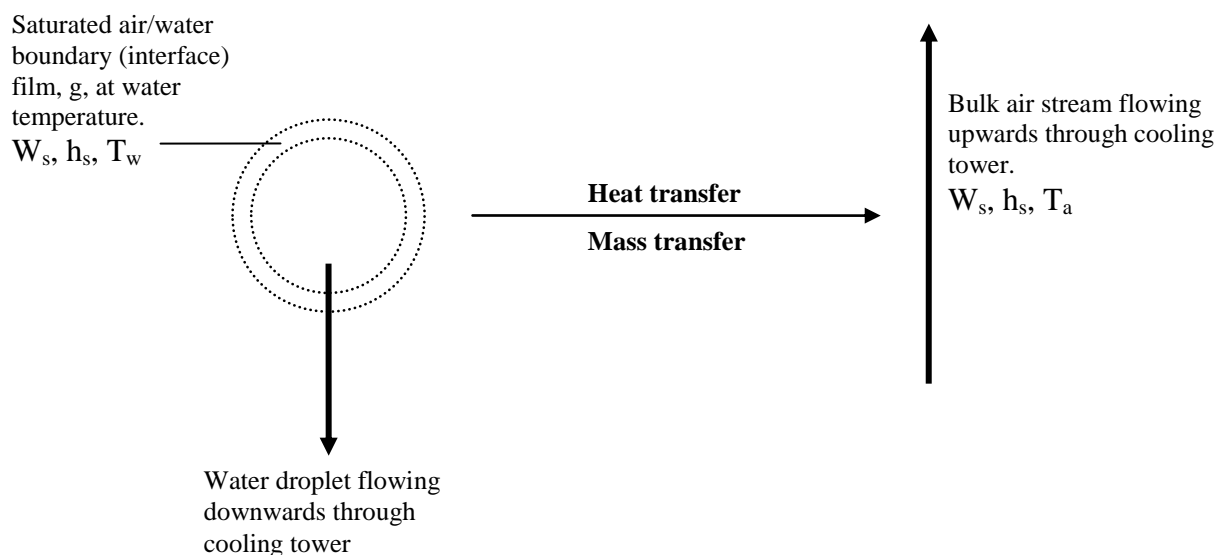


Figure 2

For the differential volume, dV , the mass transfer into the bulk air stream increases the moisture content by an amount dw_a .

$$\dot{m}_a dw_a = h_m A_v dV (w_s - w_a) \quad 1$$

where

h_m = mass transfer coefficient (kg/m² s)

A_v = interfacial (boundary film) surface area of water per unit volume (m²/m³)

w = moisture content (kg/kg dry air)

\dot{m}_a = mass flow rate of air (kg/s)

Note: Subscript s denotes the air-water vapour boundary film (saturated at the water temperature)

The change in the enthalpy of the air can be equated to the energy transfers by convection and evaporation (heat and mass transfer). That is:

$$\dot{m}_a dh_a = h_c A_v dV (T_s - T_a) + h_m A_v dV (w_s - w_a) \bar{h}_g \quad 2$$

where

h_c = convection heat transfer coefficient (W/m² K)

h_g = enthalpy of saturated water vapour (J/kg)

From the heat-mass transfer analogy equation,

$$\frac{h_c}{h_m C p_{as}} = \left(\frac{\alpha}{D} \right)^{2/3} = Le^{2/3} \quad 3$$

where

$C p_{as}$ = humid specific heat (J/kg K)

D = mass diffusivity (m²/s)

α = thermal diffusivity (m²/s)

Le = Lewis number

If the Lewis number is taken to be unity, then:

$$h_c = h_m C p_{as} \quad 4$$

Substituting Equation (3) into Equation (4) gives:

$$\dot{m}_a dh_a = h_m A_v dV \left[p_{as} (T_s - T_a) + (w_s - w_a) \bar{h}_g \right] \quad 5$$

But,

$$C_{p_{as}} (t_s - T_a) = (q_s - h_a) - 2501 (w_s - w_a) \quad 6$$

Substituting Equation (6) into Equation (5) gives:

$$m_a dh_a = h_m A_v dV [(q_s - h_a) - (q_g - 2501 (w_s - w_a))] \quad 7$$

The term $(q_g - 2501 (w_s - w_a))$ is small hence can be neglected. Thus, Equation (7) reduces to:

$$m_a dh_a = h_m A_v dV (q_s - h_a) \quad 8$$

That is, the combined heat and mass transfer process in the cooling tower can be expressed as being driven by an enthalpy difference (enthalpy driving force or potential).

From heat balance, the heat transferred to the air in the cooling tower is equal to the heat transferred from the water. That is:

$$m_a dh_a = m_w C_w dT_w \quad 9$$

where

C_w = specific heat of water (J/kg K)

Hence,

$$\frac{dh_a}{dT_w} = \frac{m_w C_w}{m_a} = \text{constant} \quad 10$$

That is, the air enthalpy varies linearly with the water temperature.

Combining Equation (8) and Equation (9) gives:

$$m_a dh_a = m_w C_w dT_w = h_m A_v dV (q_s - h_a) \quad 11$$

Rearranging Equation (11) gives:

$$\frac{h_m A_v V}{\dot{m}_a} = \int_1^2 \frac{dh_a}{h_s - h_a} \quad 12$$

or,

$$\frac{h_m A_v V}{\dot{m}_w C_w} = \int \frac{dT_w}{h_s - h_a} \quad 13$$

Equation 13 is the Merkel equation or cooling tower characteristic equation.

Appendix 9: Air Conditioning Unit Component Sizing Calculations

Absorber Design

Data:

Density of air = 1.11 kg/ m³

Density of TEG = 1120 kg/ m³

Volume rate of air = 917.8 m³/s

Number of absorber trays = 12 (3 theoretical)

Overall tray efficiency = 25 %

Spacing between trays = 610 mm

Calculation

The allowable gas velocity (for bubble-caps tray) through the absorber is calculated from Equation 3.28. That is,

$$v = 0.055 \left(\frac{1120 - 1.11}{1.11} \right)^{0.5} = 1.75 \text{ m/s}$$

The absorber diameter is calculated from Equation 3.29. That is,

$$d = \sqrt{\frac{(4)(917.8)}{(3.142)(1.75)}} = 25.87 \text{ m}$$

Total height for 12 trays = total number of trays x tray spacing

$$610 \times 12 = 7320 \text{ mm} = 7.32 \text{ m}$$

Alternatively, for a structured packing absorber, the allowable gas velocity is calculated from Equation 3.32 as:

$$v = \frac{3.0}{\sqrt{1.11}} = 2.85 \text{ m/s}$$

And, the absorber diameter (for structured packing) is calculated from Equation 3.29 as

$$d = \sqrt{\frac{(4)(917.8)}{(3.142)(2.85)}} = 20.26 \text{ m}$$

The absorber height for structured packing is calculated using a HETP conversion factor of 1.5 m. That is,

$$\text{Packing height} = 1.5 \times \text{total tray number of tray} = 1.5 \times 12 = 18 \text{ m}$$

$$\text{Absorber height} = \text{packing height} + \text{spacing} = 15 + 0.61 = 18.61 \text{ m}$$

Regenerator Design

The rich TEG concentration (wt % TEG in rich TEG solution) leaving the absorber and entering the regenerator is calculated from Equation 3.18. That is:

$$\text{RichTEG} = \frac{(1.12)(98.8)}{1.12 + (1/9.8)} = 90.5 \text{ wt \%}$$

Reboiler Design

Based on the analytical theory on reboiler in section 3.4.2.1 of Chapter 3, a design value of 250 kJ/liter TEG will be used as the “initial” duty of the reboiler. However, this may be adjusted to suite the pump capacity. Also, a heat transfer fluid (HTF) with a temperature of 235°C shall be used as the heat source for the reboiler. A heat flux of 24 kW/m² is used for the design.

Note:

- A kettle type reboiler shall be used with the HTF on the tube side and TEG on the shell side.
- The Shell Thermal Oil B is used in the NLNG gas plant as HTF (hot oil). This is adopted for this design and, hence the properties will be used in the reboiler design.
- HTF flow rate and inlet temperature would be same as the design value for NLNG plant.
- TEG outlet temperature of 204°C (thermal decomposition temperature limit for TEG) and inlet temperature of 184°C are used for calculations (Campbell, 2004).

Data:

TEG inlet temperature = 184°C

TEG outlet temperature = 204°C

TEG flow rate = 207.7 kg/s = 185.5 liters/s

TEG specific heat capacity = 327.6 J/mol K = 2.184 kJ/kg K

HTF inlet temperature = 235°C (NLNG plant design value)

HTF flow rate = 373 kg/s (NLNG plant design value)

HTF specific heat capacity = 2.902 kJ/kg K

HTF film coefficient of heat transfer = 1722 W/m² K

HTF thermal conductivity = 0.114 W/m K

HTF density = 681 kg/m³

HTF velocity = 2 m/s

Allowable pressure drop through tubes = 1.05 bar

Tube outside diameter = 19.05 mm

Tube wall thickness = 1.65 mm

Tube inside diameter = 19.05 – (2 x 1.65) = 15.75 mm

Heat flux = 24 kW/m²

Heat duty (initial selection) = 250 kJ/liter = (250 x 185.5) kJ/s = 46375 kW

Calculation:

HTF outlet temperature = 235 – [(46375) / (373 x 2.902)] = 192°C

The heat transfer surface area = 46375 / 24 = 1932 m²

Logarithm Mean Temperature Difference is

$$\begin{aligned} LMTD &= \{[(235 - 204) - (192 - 184)] / LN [(235 - 204) / (192 - 184)]\} \\ &= 17^\circ C = 17 K \quad (\text{NB: a correction factor of 1.0 has been assumed}) \end{aligned}$$

$$\begin{aligned} \text{Overall heat transfer coefficient} &= [(46375) / (1932) (17)] \\ &= 1.412 \text{ kW/m}^2 \text{ K} = 1412 \text{ W/m}^2 \text{ K} \end{aligned}$$

The cross-sectional area of one tube = $[3.142 (0.01575)^2] / 4 = 0.000195 \text{ m}^2$

Using the average HTF velocity in the tubes, the HTF flow rate and density, the total flow cross-sectional area of the HTF in the reboiler (tube-side) is:

$$= (373) / (681) (2) = 0.273862 \text{ m}^2$$

Hence, the *number of tubes required* = 0.273862 / 0.000195 = 1405

But, *heat transfer surface area* = 1405 x 3.142 x 0.01905 x length of tube

That is, 1932 = 1405 x 3.142 x 0.01905 x length of tube

Hence, *length of tube* = (1932) / (1405 x 3.142 x 0.01905) = 22.97 m (straight)

Still Column Design

A standard packed tower with a 25 mm slotted ring packing will be used as “initial” selection for design of the still column.

The size (diameter) of the still is calculated using Equation 3.19. That is,

$$\text{Still Column diameter (mm)} = 210 \times (\text{glycol circulation rate in m}^3/\text{h})^{0.5}$$

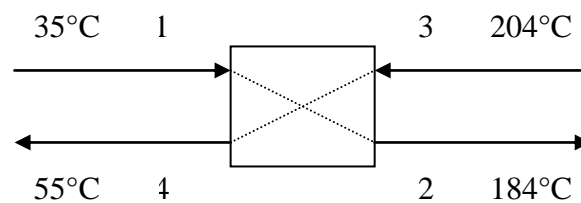
$$\text{Density of TEG} = 1.12 \text{ kg/liter} = 1120 \text{ kg/ m}^3$$

$$\text{Calculated TEG flow rate} = 747863 \text{ kg TEG/h} = 667.73 \text{ m}^3/\text{h}$$

Hence,

$$\text{Still Column diameter} = 210 \times (667.73)^{0.5} = 5425 \text{ mm} = 5.425 \text{ m}$$

Lean-Rich Exchanger Design



Note:

- A plate heat exchanger is used for this service.
- It is assumed that the rich glycol from the absorber (pt 1) enters at a temperature 5°C warmer than the inlet air (to the absorber) due to the reflux condenser duty.
- A 20°C approach is used in the heat exchanger design.
- The average specific heat of the lean glycol is taken to be 2.184 kJ/kg K.
- TEG flow rate = 207.7 kg/s

Applying Equation 3.20, the heat exchanger duty is calculated as:

$$Q = 207.7 \times 2.184 \times (204 - 55) = 67589 \text{ kW}$$

Lean Glycol Cooler

- An Air cooler shall be used for this service.
- The maximum inlet air (ambient) temperature is taken to be 30°C.
- The Lean glycol temperature at inlet to the air cooler is 55°C.
- A 5°C approach shall be used in the cooler design. Hence, Lean glycol outlet temperature from cooler shall be 35°C.

$$\text{Cooler duty, } Q = 207.7 \times 2.184 \times (55 - 35) = 9072 \text{ kW}$$

Filter Design

The following initial design specification will be used for the filters:

1. Two full-flow type filters in parallel with no by-pass.
2. A cloth fabric capable of reducing solids (in the rich TEG) to about 100 ppm by weight will be used as the filter element.
3. The filter element size is 10 μm .
4. Maximum allowable differential pressure ΔP across a filter = 170 Pa

Glycol Surge Vessel Design

The surge vessel shall be fabricated and sized to applicable international codes (ASME Pressure Vessel Code or BS 5500 – Specification for Unfired Fusion Welded Pressure Vessels) to meet the following process conditions:

- Working temperature = 240°C
- Working pressure = 1 barg
- Design temperature range = 20 to 320°C
- Design pressure = 10 barg
- Retention time between low and high levels = 20 minutes = 1200 sec
- TEG mass flow rate = 207.7 kg/s
- TEG density = 1120 kg/ m³
- *TEG volumetric flow rate = 207.7 / 1120 = 0.185 m³/s*
- *Total volume = volumetric flow rate x retention time in seconds*
0.185 x 1200 = 222 m³

Note:

- 320°C is maximum bulk temperature of Shell Thermia B oil that is used as HTF in Reboiler)
- The total volume is calculated based on glycol flow rate of 0.185 m³ and a retention time of 1200 seconds (20 minutes).
- A 20% margin shall be added to the total volume for sufficient hold-up capacity between normal and high level as well as to create allowance to accept glycol drained from the reboiler to allow repair or inspection of the reboiler.

Glycol Circulation Pump Design**Data:**

Pump head = 18 m

Pump efficiency = 0.88

Motor (driver) efficiency = 0.95

TEG volumetric flow rate = 0.185 m³/s

TEG Density = 1120 kg/m³

Acceleration due to gravity = 9.81 m/s²

Calculation:

$$Pump\ power = (1120 \times 9.81 \times 18 \times 0.185) / (0.88 \times 0.95) = 43879\ W$$

Note:

An electric motor driven centrifugal pump shall be used since this (the air conditioning system) is a low pressure service.

Air Blower Design

A fixed speed electric driven blower, sized to provide the specified air flow rate, shall be used. The density ratio of the air, based on which the fan would be selected, is 0.84

NB: The density ratio of the air is the ratio of the actual density to the density at standard conditions.

Data:

Air pressure drop across absorber = 200 Pa (2 mbar)

Blower efficiency = 75%

Volumetric air flow rate = $917.8 \text{ m}^3/\text{s}$

Calculation:

Blower (fan) power = $(917.8 \times 200)/0.75 = 244747 \text{ W}$