STUDY OF COOLING TOWER SYSTEM FOR ENERGY EFFICIENCY OPERATION

By

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Dissertation submitted in partial fulfilment of
the requirements for the
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CERTIFICATION OF APPROVAL

Study of Cooling Tower System for Energy Efficiency Operation

by

Harikrishnan Nagalingam

A project dissertation submitted to the
Chemical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
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Approved by,

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UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
JUNE 2004

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

HARIKRISHNAN NAGALINGAM
Student, UTP.

ABSTRACT

This study shows how by reducing the cooling water circulation rate, improves the efficiency of the operations in the cooling tower system. This is done without modifying the internals of the current cooling towers such as the fills, water treatment programme and air flow rate. This study is applicable for all existing plants with the cooling tower system. This study shows the redistribution of cooling water in the cooling tower water network. This study with detail calculations are tested for chillers, distillation condensers and air conditioners. The cooling tower performance curve is used as the tool for this analysis. The benefits of this study are colder cooling water temperature, reduce the compression work of the refrigeration cycle by 3.0 % for every 0.6 ^{0}C reduction in the cooling water temperature, reduce the cooling tower water consumption, reduce the cooling tower water treatment chemical consumption, reduce the cooling water circulation pump power consumption, and increase the heat transfer of the distillation condensers (increase production).

ACKNOWLEDGEMENT

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CHAPTER 1
INTRODUCTION

1.1 BACKGROUND

Nowadays, engineers are concern about energy saving to reduce the operating cost. It is not the matter of how much energy being used but how much useful work being extracted out from the energy source. The energy problem has a pronounced effect on the day-to-day operation of the plant and as such this warrants a systematic approach. With the major driving force behind the energy saving incentive attributed to a higher fuel price, the need to employ a sound and proven approach for energy saving efforts becomes inevitable. Currently, many techniques for energy saving have been employed such as good housekeeping, energy audits, improving efficiency of each units, thermal pinch analysis and so on. The cooling tower system looking the whole network in a bird's eye view have a huge potential for energy saving and should not be neglected.

1.2 PROBLEM STATEMENT

Cooling tower system consists of mainly cooling tower water network, process heat exchangers, refrigeration equipments, air conditioners and so on. Each of these units (heat exchangers, refrigeration equipments, air conditioners) have a separate loop with the cooling tower water network, thus increasing the consumption of the cooling water circulation rate. Conventionally the cooling tower system is design to meet the process demand and been over design. This study looking at the backward approach (from cooling tower system to the demand) systematically where it looks into optimization of the cooling water circulation rate and temperature. The goal is by improving the performance of the cooling tower system, the outlet temperature of the cooling water from the cooling tower should be colder.

The significant of this study, the colder the water, less energy is required to power the equipment in the refrigeration / air-conditioning system. Burger (1995) is very definite: “A 1.0 ^{0}F (0.6 ^{0}C) colder water returned to the compressors and condensers in air conditioning and refrigeration equipment calculates, by use of Enthalpy Charts, to a 3.0 % savings in electrical energy input to these machines. Therefore, a little more than 3.0 ^{0}F (1.7 ^{0}C) colder water off the tower can save 10.0 % of electrical energy” (p.7). To be energy efficient, the cold water have to be kept colder, thus it increases the driving force for heat transfer from the hot stream. This gives potential to increase the hot stream flow rate (increase production).

There are 2 ways to improve the cooling tower system, internally or externally. Internal performance improvement is generally modifying the cooling tower internal water distribution system (fill), cooling tower water treatment (remove algae formation, fouling) or adjusting the air flow rate. This is generally not favorable to the industrial practitioners because major shut down of the plant is required. This study is new in the industry where looking the cooling tower system externally, the whole cooling tower water network. The only way to improve externally is by reducing the cooling water circulation rate systematically. By using the cooling water reuse and redistribution investigation, opportunities are identified to break the separate loops or redistribute the cooling water and improve the energy efficiency of the operation. When less water flows to the existing cooling tower, the cooling water temperature and the make up water consumption can be reduced.

1.3 OBJECTIVES AND SCOPE OF STUDY

The objective of this study is to improve the efficiency of the cooling tower system externally without the internal modification of the cooling tower unit (cooling tower water treatment, fill or packing and air flow rate). The specific objective of this study is to make the cooling water temperature colder. The target is to reduce the cooling water circulation rate by cooling water reuse and redistribution investigation and using cooling tower performance curve. The success of this study will achieve the primary goal on which reduces the compression power of the refrigeration / air conditioning system and water consumption to bring savings to the company. The reduction of the compression work due to reduction of the cooling water temperature is proved thermodynamically as part of the research study. This project is relevant for all the existing running process plants utilizing the cooling tower system.

CHAPTER 2

LITERATURE REVIEW / THEORY

2.1 COOLING TOWERS

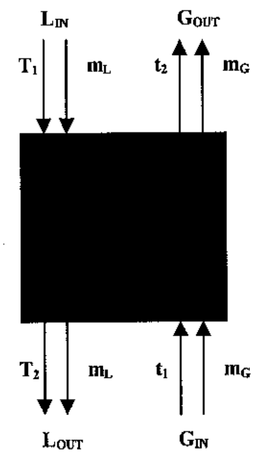
Basic Principles

The basic principal bringing about the cooling in a cooling tower is the mechanism of evaporative cooling and the exchange of the sensible heat. The air-water mixture releases latent heat of vaporization. The water exposed to cooling air streams, evaporates as the water changes to vapor. The heat is taken from the water that remains by lowering its temperature. However, there is a penalty involved, and that is loss of water which goes up the cooling tower and is discharged into the atmosphere as hot moist water vapor. Sensible heat that changes temperature is also responsible for part of the cooling tower's operation. When water is warmer than the air, there is a tendency for the air to cool the water. The air then gets hotter as it gains the sensible heat of the water and the water is cooled as its sensible heat is transferred to the air. Burger (1995) says that cooling due to the evaporative effect of the release of latent heat of vaporization amounts to approximately 75.0 % while 25.0 % of the heat exchange in the cooling tower is sensible heat transfer. There are 2 types of cooling tower; cross flow type and counter flow type.

Thermal Evaluation

The thermal evaluation of the cooling tower is important in the design of the cooling tower. Many theories have been developed describing the heat and mass transfer which takes place inside cooling towers. The cooling tower may be considered as a direct contact heat exchanger, so "there are no acceptable method for accurately calculating the total contact surfaces between the water and the air inside the tower. Therefore, a 'K' factor or heat transfer coefficient cannot be determined directly or calculated by known heat transfer theory" (Burger, 1995). The design data must be obtained by full scale tests under actual operating conditions. The results can then be plotted and formulated.

Experimentation and research utilized in analyzing cooling towers may be described as the 'Black Box Theory'. That is, the boundary conditions are defined and described by fundamental equations and mathematical modeling. The exact process by which the heat and mass transfer takes place within a direct contact heat exchanger is not fully known. However, the results may be formulated into equations which adequately evaluates and predicts performance of the device under most operating conditions. Please refer Figure 1: The Black Box Theory.



L = Liquid (Water)

G = Gas (Air)

T = Temperatures

m_L = Mass of liquid (GPM)

m_G = Mass of gas (CFM)

Figure 2.1: The Black Box Theory

The generally accepted concept of evaluating water cooling performance was developed by Merkel in 1925. His analysis and equations included the sensible and latent heat transfer into an overall heat and mass transfer process based on enthalpy difference as the basic driving force. Each water surface is surrounded by a film of saturated air at the bulk water temperature. The air becomes heated and saturated as it passes through the cooling tower. The temperature and associated physical properties of both water and air change according to their relative position. The heat and mass transfer have been related to the enthalpy difference at each point or each temperature increment. The Merkel Theory established that atmospheric water cooling is basically related to and limited by the wet bulb temperature which is an approximate measure of enthalpy. The wet bulb is determined from the psychometric chart and defined as the intersection of the ambient temperature and dew point or ambient temperature and relative humidity. The wet bulb is an indication of the availability of evaporative potential of the atmosphere. The wet bulb is also referred to as the temperature of adiabatic saturation. Water cannot be cooled to below the wet bulb temperature in a cooling tower.

Based on the development of the experimental models, a cooling tower performance curve was developed for each cooling tower unit (Please refer Appendix 1: The Cross Flow Cooling Tower Performance Curve). The curve is plotted with the thermal demand (KaV/L) as a function of the liquid to gas ratio (L/G) which is useful in this study. Both the thermal demand and liquid to gas ratio are dimensionless values. The larger the thermal demand on the ordinate scale, the more difficult it will be for a cooling tower to meet the design conditions. However, generally cooling towers are designed for worst case scenario (high wet bulb temperature), thus it is over design with high thermal loading. The approach lines (cold water temperature minus wet bulb temperature) and the demand curve which is independent of the tower design and performance of any particular fill are shown on the curve.

2.2 VAPOUR COMPRESSION REFRIGERATION CYCLES

Refrigeration equipments (chillers, air conditioners) are used to maintain a cold region at a temperature below the surrounding temperature. Chiller utilizes vapor-compression refrigeration cycle for its operation (Please refer Appendix 2 for the diagram of the refrigeration cycle). There are many types of refrigerant being used in the industry depending on its applications and types of compressors for example Ammonia (high efficiency refrigerant), Refrigerant 22, Refrigerant 134a and so on. The cycle consists of 4 processes; adiabatic process through expansion valve, isothermal process through evaporator, adiabatic process through compressor and isothermal process through condenser. This cycle deviate from the ideal Carnot Cycle for practical application. The differences are the isenthalpic valve is used for the irreversible expansion instead of turbine because generally turbine in this system has low efficiency and higher maintenance cost, the feed for the compressor should be in saturated vapor phase and not in 2 phase flow to prevent cavitations, the hot temperature slightly higher then the Carnot's hot temperature and the cold temperature is slightly lower than the Carnot's cold temperature. This cycle is important for the evaluation of the reduction of compression work for colder cooling water temperature.

CHAPTER 3

METHODOLOGY / PROJECT WORK

The following methodology has been employed for this study:

- ❖ To identify the current energy issue and the need for this study.
- ➢ To identify the background of the problem.
- ➢ To identify the current energy efficiency improvement programme.
- ➢ To identify the current cooling tower performance improvement programme.
- ➢ To identify the units associated in the cooling tower system such as cooling tower water network, cooling towers, refrigeration equipments, air conditioners, heat exchangers and so on.

- ❖ To study about the cooling tower unit.
- ➢ To identify the design of the cooling tower and the components of the cooling tower unit.
- ➢ To identify the cooling tower water balance accurate equations for calculations. This includes the range, the approach, the drift losses, evaporation losses, blow down rate, total make up water consumption and so on.

- ❖ To study about the thermal evaluation of the cooling tower unit.
- ➢ To identify the available method and modeling to evaluate the performance of the cooling tower unit (the heat and mass transfer in the cooling tower).
- ➢ The practical way to evaluate the performance of the cooling tower is by using the cooling tower performance curve supplied by the cooling tower manufacturer.

- ❖ To study about the vapor compression refrigeration cycle.
- ➢ To study about the basic Carnot's cycle and its deviation from the real vapor compression refrigeration cycle.
- ➢ To study the function of each units in the refrigeration cycles; compressor, condenser, expansion valve and evaporator.
- ➢ To study the properties of refrigerant available in industry.

- ❖ To prove the 3.0 % reduction of compression work in the refrigeration cycle for every 0.6 ^{0}C reduction in the cooling water temperature.
- ➢ To calculate the compressor work input, the heat transfer from the refrigerant to the cooling water in the condenser and the heat transfer from the product to the refrigerant in the evaporator for cooling water at 29.0 ^{0}C.
- ➢ To calculate the compressor work input, the heat transfer from the refrigerant to the cooling water in the condenser and the heat transfer from the product to the refrigerant in the evaporator for cooling water at 28.4 ^{0}C and prove the 3.0 % reduction in compression work.
- ➢ The calculation above is used based on the enthalpy table for Refrigerant 134a and Ammonia available in the reference.

- ❖ To study the cooling tower system with one chiller.
- ➢ To determine the L/G ratio and the thermal demand of the cooling tower for cooling water temperature at 28.3 ^{0}C.
- ➢ To determine the cooling water circulation rate at constant air flow rate for cooling water temperature at 28.3 ^{0}C.
- ➢ To calculate the compressor work input, the heat transfer from the refrigerant to the cooling water in the condenser and the heat transfer from the chilled water to the refrigerant in the evaporator for cooling water at 28.3 ^{0}C.
- ➢ To calculate the power consumption of the cooling tower fan, cooling water circulation pump and chilled water circulation pump.
- ➢ To calculate the evaporation losses, drift losses, blow down rate and the make up water consumption of the cooling tower unit.
- ➢ The whole steps are repeated for every drop of the cooling water temperature by 1.0 ^{0}C. The annual potential power and water savings (RM/yr) are calculated.
- ➢ The calculation above is performed using the Microsoft Excel software and the results are tabulated.

- ❖ To study the cooling tower system with one chiller, one distillation condenser and one air conditioner.
- ➢ To redistribute the cooling water to achieve the target temperature and cooling water circulation rate.

- To study the effect of 2.0 ^{0}C reduction in the cooling water temperature to the chiller.
- To study the effect of 2.0 ^{0}C reduction in the cooling water temperature to the distillation condenser.
- To study the effect of 2.0 ^{0}C reduction in the cooling water temperature to the air conditioner.
- The calculations involve simple material and energy balances.

- To study the cooling tower system with three distillation condensers.
- To redistribute the cooling water to achieve the target temperature and cooling water circulation rate.
- To study the effect of 2.0 ^{0}C reduction in the cooling water temperature to three distillation condenser.
- The calculations involve simple material and energy balances.

- To report conclusions and propose recommendations for improvement of this project.
- To identify all the benefits associated with this study.
- To identify the modifications involved in the plant to achieve the desire results.
- To propose recommendation for the improvement of this project.

CHAPTER 4

CASE STUDIES

In order to understand better about this study, many simple case studies have been made to test the theory. Based on the methodology, the following case studies have been employed:

4.1 CASE 1: VAPOR COMPRESSION REFRIGERATION CYCLE (REFRIGERANT 134a)

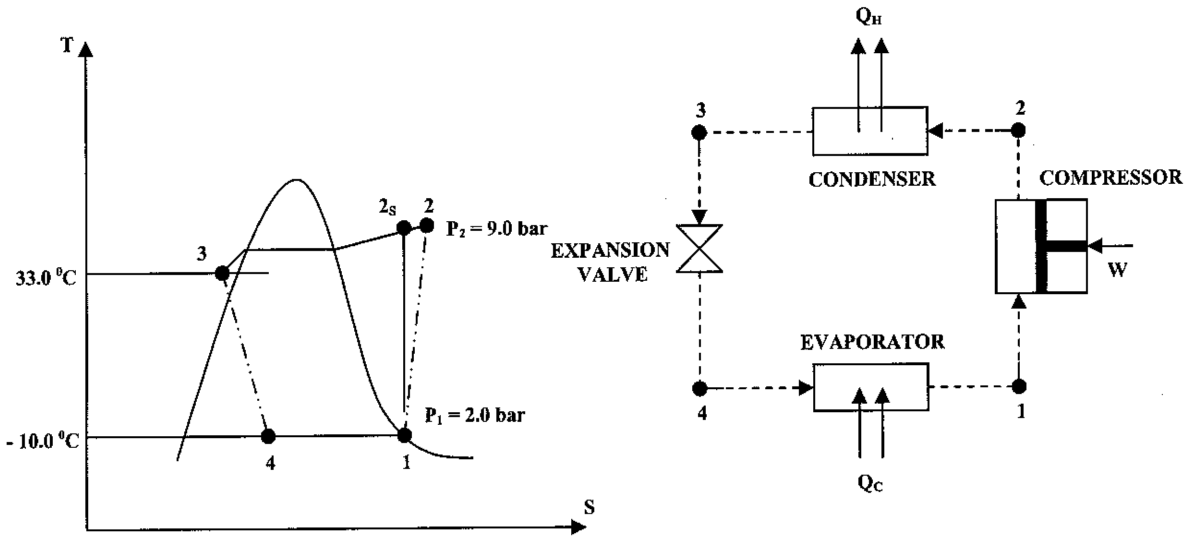


Figure 4.1: Vapor Compression Refrigeration Cycle (R-134a)

Assumptions

There are a few assumptions been made to simplify the calculations. Each component of the cycle is analyzed as a control volume at steady state. There are no pressure drops through the evaporator and condenser. The compressor operates adiabatically with an efficiency of 80.0 %. The expansion through the valve is a throttling process and operates adiabatically. Kinetic and potential energy effects are negligible. Saturated vapor refrigerant at – 10.0 ^{0}C enters the compressor, and liquid refrigerant at 33.0 ^{0}C leaves the condenser. All the processes are internally reversible except expansion valve and compressor.

Note: Please refer Appendix 3 (Case 1.1) for the detail calculations.

Results

Based on the Figure 4.1 and detail calculations in Appendix 3 (Case 1.1), the following results are obtained:

The compression work, **W = 3.104 kW**

The condenser heat load, $\mathbf{Q_H = 14.7}$ **kJ/s**

The evaporator heat load, $\mathbf{Q_C = 11.6}$ **kJ/s**

The cooling water will pass through the condenser of the refrigeration equipment to remove the heat from the refrigerant. Assuming the inlet temperature of the cooling water to the condenser is 29.0 ^{0}C and the outlet temperature of the cooling water from the condenser is 32.0 ^{0}C and the flow is counter current.

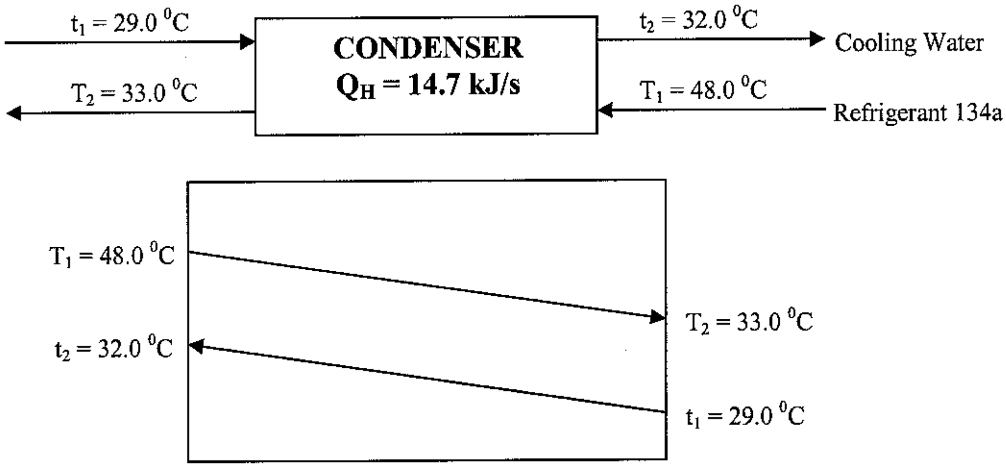


Figure 4.2: Condenser of the Refrigeration Equipment and The Temperature Profile

$\mathbf{\Delta T_{LMTD} = 8.7\ ^0C}$

Based on the reference, Burger (1995), every drop of 0.6 ^{0}C of inlet cooling water temperature, compression work can be saved by 3.0 %. Assuming the cooling water flow rate remains same.

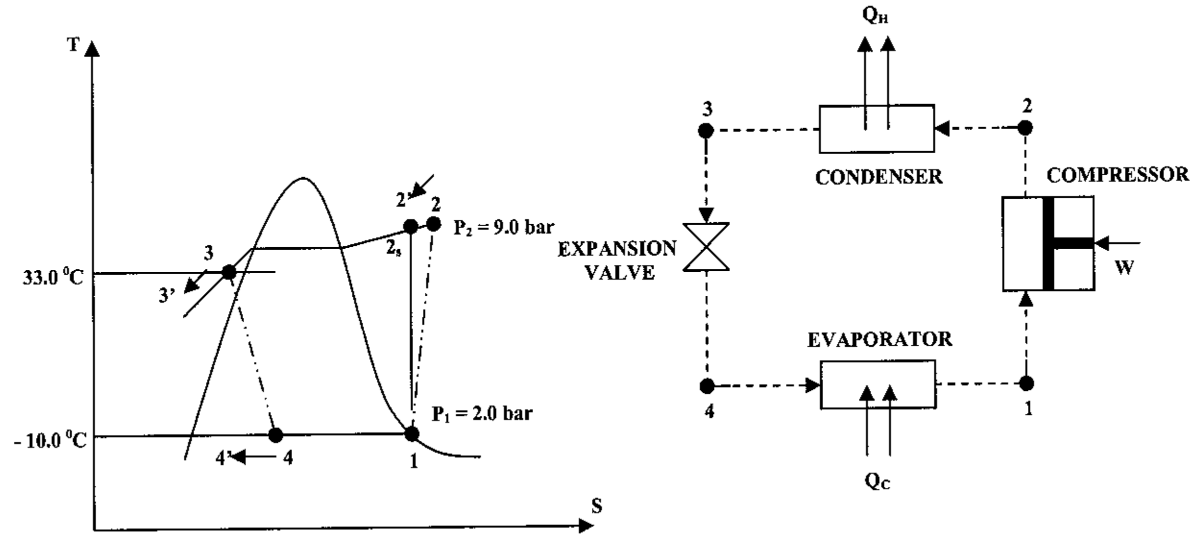


Figure 4.3: Vapor Compression Refrigeration Cycle (R-134a) after 0.6 ^{0}C Reduction in Cooling Water Temperature

After the temperature of the cooling water been reduced by 0.6 ^{0}C, the following parameters are calculated:

The new compression work, **W' = 3.011 kW**

% Reduction in compression work W = **3 %**

The condenser heat load, **Q'_H = Constant** = **14.7 kJ/s**

The evaporator heat load, **Q'_C = 11.7 kJ/s**

% Increase in Q_C = **0.8 %**

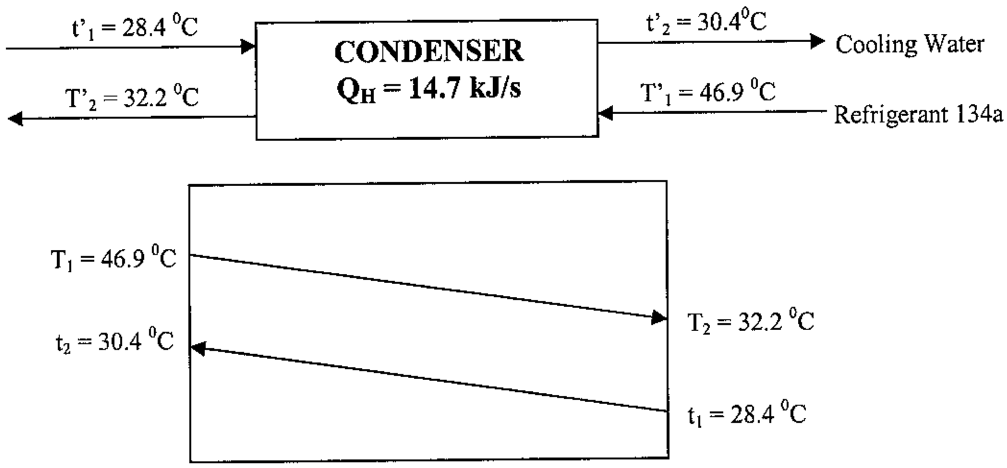


Figure 4.4: Condenser of the Refrigeration Equipment and The Temperature Profile after 0.6 ^{0}C Reduction in Cooling Water Temperature

Q_H = Constant

U = Overall heat transfer coefficient ($kW/m^2.^0C$)

A = Heat transfer area (m^2)

$Q_H = UA\Delta T_{LMTD}$ = Constant

Assume the U, A, Q_H and ΔT_{LMTD} remains same.

The refrigerant and the cooling water will adjust its temperature so that ΔT_{LMTD} remains constant because the restriction that the Q_H constant (no extra heat is removed from the refrigerant).

The log mean temperature difference, $\mathbf{\Delta T'_{LMTD} = 8.7\ ^0C}$ (remains constant)

By using trial and error method, $\mathbf{t'_2 = 30.4\ ^0C}$

% Reduction in t_2 = **5.2 %**

The cooling water return temperature to the cooling tower will be colder.

Adiabatic compression efficiency, η'_c = **82.4 %**

The efficiency of the compressor improves by **2.4 %.**

The same calculation method is done for the refrigeration cycle using Ammonia as the refrigerant and the results produced are in similar pattern. Please refer Appendix 3 (Case 1.2) for the detail calculations.

4.2 CASE 2: COOLING TOWER SYSTEM (ONE CHILLER)

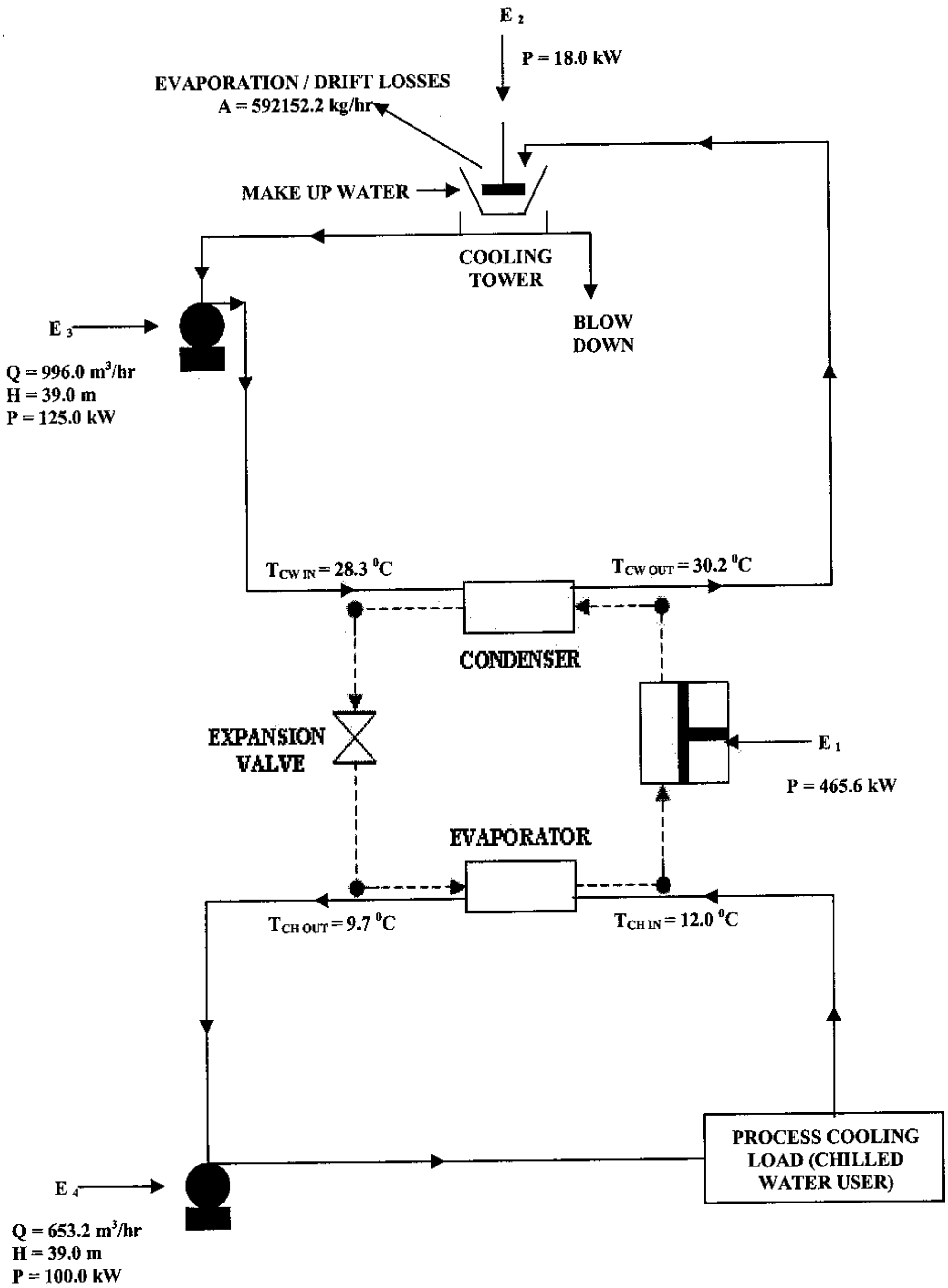


Figure 4.5: Cooling Tower System with One Chiller

Assumptions

There are a few assumptions been made to simplify the calculations. The chiller can be treated as a single unit or a combination of multiple units. The air wet bulb temperature = 21.7 ^{0}C. The air dry bulb temperature = 26.7 ^{0}C. The cooling tower contacting pattern is cross flow. The concentration ratio of the cooling tower is 6. The function of the chiller is to produce chilled water. The chiller using 12.0 kg/s Refrigerant 134a. Each component of the cycle is analyzed as a control volume at steady state. There are no pressure drops through the evaporator and condenser. The compressor operates adiabatically with an efficiency of 80.0 %. The expansion through the valve is a throttling process and operates adiabatically. Kinetic and potential energy effects are negligible. Saturated vapor refrigerant at – 10.0 ^{0}C enters the compressor, and liquid refrigerant at 33.0 ^{0}C leaves the condenser. All the processes are internally reversible except expansion valve and compressor.

Nomenclatures

E_1 = Compressor power consumption = W (kW)

E_2 = Cooling tower fan power consumption (kW)

E_2 = Cooling tower water circulation pump power consumption (kW)

E_4 = Chilled water circulation pump power consumption (kW)

A = Air flow rate (kg/hr)

Q = Water flow rate (m^3/hr)

H = Pump head requirements (m)

P = Power consumption (kW)

Q_H = Heat transfer from the refrigerant to the cooling water in the condenser (kJ/s)

Q_C = Heat transfer from the chilled water to the refrigerant in the evaporator (kJ/s)

h = Enthalpy (kJ/kg)

s = Entropy (kJ/kg.K)

η_c = Efficiency of the compressor

m = Mass flow rate of the refrigerant (kg/s)

Note: Please refer Appendix 4 (Case 2) for the detail calculations.

Results

Using the similar calculation method shown in Section 4.1 for the vapor compression refrigeration cycle and referring to the Figure 4.5 and Appendix 4 (Case 2), the following results are obtained:

$\mathbf{E_1 = 465.6\ kW}$

$\mathbf{E_2 = 18.0\ kW}$

$\mathbf{E_3 = 125.0\ kW}$

$\mathbf{E_4 = 100.0\ kW}$

$\mathbf{Q_H = 2211.6\ kJ/s}$

$\mathbf{Q_C = 1746.0\ kJ/s}$

m = 12.0 kg/s

The above results are based on the cooling tower water temperature **28.3 ^{0}C**.

The cooling tower water circulation rate, **L = 996.0 m^3/hr**

The air flow rate, **G = 592152.2 kg/hr**

The air (wet) density = 1.225 kg/m^3 (Ref: www. hypertextbook.com)

L = 4385.25 GPM

G = 284512.38 CFM

Therefore, the L/G ratio for the cooling tower water system in Figure 4.5, **L/G = 1.8**. Referring to the Appendix 1: Cross Flow Cooling Tower Performance Curve, the cooling tower thermal loading is **1.3**.

The inlet cooling water to the cooling tower temperature = 30.2 ^{0}C

The outlet cooling water from the cooling tower temperature = 28.3 ^{0}C

The air wet bulb temperature = 21.7 ^{0}C

The air dry bulb temperature = 26.7 ^{0}C

The range = 1.9 ^{0}C

The approach = 6.6 ^{0}C

Power consumption

The total electrical power consumption = $E_1 + E_2 + E_3 + E_4$ = **708.6 kW**

Assuming plant operating 333.33 days per year (8000 hr).

Total annual power cost (RM/yr) = **RM 1,247,136.00 / yr**

Water consumption (Ref: NALCO)

The cooling tower water circulation rate, **L = 996.0 m^3/hr**

The cooling tower water concentration ratio, **CR = 6**

The cooling tower water drifts losses, **DR = 149.4 m^3/hr**

The cooling tower water evaporation losses, **E = 359.2 m^3/hr**

The cooling tower water blow down rate, **BR = 71.8 m^3/hr**

The total make up water, MU:

MU = Evaporation + Drift Losses + Blow Down = **580.4 m^3/hr**

Assuming plant operating 333.33 days per year (8000 hr).

Total annual water cost (RM/yr) = **RM 13,930,560 / yr**

Performance Improvement

By improving the performance of the cooling tower, the outlet temperature of the cooling tower water can further be cooled closer to the air wet bulb temperature. Based on the reference, Burger (1995), every drop of 0.6 ^{0}C of inlet cooling water temperature, compression work can be saved by 3.0 %. Therefore every drop of 1.0 ^{0}C temperature, compression work can be saved by 5.0 % (the linear relationship applies for low temperature difference only).

Referring to the Appendix 1: Cross Flow Cooling Tower Performance Curve, in order to reduce the cooling water temperature by 1.0 ^{0}C colder, the L/G ratio should be **1.7**. The thermal loading will be **1.4**. Since the performance of the system is improved without the modification of the cooling tower internals, thus the air flow rate remains constant. The only way to reduce the L/G ratio from 1.8 to 1.7 is by reducing the cooling tower water circulation rate. Thus, the new target for cooling tower water circulation rate:

L' = 915.2 m^3/hr

By this, the cooling water circulation rate will be lesser but colder.

The following parameters are determined after reducing the circulation rate:

E'_1 = 442.3 kW

E'_2 = 18.0 kW (No changes in the air flow rate)

E'_3 = 114.9 kW (Assume Q is proportional to P)

E'_4 = 100.0 kW (No changes in the chilled water flow rate)

Q'_H = 2211.6 kJ/s (Remains fixed)

Q'_C = 1769.3 kJ/s

% Increase in Q_C = **1.3 %**

The chilled water temperature (product) will be colder as the result of the refrigeration effect.

$T_{CH\ OUT}$ = 9.7 ^{0}C

% Reduction in chilled water temperature = **0.3 %**

The cooling water will pass through the condenser with lower mass and lower temperature.

$T_{CW\ OUT}$ = 29.4 ^{0}C

% Reduction in cooling water outlet temperature = **2.8 %**

The inlet cooling water to the cooling tower temperature = 29.4 ^{0}C

The outlet cooling water from the cooling tower temperature = 27.3 ^{0}C

The air wet bulb temperature = 21.7 ^{0}C

The air dry bulb temperature = 26.7 ^{0}C

The range = 2.1 ^{0}C

The approach = 5.6 ^{0}C

Power consumption

The total electrical power consumption = $E_1 + E_2 + E_3 + E_4$ = **675.2 kW**

Assuming plant operating 333.33 days per year (8000 hr).

Total annual energy cost (RM/yr) = **RM 1,188,320.3 / yr**

Total annual power cost saving = **RM 58,815.7 / yr**

Water consumption (Ref: NALCO)

The cooling tower water circulation rate, **L' = 915.2 m^3/hr**

The cooling tower water concentration ratio, **CR = 6**

The cooling tower water drifts losses, **DR = 137.3 m^3/hr**

The cooling tower water evaporation losses, **E = 359.6 m^3/hr**

The cooling tower water blow down rate, **BR = 71.9 m^3/hr**

The total make-up water, MU:

MU = Evaporation + Drift Losses + Blow Down = **568.8 m^3/hr**

Assuming plant operating 333.33 days per year (8000 hr).

Total annual water cost (RM/yr) = **RM 13,651,200 / yr**

Total annual water cost savings = **RM 279,360 / yr**

Total Annual Savings (Water + Power) = **RM 338,175.7 / yr**

The above procedure is repeated for every drop of 1.0 ^{0}C temperature of the cooling tower water until the approach becomes negative (colder than the wet bulb temperature). All the calculation values are tabulated in Table 4.1, 4.2, 4.3, 4.4, 4.5 and 4.6. The total water and energy savings are calculated for each drop of 1.0 ^{0}C of the cooling water temperature.

No.	Cooling Tower Heat Load (kJ/hr)	$T_{CW\ OUT}$ Condenser (^0C)	$T_{CW\ IN}$ Condenser (^0C)	Range (^0C)	Approach (^0C)
1	7948080.0	30.2	**28.3**	1.9	6.6
2	7961760.0	29.4	**27.3**	2.1	5.5
3	7961760.0	28.5	**26.3**	2.2	4.5
4	7961760.0	27.8	**25.3**	2.5	3.5
5	7961760.0	26.9	**24.3**	2.6	2.5
6	7961760.0	26.3	**23.3**	3.0	1.5
7	7961760.0	25.6	**22.3**	3.3	0.5
8	7961760.0	25.0	**21.3**	3.7	-0.4

Table 4.1: The Cooling Tower Heat Load, Temperatures, Range and Approach

No.	Water Flow (m^3/hr)	Air Flow (kg/hr)	L (GPM)	G (CFM)	L/G	Thermal Demand
1	996.0	592152.2	4385.3	284512.4	1.8	1.3
2	915.2	592152.2	4029.6	284512.4	1.7	1.4
3	848.7	592152.2	3736.5	284512.4	1.5	1.4
4	765.5	592152.2	3370.2	284512.4	1.4	1.5
5	721.1	592152.2	3174.8	284512.4	1.3	1.6
6	637.9	592152.2	2808.5	284512.4	1.2	1.7
7	582.4	592152.2	2564.3	284512.4	1.1	1.8
8	510.3	592152.2	2246.8	284512.4	0.9	1.9

Table 4.2: The Cooling Tower L/G Ratio and The Thermal Demand

No.	Q_H (kJ/s)	Q_C (kJ/s)	Chilled Water Flow (m^3/hr)	$T_{CH\ IN}$ (^0C)	$T_{CH\ OUT}$ (^0C)
1	2211.6	1746.0	653.2	12.0	9.7
2	2211.6	1769.3	653.2	12.0	9.7
3	2211.6	1792.6	653.2	12.0	9.7
4	2211.6	1815.8	653.2	12.0	9.6
5	2211.6	1839.1	653.2	12.0	9.6
6	2211.6	1862.4	653.2	12.0	9.6
7	2211.6	1885.7	653.2	12.0	9.5
8	2211.6	1909.0	653.2	12.0	9.5

Table 4.3: The Chiller Loads and The Chilled Water Temperatures

No.	E_1 Compressor (kW)	E_2 CT Fan (kW)	E_3 CT Pump (kW)	E_4 CW Pump (kW)	Total Electrical Power (kW)	Power Savings (RM/yr)
1	465.6	18.0	125.0	100.0	708.6	0.0
2	442.3	18.0	114.9	100.0	675.2	58,816.7
3	419.0	18.0	106.5	100.0	643.6	114,491.7
4	395.8	18.0	96.1	100.0	609.8	173,842.4
5	372.5	18.0	90.5	100.0	581.0	224,616.7
6	349.2	18.0	80.1	100.0	547.3	283,967.3
7	325.9	18.0	73.1	100.0	517.0	337,192.0
8	302.6	18.0	64.0	100.0	484.7	394,092.2

Table 4.4: The Power Consumption of The Cooling Tower System

No.	Blow Down (m^3/hr)	Drift Losses (m^3/hr)	Evaporation Losses (m^3/hr)	Make Up Water (m^3/hr)	Water Savings (RM/yr)
1	71.8	149.4	359.2	580.4	0.0
2	72.0	137.3	359.8	569.1	273,017.1
3	72.0	127.3	359.8	559.1	512,637.3
4	72.0	114.8	359.8	546.6	812,162.6
5	72.0	108.2	359.8	539.9	971,909.4
6	72.0	95.7	359.8	527.5	1,271,434.6
7	72.0	87.4	359.8	519.1	1,471,118.1
8	72.0	76.6	359.8	508.3	1,730,706.6

Table 4.5: The Water Consumption of The Cooling Tower System

No.	Total Annual Power and Water Savings (RM/yr)
1	0.0
2	331,833.8
3	627,129.0
4	986,004.9
5	1,196,526.0
6	1,555,401.9
7	1,808,310.1
8	2,124,798.9

Table 4.6: The Total Annual Power and Water Savings

4.3 CASE 3: COOLING TOWER SYSTEM (ONE CHILLER, ONE DISTILLATION CONDENSER & ONE AIR CONDITIONER)

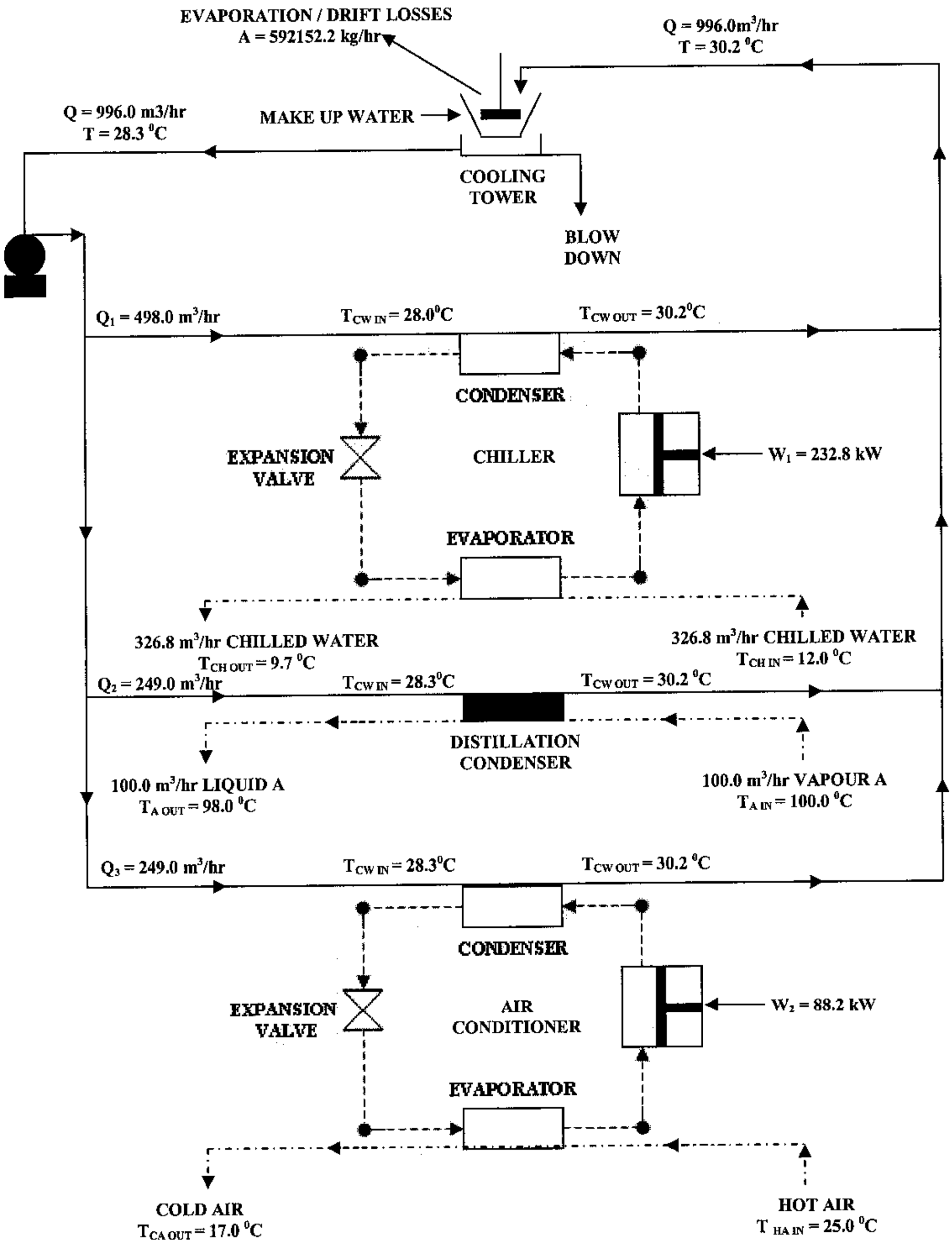


Figure 4.6: Cooling Tower System with One Chiller, One Distillation Condenser and One Air Conditioner

Assumptions

All assumptions and calculations are similar to Section 4.1 and Section 4.2. The same model of the cross flow cooling tower and its performance curve (Appendix 1) are use to determine the thermal demand and the L/G ratio. The chiller using 6.0 kg/s Refrigerant 134a and the air conditioner using 2.9 kg/s Refrigerant 22. Based on the Table 4.2: The Cooling Tower L/G ratio and The Thermal Demand, the target temperature of the cooling water is 26.3 ^{0}C from 28.3 ^{0}C previously (set by the engineer) by improving the performance of the cooling tower system. The new cooling water circulation rate is 848.7 m^3/hr to achieve 2.0 ^{0}C reduction in the temperature. Based on this new temperature, the refrigeration compression work can be saved by 10.0 %. The product of the chiller (chilled water) and the product of the air conditioner (air) will be colder. The cooling water have been redistributed throughout the entire cooling tower water network systematically by trial and error to achieve the new total cooling water circulation rate 848.7 m^3/hr. The new water circulation rate and the new temperature parameters with improved performance (work reduction) are shown in the Figure 4.8.

Note: Please refer Appendix 5 (Case 3) for the detail calculations.

Chiller

Q_H remains fixed = 1105.8 kJ/s

Q_C increases from 873.0 kJ/s to 896.3 kJ/s.

The compression work reduces from 232.8 kW to 209.5 kW (10.0 %).

The cooling water flow been modified from 498.0 m^3/hr to 424.3 m^3/hr.

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) reduces from 30.2 ^{0}C to 28.1 ^{0}C.

The chilled water temperature reduces from 9.7 ^{0}C to 9.6 ^{0}C.

Condenser (For Distillate Product A)

The cooling water flow remains fixed to 249.0 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.3 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.2 ^{0}C.

The heat load, **Q = 1987020.0 kJ/hr**

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

Q is proportional to volumetric flow rate of A.

Initially the flow rate of product A, v_A = **100.0 m^3/hr.**

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.2 ^{0}C.

Q' = 4078620 kJ/hr

v'_A = 205.3 m^3/hr

This creates opportunity to increase production due to higher driving force of the temperature.

% Increase in production = **105.3 %** (maximum)

% Increase in the heat transfer area = **105.3 %** (maximum)

Air Conditioner

Q_H remain fixed = 551.7 kJ/s

Q_C increases from 463.5 kJ/s to 472.3 kJ/s.

The compression work reduces from 88.2 kW to 79.4 kW (10.0 %).

The cooling water flow been modified from 249.0 m^3/hr to 175.3 m^3/hr.

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) reduces from 30.2 ^{0}C to 28.6 ^{0}C.

The air temperature reduces from 17.0 ^{0}C to 16.9 ^{0}C.

Assume the reference temperature = T_{ref} = 0.0 ^{0}C

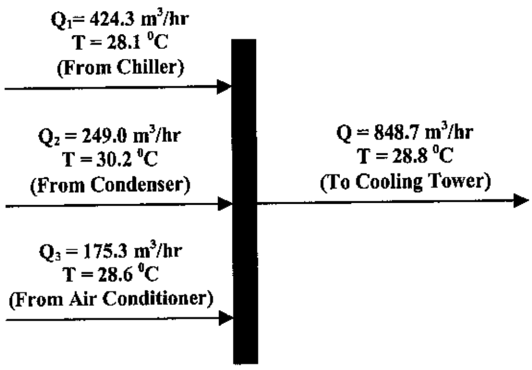


Figure 4.7: Cooling Tower Water Network Mass Balance

The cooling water return temperature to the cooling tower, **T = 28.8 ^{0}C**

This shows the temperature of the whole system overall reduces.

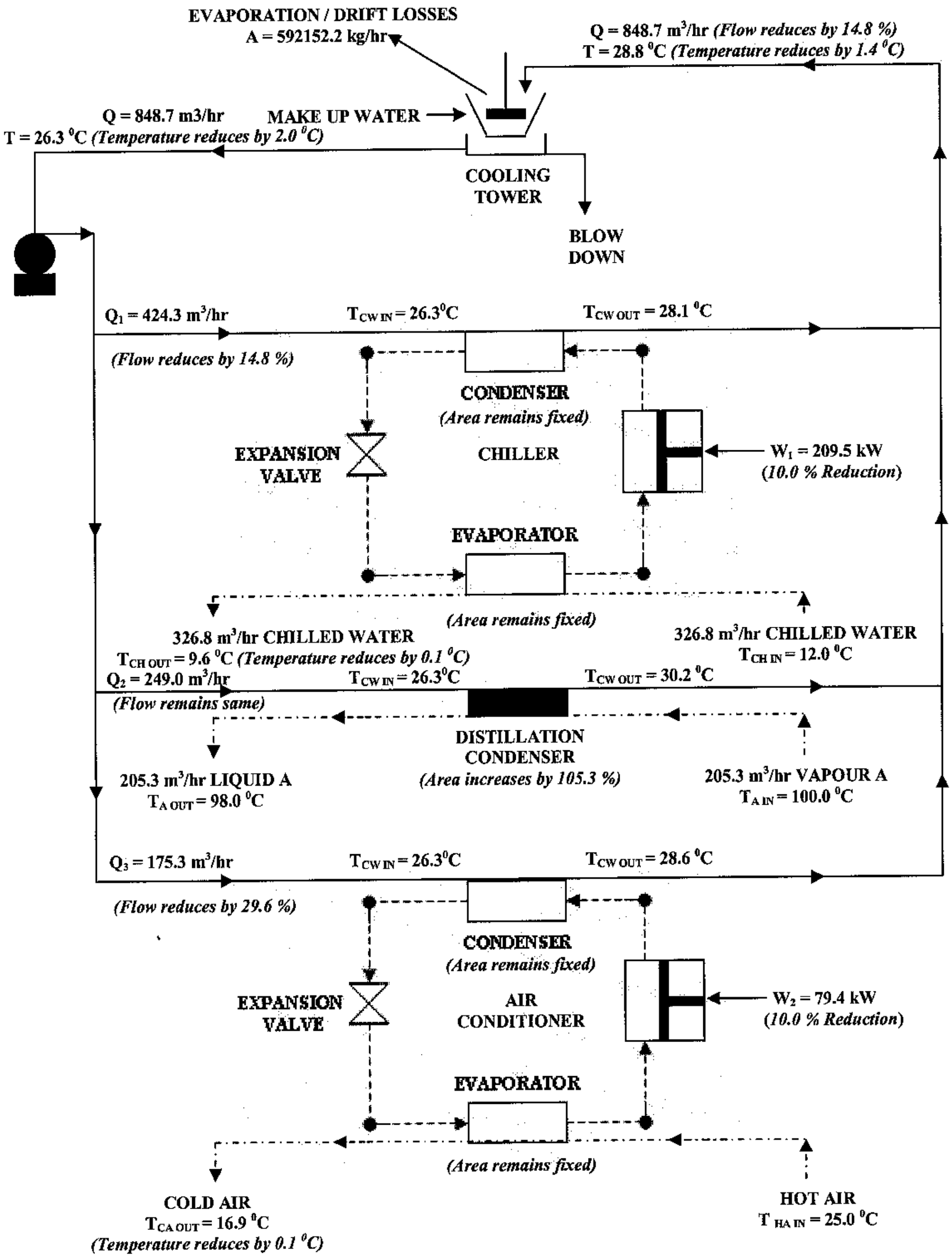


Figure 4.8: Cooling Tower System with One Chiller, One Distillation Condenser and One Air Conditioner after Redistribution of Cooling Water to Achieve 2.0 ^{0}C Reduction in Cooling Water Temperature

4.4 CASE 4: COOLING TOWER SYSTEM (THREE CONDENSERS)

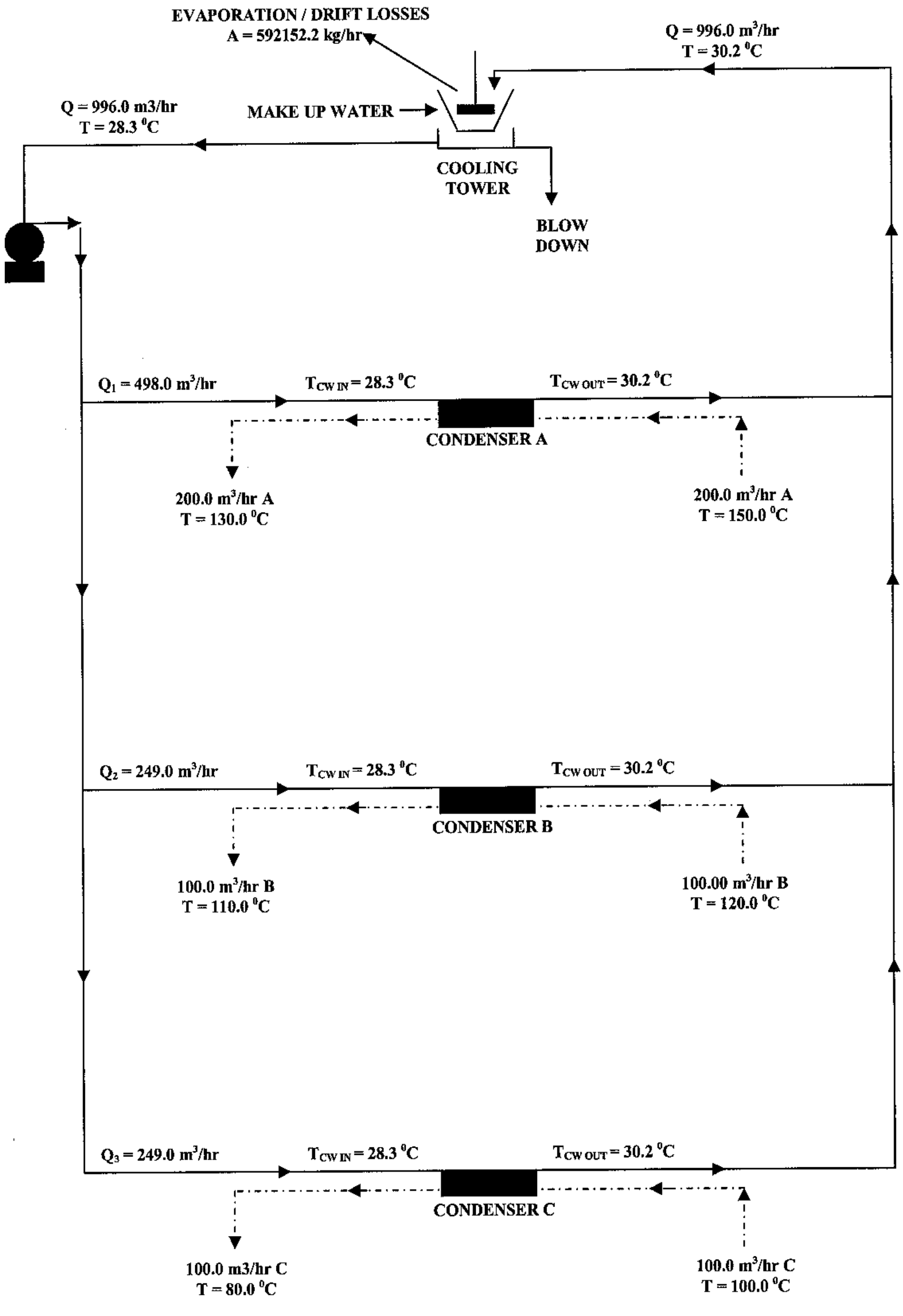


Figure 4.9: Cooling Tower System with Three Condensers

Assumptions

All assumptions and calculations are similar to Section 4.1, Section 4.2 and Section 4.3. The same model of the cross flow cooling tower and its performance curve (Appendix 1) been used to determine the thermal demand and the L/G ratio. Based on the Table 4.2: The Cooling Tower L/G ratio and The Thermal Demand, the target temperature of the cooling water is 26.3 ^{0}C from 28.3 ^{0}C previously (set by the engineer) by improving the performance of the cooling tower system. The new cooling water circulation rate is 848.7 m^3/hr to achieve 2.0 ^{0}C reduction in the temperature. The cooling water have been redistributed throughout the entire cooling tower water network systematically by trial and error to achieve the new total cooling water circulation rate 848.7 m^3/hr. Lower cooling water temperature increases higher driving force creating more room for the product to be condensed in the condenser. The condenser cooling water outlet temperature remains fixed at 30.2 ^{0}C. The new water circulation rate and the new temperature parameters with improved performance are shown in the Figure 4.10.

Note: Please refer Appendix 6 (Case 4) for the detail calculations.

Condenser A (For Distillate Product A)

The cooling water flow been modified from 498.0 m^3/hr to 448.9 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.3 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.2 ^{0}C.

The heat load , **Q = 3974040.0 kJ/hr**

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

Q is proportional to volumetric flow rate of A.

Initially the flow rate of product A, **v_A = 200.0 m^3/hr.**

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.2 ^{0}C.

Q' = 7352654.4 kJ/hr

v'_A = 370.0 m^3/hr (This creates opportunity to increase production due to higher driving force of the temperature)

% Increase in production = **85.0 %** (maximum)

% Increase in the heat transfer area = **85.0 %** (maximum)

Condenser B (For Distillate Product B)

The cooling water flow been modified from 249.0 m^3/hr to 199.9 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.3 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.2 ^{0}C.

The heat load, **Q = 1987020.0 kJ/hr**

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

Q is proportional to volumetric flow rate of B.

Initially the flow rate of product B, **v_B = 100.0 m^3/hr**.

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.2 ^{0}C.

Q' = 3274034.40 kJ/hr

v'_B = 164.77 m^3/hr (This creates opportunity to increase production due to higher driving force of the temperature)

% Increase in production = **64.8 %** (maximum)

% Increase in the heat transfer area = **64.8 %** (maximum)

Condenser C (For Distillate Product C)

The cooling water flow been modified from 249.0 m^3/hr to 199.9 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.3 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.2 ^{0}C.

The heat load, **Q = 1987020.0 kJ/hr**

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

Q is proportional to volumetric flow rate of C.

Initially the flow rate of product C, **v_C = 100.0 m^3/hr**.

The cooling water inlet temperature (condenser) reduces from 28.3 ^{0}C to 26.3 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.2 ^{0}C.

Q' = 3274034.40 kJ/hr

v'_C = 164.77 m^3/hr (This creates opportunity to increase production due to higher driving force of the temperature)

% Increase in production = **64.78 %** (maximum)

% Increase in the heat transfer area = **64.8 %** (maximum)

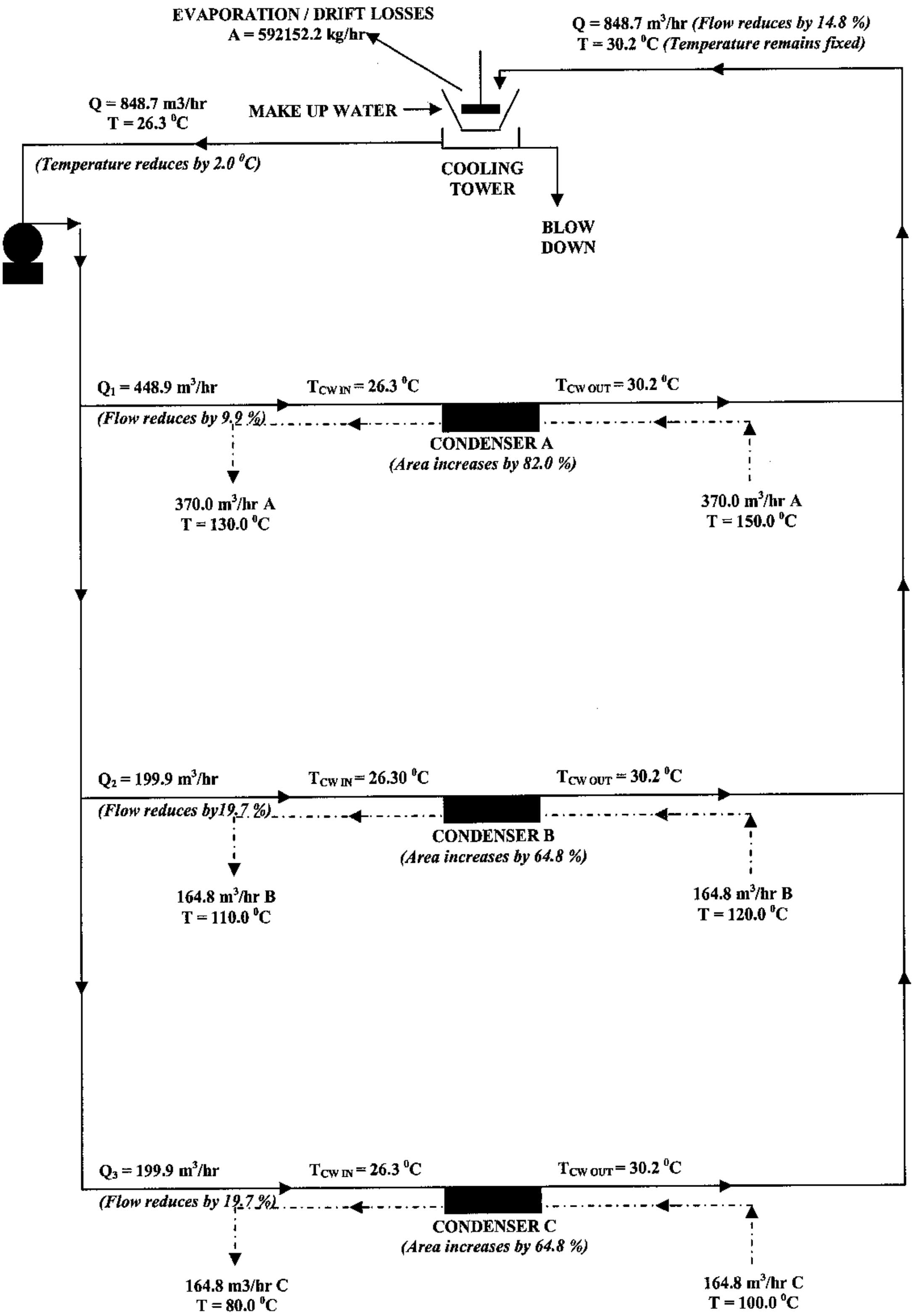


Figure 4.10: Cooling Tower System with Three Condensers after Redistribution of Cooling Water to Achieve 2.0 ^{0}C Reduction in Cooling Water Temperature

CHAPTER 5

DISCUSSION

5.1 CASE 1: VAPOUR COMPRESSION REFRIGERATION CYCLE

The results in the Section 4.1 clearly proves that the reduction of 0.6 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (29.0 ^{0}C to 28.4 ^{0}C); will reduce the compression work by 3.0 % (3.101 kW to 3.011 kW). It also improves the compressor efficiency by 2.4 % (80.0 % to 82.4 %). The heat extracted out from the product (distillate), Q_C, also increases by 0.8 % (11.6 kJ/s to 11.7 kJ/s). The heat removed to the cooling water in the condenser, Q_H, remains the same.

When the inlet cooling water temperature to the condenser is colder, it creates driving force for the refrigerant to further cooled in the high pressure sub cooled phase region in the condenser (temperature drops from 33.0 ^{0}C to 32.2 ^{0}C). Thus, its enthalpy drops. When the high pressure sub cooled liquid passes through an isenthalpic expansion valve, the liquid losses its pressure and 2 phase flow (liquid-vapor) formed. Due to the design of the isenthalpic valve (energy loss is negligible); the enthalpy of the 2 phase flow is the same as the high pressure sub cooled liquid as before. Due to low enthalpy, the 2 phase flow quality, x_4, reduces from 0.2881 to 0.2824. Thus, more liquid is present in the two phase flow. As the 2 phase flow refrigerant passes through the evaporator, it requires more latent heat to completely vaporize to be fully saturated vapor before entering the compressor. Thus, it needs to extract more heat from the evaporator (from the distillate stream). Now, the refrigerant has more energy as previously. The function of the compressor is to bring up the temperature of the refrigerant to a higher elevation by supplying compression work. Since the refrigerant have more energy now, it makes the compressor easier to bring to higher elevation (reduces the compression work). This is governed by the basic heat pump equation:

$$\mathbf{Q_H = Q_C + W}$$

When Q_C increases, W reduces at constant Q_H.

Due to lower temperature, the cooling water can actually remove more heat from the refrigerant in the condenser (increasing Q_H). However, the heat transfer area and heat discharged by the cooling tower to the atmosphere to be increased. This means that more heat is wasted to the atmosphere and the return back cooling water temperature to the cooling tower increases. It increases the heat load of the cooling tower (inefficiency) and may damage the packings. The condenser is only dealing with cooling water and refrigerant (utility streams), so there are no economic value to improve the heat transfer. Thus, the Q_H is restricted to be constant. The temperature of the refrigerant still drops because the system (both the streams) adjusts its temperature so that ΔT_{LMTD} remains constant. As the result of this, the outlet temperature of the cooling water from the condenser drops by 5.2 % (from 32.0 ^{0}C to 30.4 ^{0}C) on which is favorable for the cooling tower. Since the goal of this study is for energy saving, the Q_H value is restricted to be constant, thus no modification is done to the condenser heat transfer area.

Same principle is applied to the evaporator. There are no modifications done for the evaporator heat transfer area. The heat load Q_C is allowed to increase by increasing the log mean temperature difference at constant heat transfer area. The system will adjust itself to achieve the equilibrium. The product stream (distillate) and the refrigerant will become colder.

This refrigeration effect is applicable for all types of refrigerant (Ammonia, R-22 and so on). Please refer to Appendix 3 (Case 1.2) for the calculation of the Vapor Compression Refrigeration Cycle using Ammonia as the refrigerant to prove the same effect to other refrigerants. The relationship between cooling water temperature reduction by 0.6 ^{0}C and refrigeration compression work reduction by 3.0 % can assume to be linear. Thus, reduction in cooling water temperature by 2.0 ^{0}C can reduce the compression work by 10.0 %. However, this assumption is only valid for small temperature changes (0.0 ^{0}C– 5.0 ^{0}C) because in real life the profile is nonlinear due to refrigerant real physical properties. Since, this study is about improving the current cooling tower system, the study only deals with small temperature changes.

5.2 CASE 2: COOLING TOWER SYSTEM (ONE CHILLER)

Section 4.2 shows the effect to the cooling tower system with one chiller for each drop in 1.0 ^{0}C of the cooling water temperature starting from 28.3 ^{0}C until the ambient wet bulb temperature (21.7 ^{0}C). The chiller can be treated as a single unit or a combination of multiple units. This section provides the methodology for an engineer to calculate the performance of the entire cooling tower system systematically with the aid of the given cooling tower performance curve by the cooling tower manufacturer.

Based on the Table 4.1 and Table 4.2 (Section 4.2), when the cooling water circulation rate been reduced and it enters the cooling tower with the constant air flow rate, the temperature of the cooling tower will be colder. This is because the fill in the cooling tower remains unmodified (excess contacting surface area) and because the air is in excess, more room for the air to absorb heat from the cooling water. The tendency for the heat to transfer from the cooling water to the air is more due to higher driving force.

The current way to know how much water need to be reduced is by looking at the cooling tower performance curve. This curve is very specific for that particular cooling tower and tested by the manufacturer. If the temperature of the cooling tower needs to be colder by 1.0 ^{0}C, the associated L/G ratio and the thermal demand of the cooling tower are determined. Thus, the L/G ratio reduces and the thermal demand increases for every 1.0 ^{0}C drop of the cooling water temperature. Since, the air flow rate is constant, the circulation rate can be determined using the ratio. However, take note that the cooling water cannot be cooled below the ambient wet bulb temperature and it serves as the limit.

The higher the thermal demand, the difficult for the cooling tower to achieve the target temperature. However, in this case, cooling towers are usually design for worst case scenario where at high wet bulb temperature and high thermal demand on which it rarely happens. Thus, there is a room of opportunity for the cooling tower performance improvement limited by the range of thermal demand for that particular cooling tower given by the manufacturer and the ambient wet bulb temperature.

Based on Table 4.5 (Section 4.2), the drift losses are the function of the circulation rate and it reduces when the circulation rate reduces. The evaporation losses is a function of the circulation rate and the cooling tower range, when the range increases due to lower cooling water temperature and the circulation rate decreases, the evaporation losses is approximately the same. The blow down rate is the function of the evaporation losses and the concentration ratio. Since the evaporation losses and the concentration ratio remain same, the blow down rate is approximately the same. Overall, the make up water consumption reduces when the temperature of the cooling water reduces and savings can be calculated annually.

When the flow velocity reduces (constant area), it also reduces the heat transfer coefficient for the heat exchangers because it reduces the Reynolds number and Nusselt number. However, all the heat exchangers are over design to take care of uncertainty of low heat transfer coefficient initially during design stage. Thus, excess area is given to all the heat exchangers. This opportunity is used for the water redistribution.

Based on Table 4.4 (Section 4.2), the compressor power consumption reduces by 5.0 % for every 1.0 ^{0}C drop in the cooling water temperature due to the refrigeration effect. The condenser heat load (Q_H) remains constant. Since the air flow rate remains constant, the cooling tower fan power consumption remains the same. The chilled water circulation remains constant, thus the pump power consumption remains the same.

When the circulation rate reduces, at constant pipe cross sectional area, the velocity reduces. Thus, it reduces the pressure drop across the pipe (reduces the pump head requirements) and reduces the pump flow requirements, thus it reduces the circulation pump power consumption. Overall, the power consumption reduces when the temperature of the cooling water reduces and savings can be calculated annually.

Based on Table 4.3 (Section 4.2), the heat load of the evaporator (Q_C) of the chiller increases as the temperature of the cooling water decreases due to the refrigeration effect. Thus, the chilled water will be colder.

Based on Table 4.1, when the circulation flow rate reduces, the temperature return to the cooling tower reduces also and it is favorable to the cooling tower. High TDS problem due to higher temperature return back to cooling tower will not occur in this situation and have been taken into consideration in this study. Although the mass is lesser, the temperature does not increase because the cooling water is colder now. The range increases, but the mass reduces, maintaining the same heat load.

The cooling tower water treatment chemicals can be reduced because the circulation rate (mass) reduces and because the temperature of the overall cooling tower water network reduces, the hardness ions will not precipitate out (scaling).

Supposing the performance curve of that particular cooling tower is not available, it have been found that an average of 10.0 % reduction of cooling water circulation rate is required to achieve reduction in temperature by 1.0 ^{0}C. Thus, the cooling tower can be tested for each 10 % reduction of circulation rate and its performace can be plotted in a graph for analysis. The optimum flow can be achieved by trial and error.

5.3 CASE 3: COOLING TOWER SYSTEM (ONE CHILLER, ONE DISTILLATION CONDENSER & ONE AIR CONDITIONER)

Figure 4.6 (Section 4.3) indicates the performance of a simple cooling tower system with one chiller, one distillation condenser and one air conditioner. The initial cooling water temperature is 28.3 ^{0}C. Assuming the same cross flow cooling tower model been used, based on the Table 4.2 (Section 4.2), an engineer have identified the target temperature for the cooling water to be 26.3 ^{0}C on which it is higher than the ambient wet bulb temperature and the thermal demand of the cooling tower still in the allowable range given by the cooling tower manufacturer. The engineer have identified that the water circulation in the cooling tower water network need to be redistributed in order to achieve new target circulation flow rate (996.0 m^3/hr to 848.7 m^3/hr). By using the trial and error method, the cooling water is redistributed in the entire cooling water network as shown in Figure 4.8.

For the chiller, the calculations in the Section 4.3 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will reduce the compression work by 10.0 % (232.8 kW to 209.5 kW). The heat extracted out from the chilled water, Q_C, also increases from 873.0 kJ/s to 896.3 kJ/s. The heat removed to the cooling water in the condenser, Q_H, remains the same (1105.8 kJ/s). The temperature of the chilled water drops from 9.7 ^{0}C to 9.6 ^{0}C due to the refrigeration effect. The cooling water outlet temperature (condenser) reduces from 30.2 ^{0}C to 28.1 ^{0}C.

For the distillation condenser, the calculations in the Section 4.3 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will increase the heat load of the condenser from 1987020.0 kJ/hr to 4078620.0 kJ/hr. The production can be increase by 105.3 % (100.0 m^3/hr to 205.3 m^3/hr) where it is the maximum production that can be achieved for a particular cooling water temperature. This is because the colder cooling water creates driving force where more room opportunity for heat transfer in the condenser. However, the condenser needs debottlenecking to increase the heat transfer area by 105.3 % to achieve the desire result. The outlet temperature of the condenser remains fixed at 30.2 ^{0}C and the cooling water flow rate remains fixed at 249.0 m^3/hr. However, both this parameters can be modified according to the preference of the engineer.

For the air conditioner, the calculations in the Section 4.3 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will reduce the compression work by 10.0 % (88.2 kW to 79.4 kW). The heat extracted out from the hot air, Q_C, also increases from 463.5 kJ/s to 472.3 kJ/s. The heat removed to the cooling water in the condenser, Q_H, remains the same (551.7 kJ/s). The temperature of the cold air drops from 17.0 ^{0}C to 16.9 ^{0}C due to the refrigeration effect. The cooling water outlet temperature (condenser) reduces from 30.2 ^{0}C to 28.6 ^{0}C.

The chiller and the air conditioner can be treated as a single unit or a combination of the multiple units as a whole provided using the same refrigerant. Assuming the chiller is the combination of multiple chillers, the new cooling water flow rate that been set for the chiller requirements have to be redistributed internally among all the chillers so that the total output flow from the chiller units as a whole remains the same.

Referring to the Figure 4.7 (Section 4.3), the cooling water return to cooling tower flow rate and temperature to the cooling tower been calculated to prove that the target circulation rate been achieved and the temperature is lower than before, after redistribution of the cooling water in the cooling tower water network.

5.4 CASE 4: COOLING TOWER SYSTEM (THREE CONDENSERS)

Figure 4.9 (Section 4.4) indicates the performance of a simple cooling tower system with three distillation condensers (Product A, B and C). The initial cooling water temperature is 28.3 ^{0}C. Assuming the same cross flow cooling tower model been used, based on the Table 4.2 (Section 4.2), an engineer have identified the target temperature for the cooling water to be 26.3 ^{0}C on which it is higher than the ambient wet bulb temperature and the thermal demand of the cooling tower still in the allowable range given by the cooling tower manufacturer. The engineer have identified that the water circulation in the cooling tower water network need to be redistributed in order to achieve new target circulation flow rate (996.0 m^3/hr to 848.7 m^3/hr). By using the trial and error method, the cooling water is redistributed in the entire cooling water network as shown in Figure 4.10.

For the distillation condenser A, the calculations in the Section 4.4 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will increase the heat load of the condenser from 3974040.0 kJ/hr to 7352654.4 kJ/hr. The production can be increase by 85.0 % (200.0 m^3/hr to 370.0 m^3/hr) where it is the maximum production that can be achieved for a particular cooling water temperature. This is because the colder cooling water creates driving force where more room opportunity for heat transfer in the condenser. However, the condenser needs debottlenecking to increase the heat transfer area by 85.0 % to achieve the desire result. The outlet temperature of the condenser remains fixed at 30.2 ^{0}C and the cooling water flow rate has been reduced from 498.0 m^3/hr to 448.9 m^3/hr. However, both this parameters can be modified according to the preference of the engineer.

For the distillation condenser B, the calculations in the Section 4.4 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will increase the heat load of the condenser from 1987020.0 kJ/hr to 3274034.4 kJ/hr. The production can be increase by 64.8 % (100.0 m^3/hr to 164.8 m^3/hr) where it is the maximum production that can be achieved for a particular cooling water temperature. This is because the colder cooling water creates driving force where more room opportunity for heat transfer in the condenser. However, the condenser needs debottlenecking to increase the heat transfer area by 64.8 % to achieve the desire result. The outlet temperature of the condenser remains fixed at 30.2 ^{0}C and the cooling water flow rate has been reduced from 249.0 m^3/hr to 199.9 m^3/hr. However, both this parameters can be modified according to the preference of the engineer.

For the distillation condenser C, the calculations in the Section 4.4 clearly proves that the reduction of 2.0 ^{0}C in the cooling water temperature to the condenser of the refrigeration cycle (28.3 ^{0}C to 26.3 ^{0}C); will increase the heat load of the condenser from 1987020.0 kJ/hr to 3274034.4 kJ/hr. The production can be increase by 64.8 % (100.0 m^3/hr to 164.8 m^3/hr) where it is the maximum production that can be achieved for a particular cooling water temperature. This is because the colder cooling water creates driving force where more room opportunity for heat transfer in the condenser. However, the condenser needs debottlenecking to increase the heat trasnfer area by 64.8 % to achieve the desire result. The outlet temperature of the condenser remains fixed at 30.2 ^{0}C and the cooling water flow rate has been reduced from 249.0 m^3/hr to 199.9 m^3/hr. However, both this parameters can be modified according to the preference of the engineer.

The loops in the cooling tower water network can be broken to reuse the cooling water if the cooling water has sufficient driving force. However, it depends on the practical constraits of each plants. The cooling water reuse investigation need to be assessed systematically with practical consideration and subject to the specific plants only. The control system in the cooling tower system may need to be modified too.

Usually the make up water is higher temperature than the cooling tower water temperature and it is directly mixed in the cooling tower sump. This defeats the purpose for the performance improvement. The cooling tower make up water stream is suggested to be connected with the cooling tower return stream from plant and not directly to the cooling tower sump.

CHAPTER 6
CONCLUSION

By systematic redistribution of the cooling water in the cooling tower water network to reduce the cooling water circulation flow rate, the energy efficiency operation of the cooling tower system can be improved.

The benefits are as follow:

1. Colder cooling water temperature.
2. Reduce the compression work of the refrigeration cycle by 3.0 % for every 0.6 ^{0}C reduction in the cooling water temperature.
3. Reduce the cooling tower water consumption.
4. Reduce the cooling tower water treatment chemicals consumption.
5. Reduce the cooling water circulation pump power consumption.
6. Increase the heat transfer of the distillation condensers (increase production).

Modifications:

1. Debottlenecking of the heat exchangers; distillation condensers (investment to increase the heat transfer area).
2. Modification in the control system.
3. Cooling tower make up water stream to be connected with the cooling tower return stream from the plant and not directly to the cooling tower sump.

CHAPTER 7
RECOMMENDATIONS

Based on the conclusion, this study is purely case specific and needs to be tested with many real plants with complete complex cooling tower system for better understanding. Also the study need to be taken into consideration the effect of variation in wet bulb temperature (day and night).

An empirical model should be developed for the thermal evaluation of the cooling tower to predict the performance of the cooling tower system and not based on the cooling tower performance curve.

A better systematic approach for the cooling water redistribution method should be developed. Any possibilities for the cooling water reuse should be investigated. The effect of reducing the mass and to avoid pinch should be studied for the cooling tower system. Any possibilities to break the cooling tower water loops in the cooling tower water network should be explored.

Nowadays advanced process control able to take multiple inputs and generate multiple outputs using soft sensors (software). This enables the controller to control the flow distribution in the cooling tower water network more accurately and the system can be flexible to the surrounding wet bulb temperature. Study has to be done using advanced control system to extract out the best from the cooling tower system.

Lastly, this study can be integrated with the improvement of the cooling tower internals such as the fills, the water treatment programme and varying the air flow rate.

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APPENDICES

APPENDIX 1

THE CROSS FLOW COOLING TOWER PERFORMANCE CURVE

APPENDIX 1

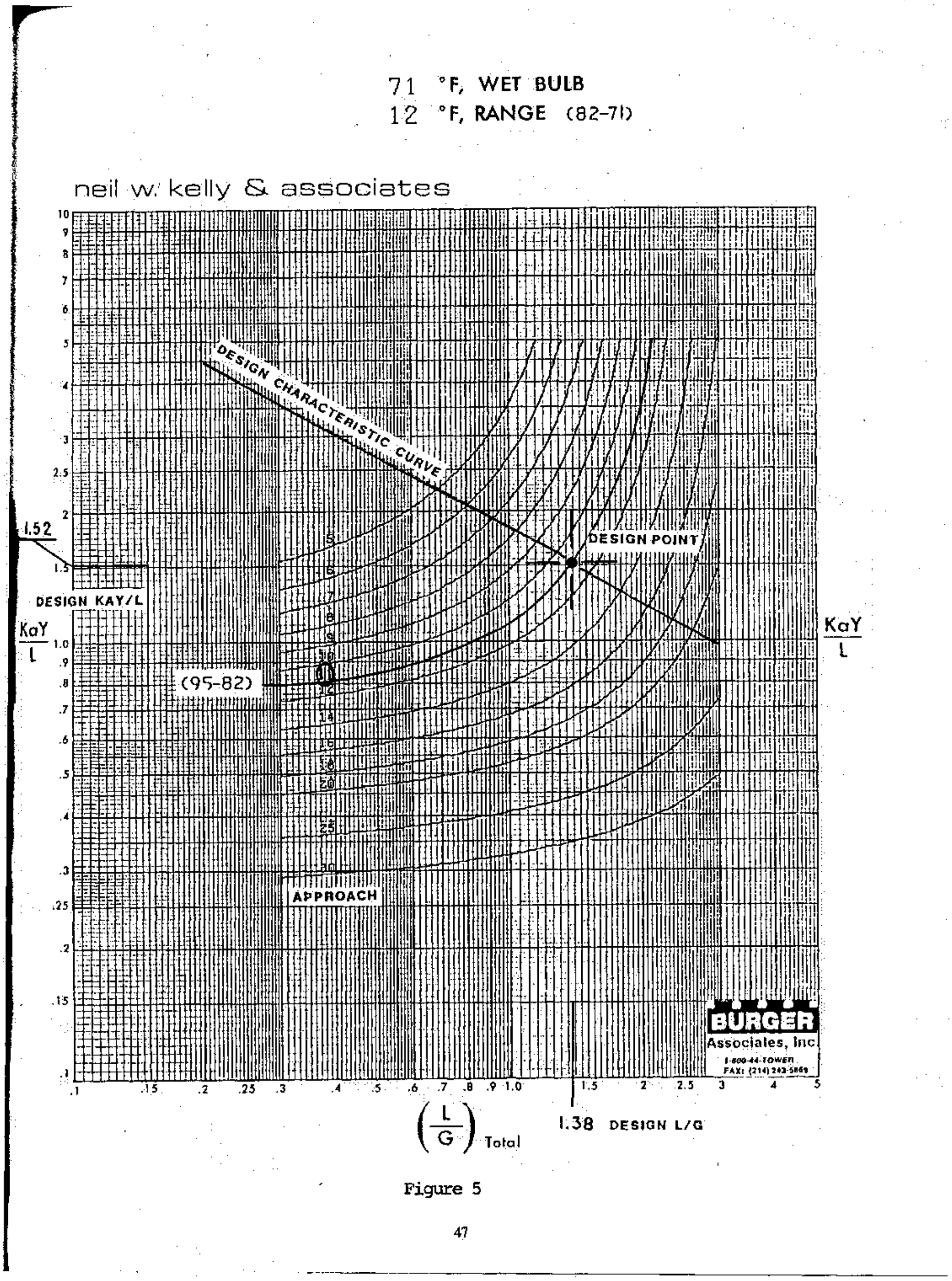

Figure A1.1: The Cross Flow Cooling Tower Performance Curve

Ref: Burger (1995)

APPENDIX 2

THE CHILLER VAPOR COMPRESSION REFRIGERATION CYCLE

APPENDIX 2

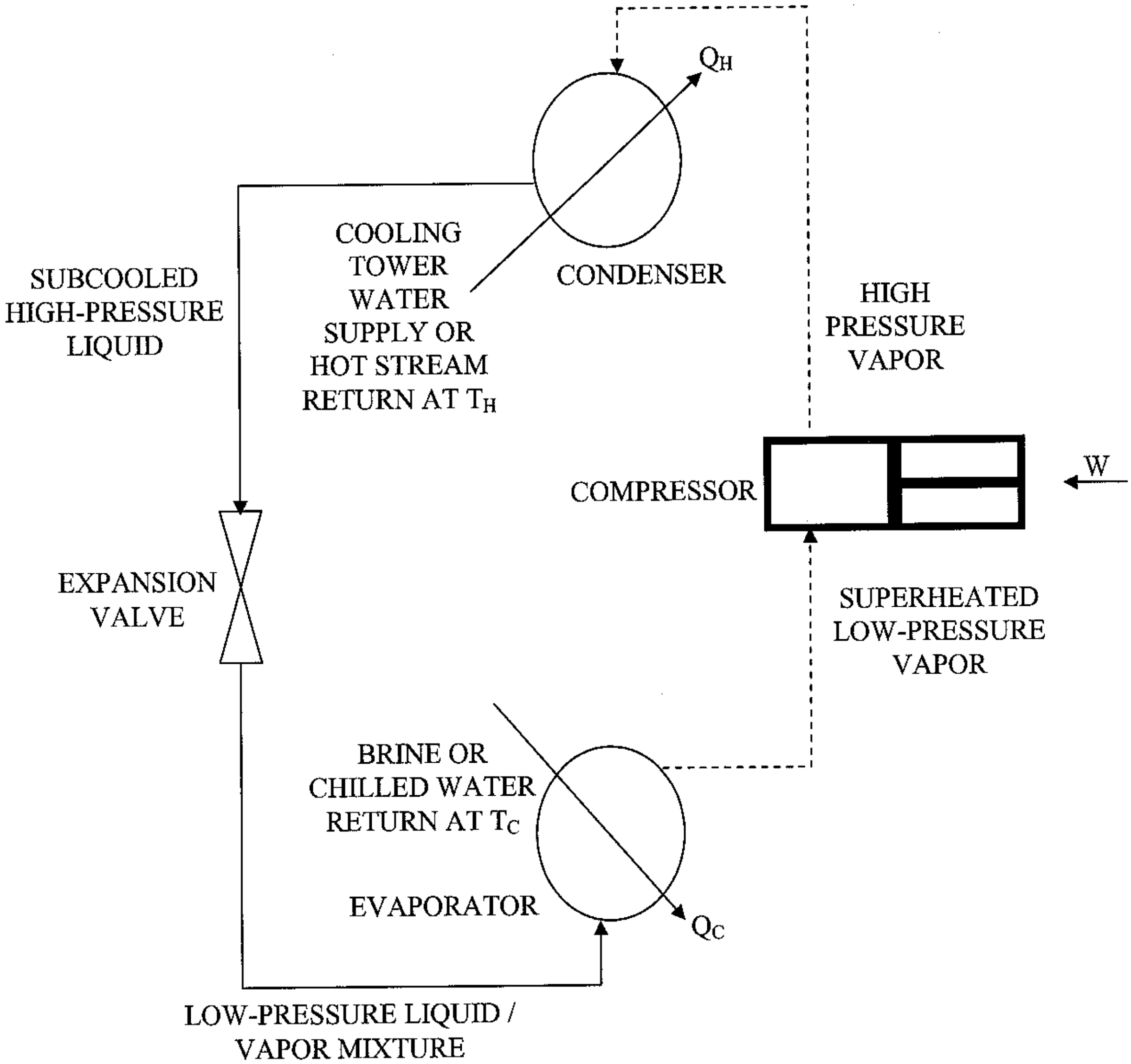

Figure A2.1: The Chiller Vapor Compression Refrigeration Cycle

Ref: Bela G. Liptak. (1987)

APPENDIX 3

CALCULATION
CASE 1: VAPOR COMPRESSION REFRIGERATION CYCLE (R-134a AND AMMONIA)

APPENDIX 3

CALCULATION

CASE 1.1: VAPOR COMPRESSION REFRIGERATION CYCLE (REFRIGERANT 134a)

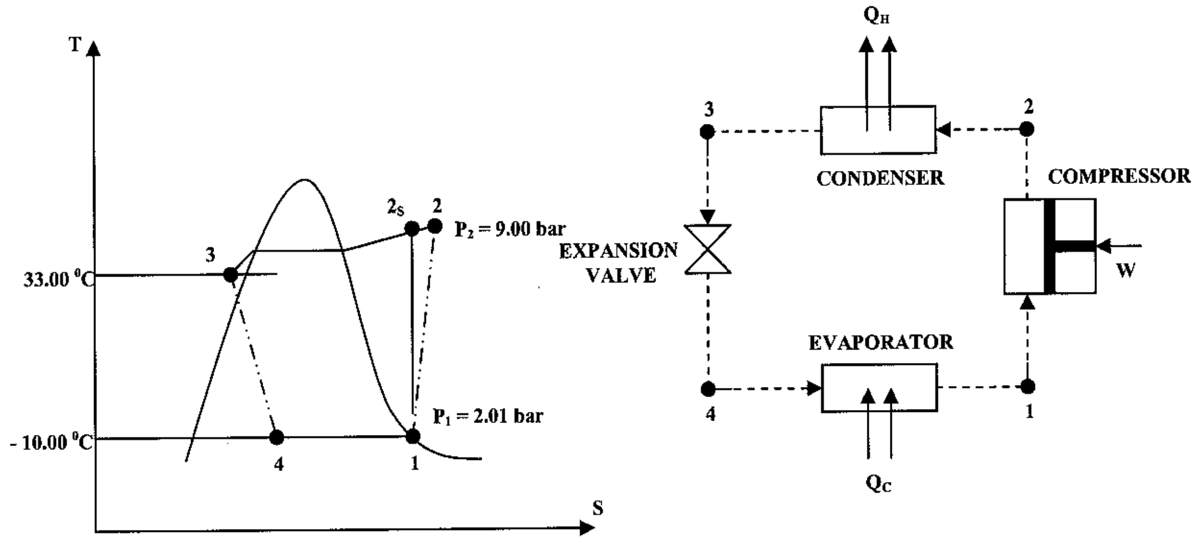

Figure A3.1: Vapor Compression Refrigeration Cycle (R-134a)

Assumptions

1. Each component of the cycle is analyzed as a control volume at steady state.
2. There are no pressure drops through the evaporator and condenser.
3. The compressor operates adiabatically with an efficiency of 80.00 %. The expansion through the valve is a throttling process and operates adiabatically.
4. Kinetic and potential energy effects are negligible.
5. Saturated vapor refrigerant at – 10.00 ^{0}C enters the compressor, and liquid refrigerant at 33.00 ^{0}C leaves the condenser.
6. All the process is internally reversible except expansion valve and compressor.

Point 1 – Saturated vapor condition (P = 2.01 bar, T = - 10.00 ^{0}C)

Point 2 – Superheated vapor condition (P = 9.00 bar, T = 48.03 ^{0}C)

Point 2s – Superheated vapor condition (P = 9.00 bar, T = 41.03 ^{0}C)

Point 3 – Sub cooled liquid condition (P = 9.00 bar, T = 33.00 ^{0}C)

Point 4 – Saturated liquid – vapor mixture (P = 2.01 bar, T = - 10.00 ^{0}C)

Note: The enthalpy data is taken from Michael J. Moran & Howard N. Shapiro 2000, *Fundamental of Engineering Thermodynamics*, 4^{th} Edition, New York, John Wiley & Sons, Inc.

Nomenclatures

W = Compression work (kW)

Q_H = Heat transfer from the refrigerant to the cooling water in the condenser (kJ/s)

Q_C = Heat transfer from the distillate to the refrigerant in the evaporator (kJ/s)

h = Enthalpy (kJ/kg)

s = Entropy (kJ/kg.K)

η_c = Efficiency of the compressor

m = Mass flow rate of the refrigerant (kg/s)

x = The quality of the vapor-liquid mixture

Calculations

Point 1: Saturated Vapor at – 10.00 ^{0}C

$h_1 = 241.350$ kJ/kg

$s_1 = 0.9253$ kJ/kg.K

Point 2_s: Superheated Vapor at 9.00 bar

$h_{2s} = 272.390$ kJ/kg

$s_{2s} = 0.9253$ kJ/kg.K

Point 2: Superheated Vapor at 9.00 bar

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} = 0.80$$

$$h_2 = \frac{h_{2s} - h_1}{\eta_c} + h_1 = 280.150 \text{ kJ/kg}$$

Point 3: Sub Cooled Liquid at 33.00 ^{0}C

$h_3 = h_{f@33\,^0C} = 95.850$ kJ/kg

Point 4: Saturated Liquid-Vapor Mixture at – 10.00 ^{0}C

It is a throttling expansion, $h_3 = h_4 = 95.850$ kJ/kg

The quality, $x_4 = \dfrac{h_4 - h_{f4}}{h_{g4} - h_{f4}} = \dfrac{95.850 - 36.970}{204.390} = \mathbf{0.2881}$

m = 0.08 kg/s

$W = m (h_2 - h_1)$

W = (0.08 kg/s) x (280.150 kJ/kg – 241.350 kJ/kg) x (1 kW/kJ/s)

W = 3.104 kW

$Q_H = m (h_2 - h_3)$

Q_H = (0.08 kg/s) x (280.150 kJ/kg – 95.850 kJ/kg)

Q_H = 14.744 kJ/s

$Q_C = m (h_1 - h_4)$

Q_C = (0.08 kg/s) x (241.350 kJ/kg – 95.850 kJ/kg)

Q_C = 11.640 kJ/s

The cooling water will pass through the condenser of the refrigeration equipment to remove the heat from the refrigerant. Assuming the inlet temperature of the cooling water to the condenser is 29.00 ^{0}C and the outlet temperature of the cooling water from the condenser is 32.00 ^{0}C and the flow is counter current.

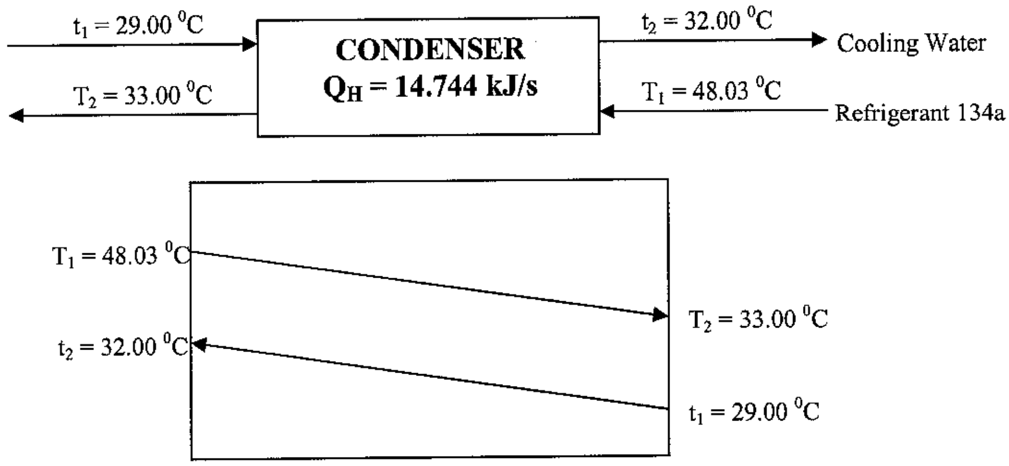

Figure A3.2: Condenser of the Refrigeration Equipment and the Temperature Profile

The log mean temperature difference:

$$\Delta T_{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln[(T_1 - t_2) / (T_2 - t_1)]}$$

$$\Delta T_{LMTD} = \frac{(48.03 - 32.00) - (33.00 - 29.00)}{\ln[(48.03 - 32.00) / (33.00 - 29.00)]}$$

$$\mathbf{\Delta T_{LMTD} = 8.67\ ^{0}C}$$

Based on the reference, Burger (1995), every drop of 0.60 ^{0}C of inlet cooling water temperature, compression work can be saved by 3.00 %. Assuming the cooling water flow rate remains same.

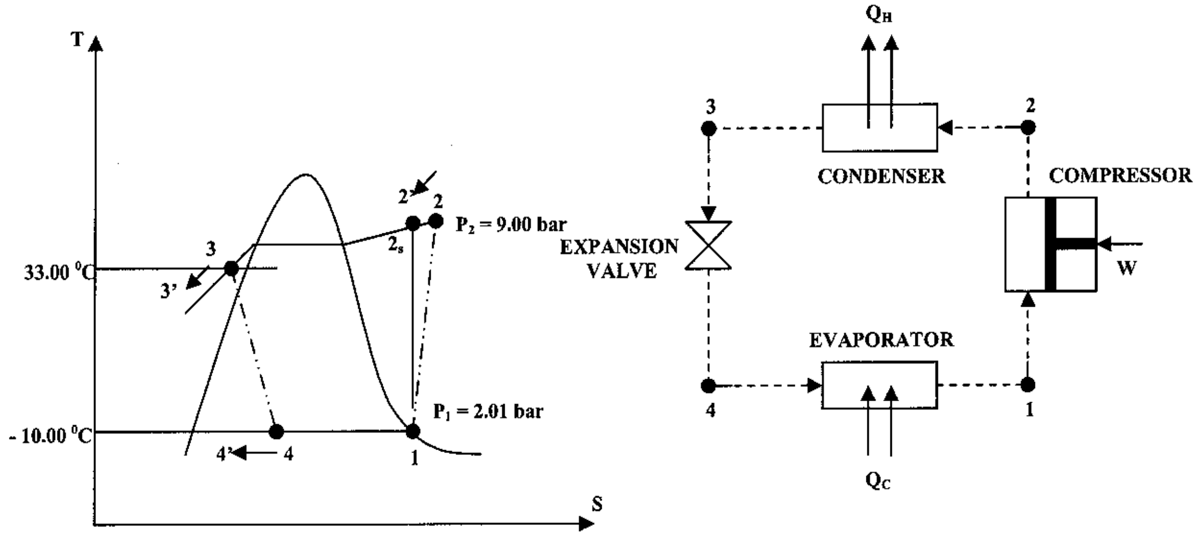

Figure A3.3: Vapor Compression Refrigeration Cycle (R-134a)
after 0.60 ^{0}C Reduction in Cooling Water Temperature

Point 1 – Saturated vapor condition (P = 2.01 bar, T = - 10.00 ^{0}C) [Assume Fixed]

Point 2' – Superheated vapor condition (P = 9.00 bar, T = 46.94 ^{0}C)

Point 2s – Superheated vapor condition (P = 9.00 bar, T = 41.03 ^{0}C) [Adiabatic]

Point 3' – Sub cooled liquid condition (P = 9.00 bar, T = 32.19 ^{0}C)

Point 4' – Saturated liquid – vapor mixture (P = 2.01 bar, T = - 10.00 ^{0}C)

Backward Calculations

Point 1: Saturated Vapor at – 10.00 ^{0}C

h_1 = 241.350 kJ/kg

s_1 = 0.9253 kJ/kg.K

Compression work reduces by 3.00 %

W' = W – (W x 3.00 %)

W' = 3.104 kW – (3.104 kW x 3.00 %)

W' = 3.011 kW

W' = (3.011 kJ/s) x (1 s / 0.08 kg)

W' = 37.638 kJ/kg

$W' = h'_2 - h_1$

$h'_2 = W' + h_1$

h'_2 = 37.638 kJ/kg + 241.350 kJ/kg

h'_2 = 278.988 kJ/kg

Q'_H = Constant = 14.744 kJ/s

$Q'_H = h'_2 - h'_3$

$h'_3 = h'_2 - Q'_H$

h'_3 = 278.988 kJ/kg - (14.744 kJ/s) x (1 s / 0.08 kg)

h'_3 = 94.688 kJ/kg

Isenthalpic Expansion

$h'_3 = h'_4$ = 94.688 kJ/kg

The quality, $x'_4 = \dfrac{h'_4 - h_{f4}}{h_{g4} - h_{f4}} = \dfrac{94.688 - 36.970}{204.390} = \mathbf{0.2824}$

$Q'_C = m (h_1 - h'_4)$

Q'_C = (0.08 kg/s) x (241.350 kJ/kg – 94.688 kJ/kg)

Q'_C = 11.733 kJ/s

% Increase in Q_C = (11.733 – 11.640) / 11.640 x 100.00 % = **0.80 %**

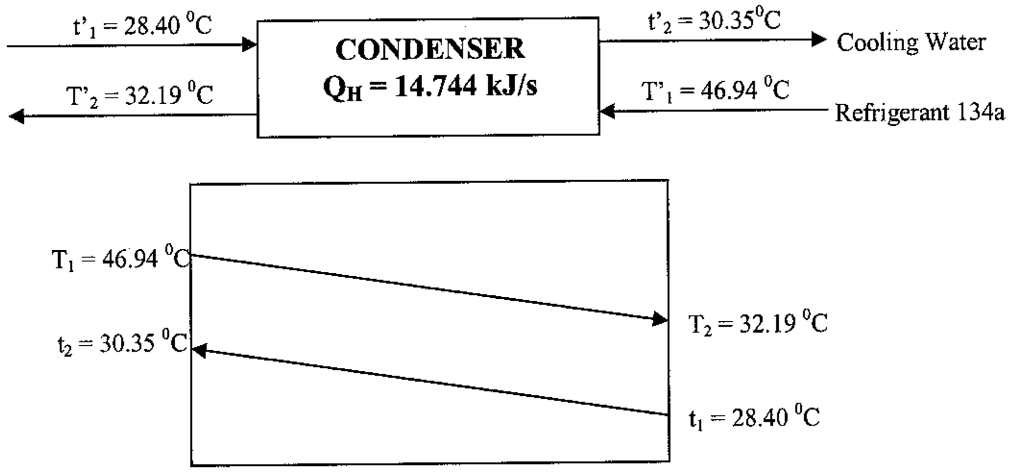

Figure A3.4: Condenser of the Refrigeration Equipment and the Temperature Profile after 0.60 ^{0}C Reduction in Cooling Water Temperature

Q_H = Constant

U = Overall Heat Transfer Coefficient (kW/m^2.^{0}C)

A = Heat Transfer Area (m^2)

$Q_H = UA\Delta T_{LMTD}$ = Constant

Assume the U, A, Q_H and ΔT_{LMTD} remains same.

The refrigerant and the cooling water will adjust its temperature so that ΔT_{LMTD} remains constant because the restriction that the Q_H constant (no extra heat is removed from the refrigerant).

The log mean temperature difference:

$$\Delta T'_{LMTD} = \frac{(T'_1 - t'_2) - (T'_2 - t'_1)}{\ln[(T'_1 - t'_2)/(T'_2 - t'_1)]} = \mathbf{8.67\ ^0C}$$

To find cooling water outlet from condenser temperature, t_2:

$$\Delta T'_{LMTD} = \frac{(46.94 - t'_2) - (32.19 - 28.40)}{\ln[(46.94 - t'_2)/(32.19 - 28.40)]} = \mathbf{8.67\ ^0C}$$

By using trial and error method, $\mathbf{t'_2 = 30.35\ ^0C}$

% Reduction in t_2 = (32.00 – 30.35) / 32.00 x 100.00 % = **5.16 %**

Adiabatic Compression Efficiency, η'_c:

$$\eta'_c = \frac{h_{2s} - h_1}{h'_2 - h_1}$$

$\eta'_c =$ [272.390 - 241.350] / [278.988 – 241.350] x 100.00 % = **82.44 %**

The efficiency of the compressor improves by **2.44 %.**

APPENDIX 3

CALCULATION

CASE 1.2: VAPOR COMPRESSION REFRIGERATION CYCLE

(AMMONIA – HIGH EFFICIENCY REFRIGERANT)

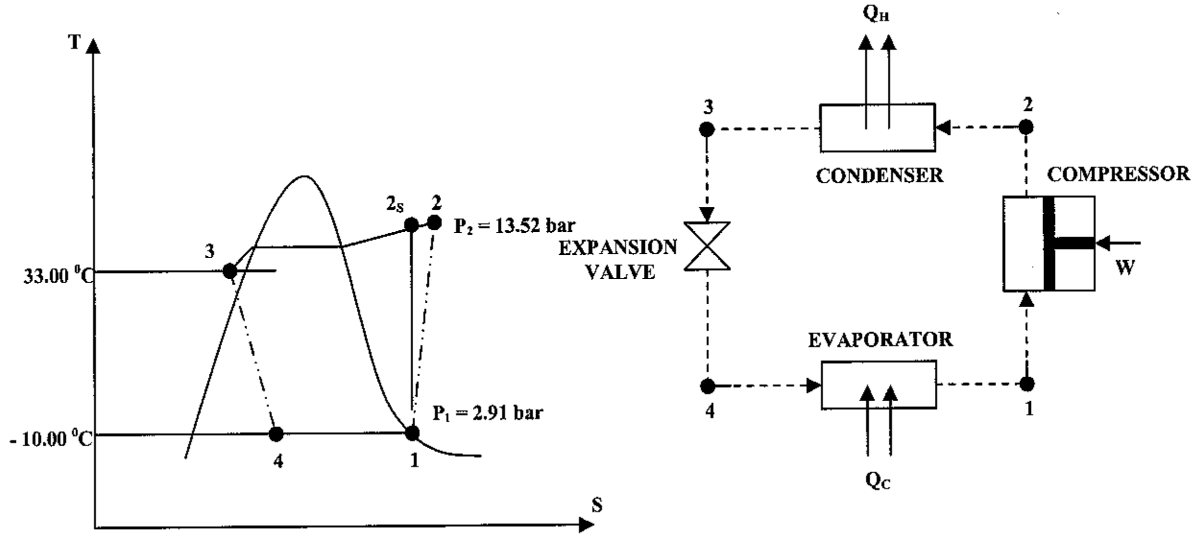

Figure A3.5: Vapor Compression Refrigeration Cycle (Ammonia)

Assumptions

1. Each component of the cycle is analyzed as a control volume at steady state.
2. There are no pressure drops through the evaporator and condenser.
3. The compressor operates adiabatically with an efficiency of 80.00 %. The expansion through the valve is a throttling process and operates adiabatically.
4. Kinetic and potential energy effects are negligible.
5. Saturated vapor refrigerant at – 10.00 ^{0}C enters the compressor, and liquid refrigerant at 33.00 ^{0}C leaves the condenser.
6. All the process is internally reversible except expansion valve and compressor.

Point 1 – Saturated vapor condition (P = 2.91 bar, T = - 10.00 ^{0}C)

Point 2 – Superheated vapor condition (P = 13.52 bar, T = 122.37 ^{0}C)

Point 2s – Superheated vapor condition (P = 13.52 bar, T = 100.37 ^{0}C)

Point 3 – Sub cooled liquid condition (P = 13.52 bar, T = 33.00 ^{0}C)

Point 4 – Saturated liquid – vapor mixture (P = 2.91 bar, T = - 10.00 ^{0}C)

Note: The enthalpy data is taken from Michael J. Moran & Howard N. Shapiro 2000, *Fundamental of Engineering Thermodynamics*, 4^{th} Edition, New York, John Wiley & Sons, Inc.

Nomenclatures

W = Compression work (kW)

Q_H = Heat transfer from the refrigerant to the cooling water in the condenser (kJ/s)

Q_C = Heat transfer from the distillate to the refrigerant in the evaporator (kJ/s)

h = Enthalpy (kJ/kg)

s = Entropy (kJ/kg.K)

η_c = Efficiency of the compressor

m = Mass flow rate of the refrigerant (kg/s)

x = The quality of the vapor-liquid mixture

Calculations

Point 1: Saturated Vapor at – 10.00 ^{0}C

$h_1 = 1430.550$ kJ/kg

$s_1 = 5.4662$ kJ/kg.K

Point 2_s: Superheated Vapor at 13.52 bar

$h_{2s} = 1653.699$ kJ/kg

$s_{2s} = 5.4662$ kJ/kg.K

Point 2: Superheated Vapor at 13.52 bar

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} = 0.80$$

$$h_2 = \frac{h_{2s} - h_1}{\eta_c} + h_1 = 1709.486 \text{ kJ/kg}$$

Point 3: Sub Cooled Liquid at 33.00 ^{0}C

$h_3 = h_{f@33\,^0C} = 337.050$ kJ/kg

Point 4: Saturated Liquid-Vapor Mixture at – 10.00 ^{0}C

It is a throttling expansion, $h_3 = h_4 = 337.050$ kJ/kg

The quality, $x_4 = \dfrac{h_4 - h_{f4}}{h_{g4} - h_{f4}} = \dfrac{337.050 - 133.940}{1296.610} = \mathbf{0.1566}$

m = 0.08 kg/s

$W = m\,(h_2 - h_1)$

W = (0.08 kg/s) x (1709.486 kJ/kg – 1430.550 kJ/kg) x (1 kW/kJ/s)

W = 22.315 kW

$Q_H = m\,(h_2 - h_3)$

Q_H = (0.08 kg/s) x (1709.486 kJ/kg – 337.050 kJ/kg)

Q_H = 109.795 kJ/s

$Q_C = m\,(h_1 - h_4)$

Q_C = (0.08 kg/s) x (1430.550 kJ/kg – 337.050 kJ/kg)

Q_C = 87.480 kJ/s

The cooling water will pass through the condenser of the refrigeration equipment to remove the heat from the refrigerant. Assuming the inlet temperature of the cooling water to the condenser is 29.00 ^{0}C and the outlet temperature of the cooling water from the condenser is 32.00 ^{0}C and the flow is counter current.

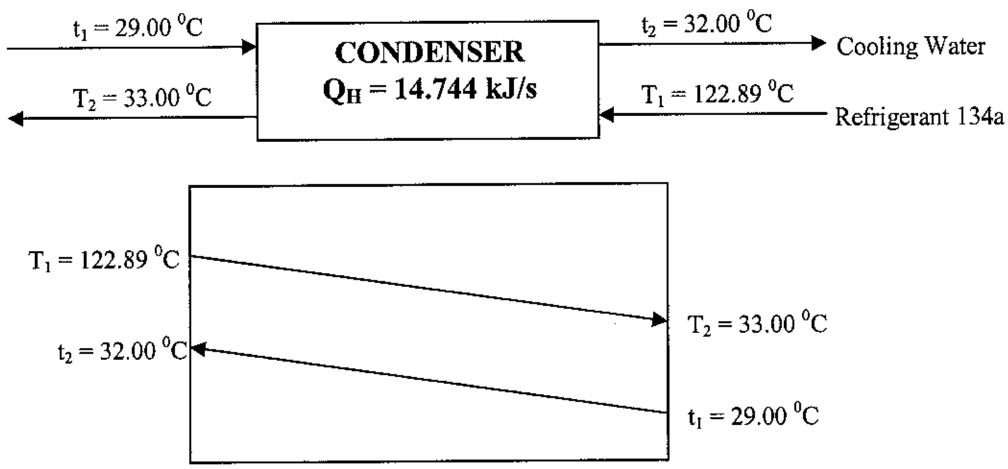

Figure A3.6: Condenser of the Refrigeration Equipment (Ammonia) and the Temperature Profile

The log mean temperature difference:

$$\Delta T_{LMTD} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln[(T_1 - t_2)/(T_2 - t_1)]}$$

$$\Delta T_{LMTD} = \frac{(122.89 - 32.00) - (33.00 - 29.00)}{\ln[(122.89 - 32.00)/(33.00 - 29.00)]}$$

$$\mathbf{\Delta T_{LMTD} = 27.82\ {}^0C}$$

Based on the reference, Burger (1995), every drop of 0.60 ^{0}C of inlet cooling water temperature, compression work can be saved by 3.00 %. Assuming the cooling water flow rate remains same.

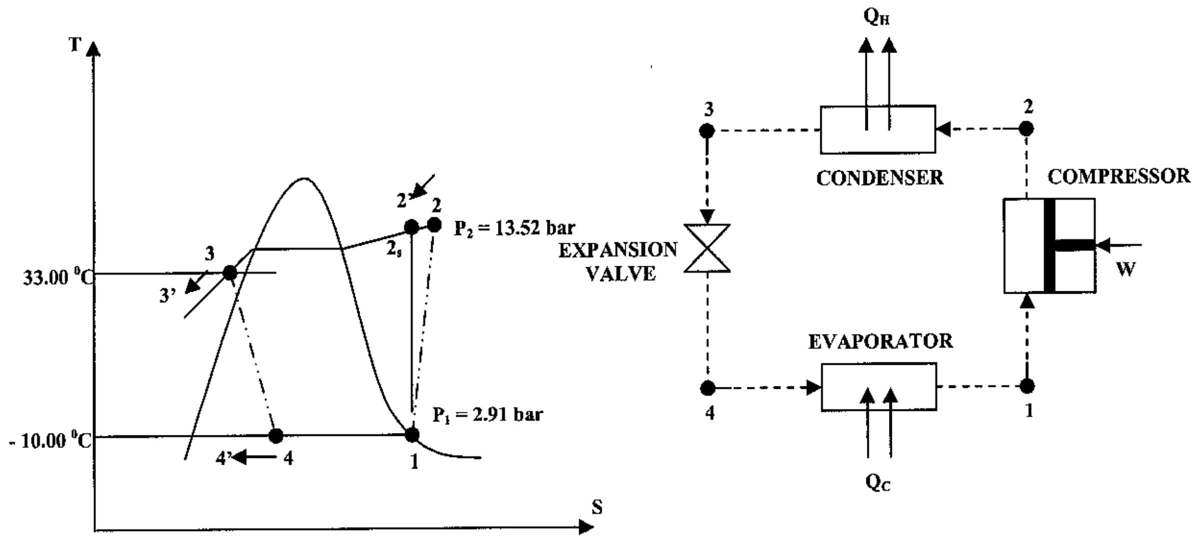

Figure A3.7: Vapor Compression Refrigeration Cycle (Ammonia)
after 0.60 ^{0}C Reduction in Cooling Water Temperature

Point 1 – Saturated vapor condition (P = 2.91 bar, T = - 10.00 ^{0}C) [Assume Fixed]

Point 2' – Superheated vapor condition (P = 13.52 bar, T = 119.04 ^{0}C)

Point 2s – Superheated vapor condition (P = 13.52 bar, T = 100.37 ^{0}C) [Adiabatic]

Point 3' – Sub cooled liquid condition (P = 13.52 bar, T = 31.28 ^{0}C)

Point 4' – Saturated liquid – vapor mixture (P = 2.91 bar, T = - 10.00 ^{0}C)

Backward Calculations

Point 1: Saturated Vapor at – 10.00 ^{0}C

h_1 = 1430.550 kJ/kg

s_1 = 5.4662 kJ/kg.K

Compression work reduces by 3.00 %

W' = W – (W x 3.00 %)

W' = 22.315 kW – (22.315 kW x 3.00 %)

W' = 21.646 kW

W' = (21.646 kJ/s) x (1 s / 0.08 kg)

W' = 270.575 kJ/kg

W' = $h'_2 - h_1$

h'_2 = W' + h_1

h'_2 = 270.575 kJ/kg + 1430.550 kJ/kg

h'_2 = 1701.125 kJ/kg

Q'_H = Constant = 109.795 kJ/s

$Q'_H = h'_2 - h'_3$

$h'_3 = h'_2 - Q'_H$

h'_3 = 1701.125 kJ/kg - (109.795 kJ/s) x (1 s / 0.08 kg)

h'_3 = 328.688 kJ/kg

Isenthalpic Expansion

$h'_3 = h'_4$ = 328.688 kJ/kg

The quality, $x'_4 = \dfrac{h'_4 - h_{f4}}{h_{g4} - h_{f4}} = \dfrac{328.688 - 133.940}{1296.610}$ = **0.1502**

$Q'_C = m\ (h_1 - h'_4)$

Q'_C = (0.08 kg/s) x (1430.550 kJ/kg – 328.688 kJ/kg)

Q'_C = 88.149 kJ/s

% Increase in Q_C = (88.149 – 87.480) / 87.480 x 100.00 % = **0.76 %**

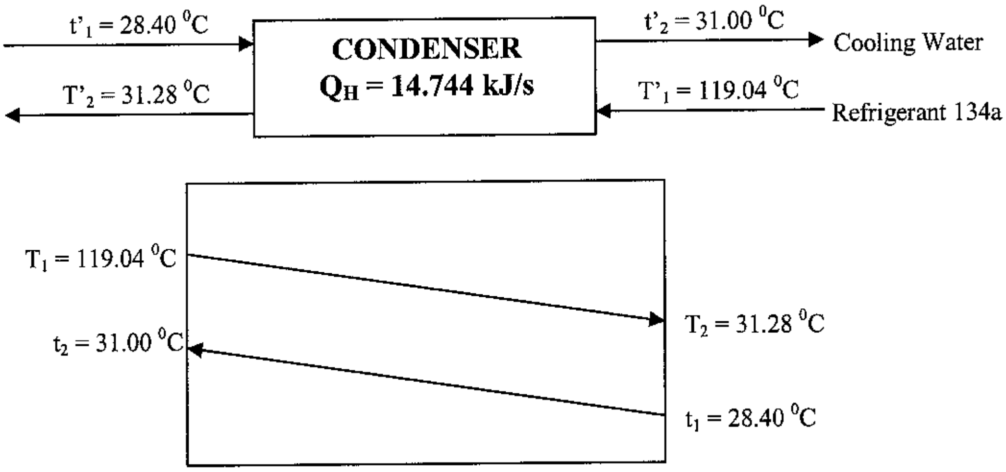

Figure A3.8: Condenser of the Refrigeration Equipment (Ammonia) and the Temperature Profile after 0.60 ^{0}C Reduction in Cooling Water Temperature

Q_H = Constant

U = Overall Heat Transfer Coefficient (kW/m^2.^{0}C)

A = Heat Transfer Area (m^2)

$Q_H = UA\Delta T_{LMTD}$ = Constant

Assume the U, A, Q_H and ΔT_{LMTD} remains same.

The refrigerant and the cooling water will adjust its temperature so that ΔT_{LMTD} remains constant because the restriction that the Q_H constant (no extra heat is removed from the refrigerant).

The log mean temperature difference:

$$\Delta T'_{LMTD} = \frac{(T'_1 - t'_2) - (T'_2 - t'_1)}{\ln[(T'_1 - t'_2)/(T'_2 - t'_1)]} = \mathbf{27.82\ ^0C}$$

To find cooling water outlet from condenser temperature, t_2:

$$\Delta T'_{LMTD} = \frac{(119.04 - t'_2) - (31.28 - 28.40)}{\ln[(119.04 - t'_2)/(31.28 - 28.40)]} = \mathbf{27.82\ ^0C}$$

By using trial and error method, **t'_2 = 29.00 ^{0}C**

% Reduction in t_2 = (32.00 – 29.00) / 32.00 x 100.00 % = **9.38 %**

Adiabatic Compression Efficiency, η'_c:

$$\eta'_c = \frac{h_{2s} - h_1}{h'_2 - h_1}$$

$\eta'_c =$ [1653.699 – 1430.550] / [1701.125 – 1430.550] x 100.00 % = **82.47 %**

The efficiency of the compressor improves by **2.47 %.**

APPENDIX 4

CALCULATION
CASE 2: COOLING TOWER SYSTEM
(ONE CHILLER)

APPENDIX 4

CALCULATION

CASE 2: COOLING TOWER SYSTEM

(ONE CHILLER)

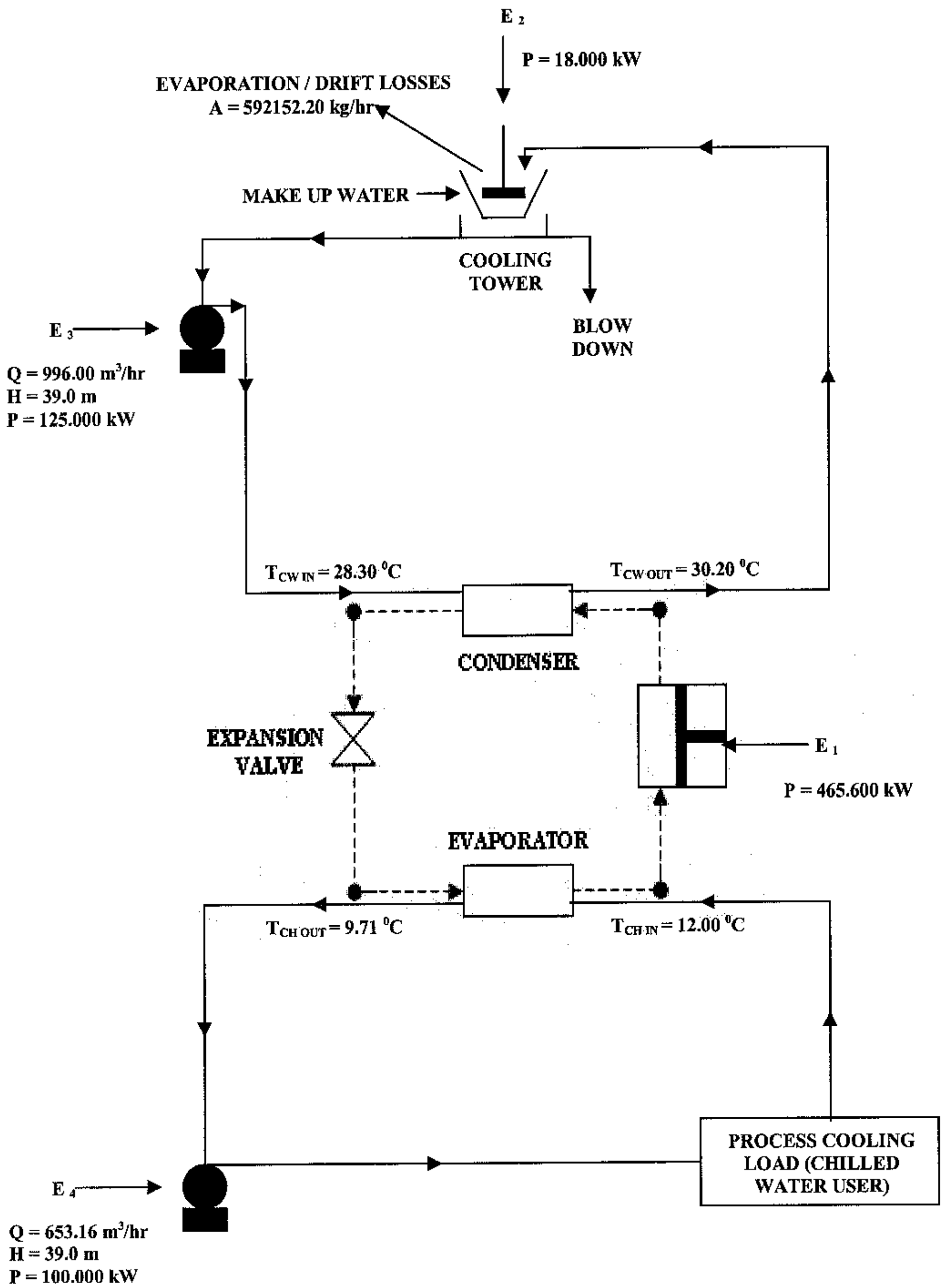

Figure A4.1: Cooling Tower System with One Chiller

Assumptions

1. The chiller can be treated as a single unit or a combination of multiple units.
2. The air wet bulb temperature = 21.67 ^{0}C.
3. The air dry bulb temperature = 26.67 ^{0}C.
4. The cooling tower contacting pattern is cross flow.
5. The concentration ratio of the cooling tower is 6.
6. The function of the chiller is to produce chilled water.
7. The chiller using 12.00 kg/s Refrigerant 134a.
8. Each component of the cycle is analyzed as a control volume at steady state.
9. There are no pressure drops through the evaporator and condenser.
10. The compressor operates adiabatically with an efficiency of 80.00 %. The expansion through the valve is a throttling process and operates adiabatically.
11. Kinetic and potential energy effects are negligible.
12. Saturated vapor refrigerant at – 10.00 ^{0}C enters the compressor, and liquid refrigerant at 33.00 ^{0}C leaves the condenser.
13. All the process is internally reversible except expansion valve and compressor.

Nomenclatures

E_1 = Compressor power consumption = W (kW)

E_2 = Cooling tower fan power consumption (kW)

E_2 = Cooling tower water circulation pump power consumption (kW)

E_4 = Chilled water circulation pump power consumption (kW)

A = Air flow rate (kg/hr)

Q = Water flow rate (m^3/hr)

H = Pump head requirements (m)

P = Power consumption (kW)

Q_H = Heat transfer from the refrigerant to the cooling water in the condenser (kJ/s)

Q_C = Heat transfer from the chilled water to the refrigerant in the evaporator (kJ/s)

h = Enthalpy (kJ/kg)

s = Entropy (kJ/kg.K)

η_c = Efficiency of the compressor

m = Mass flow rate of the refrigerant (kg/s)

Calculations

Using the similar calculation method shown in Appendix 3 for the vapor compression refrigeration cycle and referring to the Figure A4.1, the following results are obtained:

E_1 = 465.600 kW

E_2 = 18.000 kW

E_3 = 125.000 kW

E_4 = 100.000 kW

Q_H = 2211.600 kJ/s

Q_C = 1746.000 kJ/s

m = 12.00 kg/s

The above results are based on the cooling tower water temperature 28.30 ^{0}C.

The cooling tower water circulation rate, **L = 996.00 m^3/hr**

The air flow rate, **G = 592152.20 kg/hr**

The air (wet) density = 1.225 kg/m^3 (Ref: www. hypertextbook.com)

L = (996.00 m^3 / 1hr) x (1hr / 60 min) x (1000 lit / 1 m^3) x (0.264172 gal (US) / 1 lit)

L = 4385.25 GPM

G = (592152.20 kg / 1 hr) x (1 hr / 60 min) x (1 m^3 / 1.225 kg) x (3.28084^3 ft^3 / 1 m^3)

G = 284512.38 CFM

To determine the thermal demand of the cooling tower:

L = numerical value of water in GPM

G = numerical value of air in CFM

GPM = (L x gal / min) x (60 min / 1 hr) x (0.1337 ft^3 / 1 gal) x (62.06 lbs / ft^3)

GPM = L x (497.84 lbs / 1 hr)

CFM = (G x ft^3 / min) x (60 min / 1 hr) x (1 lbs / 14.078 ft^3)

CFM = G x (4.262 lbs / hr)

L/G = (497.84 lbs water / 1 hr) / (4.262 lbs air / 1 hr) = 116.8

In actuality, the constant arise from approximately 115.9 to 117.1 depending upon altitude and temperatures. However, the average 116.5 is selected and the variation error can be easily absorbed by the width of a pencil mark on the performance curves since the curves are not that accurate to the third decimal place.

Therefore, the L/G ratio for the cooling tower water system in Figure A4.1:

L/G = (4385.25 GPM / 284512.38 CFM) x 116.5 = **1.80**

Referring to the Appendix 1: Cross Flow Cooling Tower Performance Curve, the cooling tower thermal loading is **1.30**.

The inlet cooling water to the cooling tower temperature = 30.20 ^{0}C

The outlet cooling water from the cooling tower temperature = 28.30 ^{0}C

The air wet bulb temperature = 21.67 ^{0}C

The air dry bulb temperature = 26.67 ^{0}C

The range = T Hot In – T Cold Out = 30.20 ^{0}C – 28.30 ^{0}C = **1.90 ^{0}C**

The approach = T Cold Out – T Wet Bulb = 28.30 ^{0}C – 21.67 ^{0}C = **6.63 ^{0}C**

Power consumption

The total electrical power consumption = $E_1 + E_2 + E_3 + E_4$ = 708.600 kW

Assuming plant operating 333.33 days per year (8000 hr).

Total annual power cost (RM/yr):

= (708.600 kW) x (8000 hr / 1 yr) x (RM 0.22 / 1 kWhr)

= **RM 1,247,136.00 / yr**

Water consumption (Ref: NALCO)

The cooling tower water circulation rate, **L = 996 m^3/hr**

The cooling tower water concentration ratio, **CR** = **6**

The cooling tower water drifts losses, DR:

DR = 0.15 % x Circulation Rate

DR = 0.15 % x 996 m^3/hr = **149.4 m^3/hr**

The cooling tower water evaporation losses, E:

E = % E x Circulation Rate

% E = [(T Dry Bulb – 1.6667) x Km + 0.1098] x Range (^{0}C)

Range (^{0}C)	Percent Relative Humidity		
	< 30 %	30 % - 90 %	> 90 %
> 7.2 ^{0}C	Km = 0.0013	Km = 0.0013	Km = 0.0013
3.9 ^{0}C – 7.2 ^{0}C	Km = 0.0029	Km = 0.0019	Km = 0.0010
< 3.9 ^{0}C	Km = 0.0058	Km = 0.0032	Km = 0.0010

Table A4.1: The Km Factor Based on Range and Relative Humidity

Referring to the psychometric chart (Felder & Rousseau 2000), the relative humidity = **40 %**

Therefore, the Km factor (based on Table A4.1) = 0.0032

% E = [(26.67 – 1.6667) x 0.0032 + 0.1098] x Range

% E = 0.18981 x Range

Therefore, the cooling tower water evaporation losses, E:

E = 0.18981 x 1.9 x (996.00 m^3 / 1 hr)

E = 359.20 m^3/hr

The cooling tower water blow down rate, BR:

BR = E / (CR -1)

BR = (359.20 m^3 / 1 hr) / (6 – 1)

BR = 71.84 m^3/hr

The total make up water, MU:

MU = Evaporation + Drift Losses + Blow Down

MU = 359.20 m^3/hr + 149.4 m^3/hr + 71.84 m^3/hr

MU = 580.44 m^3/hr

Assuming plant operating 333.33 days per year (8000 hr).

Total annual water cost (RM/yr)

= (580.44 m^3 / 1 hr) x (8000 hr / 1 yr) x (RM 3.00 / 1 m^3)

= **RM 13,930,560 / yr**

Performance Improvement

By improving the performance of the cooling tower, the outlet temperature of the cooling tower water can further be cooled closer to the air wet bulb temperature. Based on the reference, Burger (1995), every drop of 0.60 ^{0}C of inlet cooling water temperature, compression work can be saved by 3.00 %. Therefore every drop of 1.00 ^{0}C temperature, compression work can be saved by 5.00 % (the linear relationship applies for low temperature difference only).

Referring to Appendix 1: Cross Flow Cooling Tower Performance Curve, in order to reduce the cooling water temperature by 1.00 ^{0}C colder, the L/G ratio should be **1.65**.

The thermal loading will be **1.35**. Since the performance of the system is improved without the modification of the cooling tower internals, thus the air flow rate remains constant. The only way to reduce the L/G ratio from 1.80 to 1.65 is by reducing the cooling tower water circulation rate. Thus, the new target for cooling tower water circulation rate:

L' = 1.65 x 284512.38 CFM / 116.5 = 4029.57 GPM = **915.22 m^3/hr**

By this, the cooling water circulation rate will be lesser but colder.

The following parameters are determined:

E'_1 = 465.600 kW – (465.600 kW x 5.00 %) = **442.320 kW**

E'_2 = **18.000 kW** (No changes in the air flow rate)

E'_3 = 125.000 kW x (915.22 / 996) = **114.862 kW** (Assume Q is proportional to P)

E'_4 = **100.000 kW** (No changes in the chilled water flow rate)

Q'_H = **2211.600 kJ/s** (Remains fixed)

$Q'_C = Q'_H - W'$ = 2211.600 kJ/s – 442.320 kJ/s = **1769.280 kJ/s**

% Increase in Q_C = (1769.280 – 1746.000) / 1746.000 x 100.00 % = **1.33 %**

The chilled water temperature (product) will be colder as the result of the refrigeration effect.

$Q'_C = m_{CH} \times Cp \times (T_{CH\ IN} - T_{CH\ OUT})$

1769.280 kJ / 1 s = 653.16 m^3 / 1hr x (1 hr / 3600 s) x (1000 kg / 1 m^3) x (4.2 kJ / 1 kg.^{0}C) x (12 ^{0}C – $T_{CH\ OUT}$)

$T_{CH\ OUT}$ = 9.68 ^{0}C

% Reduction in chilled water temperature

= (9.71 – 9.68) / 9.71 x 100.00 % = **0.31 %**

The cooling water will pass through the condenser with lower mass and lower temperature.

$Q'_H = m_{CW} \times Cp \times (T_{CW\ OUT} - T_{CW\ IN})$

2211.600 kJ / 1 s = 915.22 m^3 / 1 hr x (1 hr / 3600 s) x (1000 kg / 1 m^3) x (4.2 kJ / 1 kg.^{0}C) x ($T_{CW\ OUT}$ - 27.30 ^{0}C)

$T_{CW\ OUT}$ = 29.37 ^{0}C

% Reduction in cooling water outlet temperature

= (30.20 – 29.37) / 30.2 x 100.00 % = **2.75 %**

The inlet cooling water to the cooling tower temperature = 29.37 ^{0}C

The outlet cooling water from the cooling tower temperature = 27.30 ^{0}C

The air wet bulb temperature = 21.67 ^{0}C

The air dry bulb temperature = 26.67 ^{0}C

The range = T Hot In – T Cold Out = 29.37 ^{0}C – 27.30 ^{0}C = **2.07 ^{0}C**

The approach = T Cold Out – T Wet Bulb = 27.30 ^{0}C – 21.67 ^{0}C = **5.63 ^{0}C**

Power consumption

The total electrical power consumption = $E_1 + E_2 + E_3 + E_4$ = 675.182 kW

Assuming plant operating 333.33 days per year (8000 hr).

Total annual energy cost (RM/yr)

= (675.182 kW) x (8000 hr / 1 yr) x (RM 0.22 / 1 kWhr)

= **RM 1,188,320.32 / yr**

Total annual power cost savings

= RM 1,247,136.00 / yr - RM 1,188,320.32 / yr

= **RM 58,815.68 / yr**

Water consumption (Ref: NALCO)

The cooling tower water circulation rate, **L' = 915.22 m^3/hr**

The cooling tower water concentration ratio, **CR = 6**

The cooling tower water drifts losses, DR:

DR = 0.15 % x Circulation Rate

DR = 0.15 % x 915.22 m^3/hr

DR = 137.28 m^3/hr

The cooling tower water evaporation losses, E:

E = % E x Circulation Rate

% E = [(T Dry Bulb – 1.6667) x Km + 0.1098] x Range (^{0}C)

Referring to the psychometric chart (Felder & Rousseau 2000), the relative humidity = **40 %**

Therefore, the Km factor (based on Table A4.1) = 0.0032

% E = [(26.67 – 1.6667) x 0.0032 + 0.1098] x Range

% E = 0.18981 x Range

Therefore, the cooling tower water evaporation losses, E:

$E = 0.18981 \times 2.07 \times 915.22\ m^3/hr$

$\mathbf{E = 359.60\ m^3/hr}$

The cooling tower water blow down rate, BR:

$BR = E / (CR - 1)$

$BR = (359.60\ m^3 / 1\ hr) / (6 - 1)$

$\mathbf{BR = 71.92\ m^3/hr}$

The total make-up water, MU:

MU = Evaporation + Drift Losses + Blow Down

$MU = 359.60\ m^3/hr + 137.28\ m^3/hr + 71.92\ m^3/hr$

$\mathbf{MU = 568.80\ m^3/hr}$

Assuming plant operating 333.33 days per year (8000 hr).

Total annual water cost (RM/yr)

$= (568.80\ m^3 / 1\ hr) \times (8000\ hr / 1\ yr) \times (RM\ 3.00 / m^3)$

= **RM 13,651,200 / yr**

Total annual water cost savings

= RM 13,930,560 / yr - RM 13,651,200 /yr

= **RM 279,360 / yr**

Total Annual Savings (Water + Power) = RM 279,360 / yr + RM 58,815.68 / yr

Total Annual Savings (Water + Power) = RM 338,175.68 / yr

The above procedure is repeated for every drop of 1.00 ^{0}C temperature of the cooling tower water until the approach becomes negative (colder than the wet bulb temperature). All the calculation values are tabulated in Table A4.2, A4.3, A4.4, A4.5, A4.6 and A4.7. The total water and energy savings are calculated for each drop of 1.00 ^{0}C of the cooling water temperature.

No.	Cooling Tower Heat Load (kJ/hr)	$T_{CW\ OUT}$ Condenser (^0C)	$T_{CW\ IN}$ Condenser (^0C)	Range (^0C)	Approach (^0C)
1	7948080	30.20	**28.30**	1.90	6.63
2	7961760	29.37	**27.30**	2.07	5.53
3	7961760	28.53	**26.30**	2.23	4.53
4	7961760	27.78	**25.30**	2.48	3.53
5	7961760	26.93	**24.30**	2.63	2.53
6	7961760	26.27	**23.30**	2.97	1.53
7	7961760	25.55	**22.30**	3.25	0.53
8	7961760	25.01	**21.30**	3.71	-0.37

Table A4.2: The Cooling Tower Heat Load, Temperatures, Range and Approach

No.	Water Flow (m^3/hr)	Air Flow (kg/hr)	L (GPM)	G (CFM)	L/G	Thermal Demand
1	996.00	592152.20	4385.26	284512.35	1.80	1.30
2	915.22	592152.20	4029.57	284512.35	1.65	1.35
3	848.65	592152.20	3736.51	284512.35	1.53	1.43
4	765.45	592152.20	3370.19	284512.35	1.38	1.52
5	721.08	592152.20	3174.82	284512.35	1.30	1.60
6	637.88	592152.20	2808.49	284512.35	1.15	1.70
7	582.41	592152.20	2564.27	284512.35	1.05	1.80
8	510.30	592152.20	2246.79	284512.35	0.92	1.90

Table A4.3: The Cooling Tower L/G Ratio and The Thermal Demand

No.	Q_H (kJ/s)	Q_C (kJ/s)	Chilled Water Flow (m^3/hr)	$T_{CH\ IN}$ (^0C)	$T_{CH\ OUT}$ (^0C)
1	2211.60	1746.00	653.16	12.00	9.71
2	2211.60	1769.28	653.16	12.00	9.68
3	2211.60	1792.56	653.16	12.00	9.65
4	2211.60	1815.84	653.16	12.00	9.62
5	2211.60	1839.12	653.16	12.00	9.59
6	2211.60	1862.40	653.16	12.00	9.56
7	2211.60	1885.68	653.16	12.00	9.53
8	2211.60	1908.96	653.16	12.00	9.49

Table A4.4: The Chiller Loads and The Chilled Water Temperatures

No.	E_1 Compressor (kW)	E_2 CT Fan (kW)	E_3 CT Pump (kW)	E_4 CW Pump (kW)	Total Electrical Power (kW)	Power Savings (RM/yr)
1	465.60	18.00	125.00	100.00	708.60	0.00
2	442.32	18.00	114.86	100.00	675.18	58,816.65
3	419.04	18.00	106.51	100.00	643.55	114,491.72
4	395.76	18.00	96.07	100.00	609.83	173,842.35
5	372.48	18.00	90.50	100.00	580.98	224,616.66
6	349.20	18.00	80.05	100.00	547.25	283,967.29
7	325.92	18.00	73.09	100.00	517.01	337,191.98
8	302.64	18.00	64.04	100.00	484.68	394,092.23

Table A4.5: The Power Consumption of The Cooling Tower System

No.	Blow Down (m^3/hr)	Drift Losses (m^3/hr)	Evaporation Losses (m^3/hr)	Make Up Water (m^3/hr)	Water Savings (RM/yr)
1	71.84	149.40	359.20	580.44	0.00
2	71.96	137.28	359.81	569.06	273,017.13
3	71.96	127.30	359.81	559.08	512,637.32
4	71.96	114.82	359.81	546.60	812,162.56
5	71.96	108.16	359.81	539.94	971,909.35
6	71.96	95.68	359.81	527.46	1,271,434.59
7	71.96	87.36	359.81	519.14	1,471,118.08
8	71.96	76.55	359.81	508.32	1,730,706.62

Table A4.6: The Water Consumption of The Cooling Tower System

No.	Total Annual Power and Water Savings (RM/yr)
1	0.00
2	331,833.78
3	627,129.04
4	986,004.91
5	1,196,526.01
6	1,555,401.88
7	1,808,310.06
8	2,124,798.85

Table A4.7: The Total Annual Power and Water Savings

APPENDIX 5

CALCULATION

CASE 3: COOLING TOWER SYSTEM (ONE CHILLER, ONE DISTILLATION CONDENSER AND ONE AIR CONDITIONER)

APPENDIX 5

CALCULATION

CASE 3: COOLING TOWER SYSTEM

(ONE CHILLER, ONE DISTILLATION CONDENSER & ONE AIR CONDITIONER)

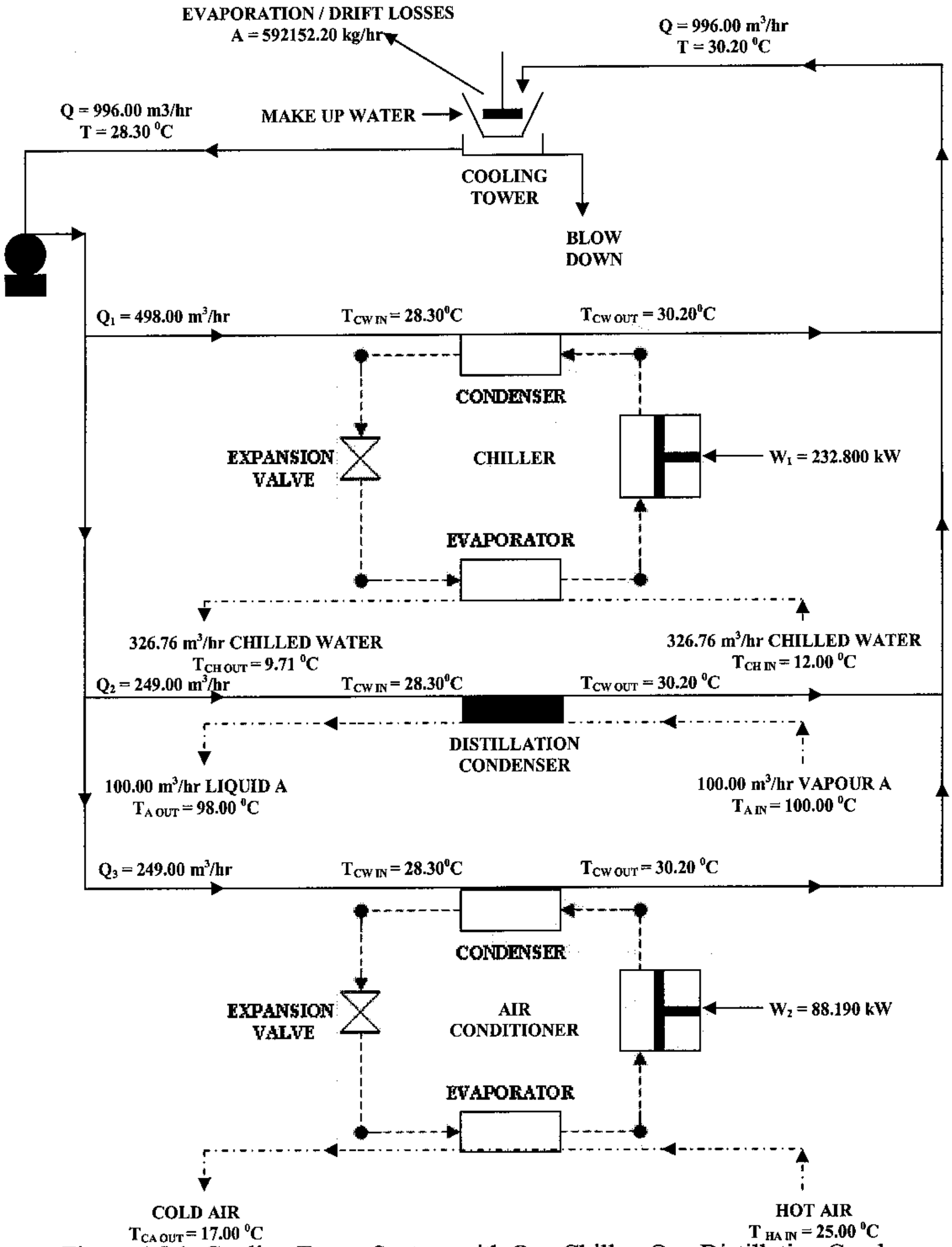

Figure A5.1: Cooling Tower System with One Chiller, One Distillation Condenser and One Air Conditioner

Assumptions

1. All assumptions and calculations are similar to Appendix 3 and Appendix 4.
2. The same model of the cross flow cooling tower and its performance curve (Appendix 1) been used to determine the thermal demand and the L/G ratio.
3. The chiller using 6.00 kg/s Refrigerant 134a and the air conditioner using 2.86 kg/s Refrigerant 22.
4. Based on the Table A4.3: The Cooling Tower L/G ratio and The Thermal Demand, the target temperature of the cooling water is 26.30 ^{0}C from 28.30 ^{0}C previously (set by the engineer) by improving the performance of the cooling tower system. The new cooling water circulation rate is 848.65 m^3/hr to achieve 2.00 ^{0}C reduction in temperature.
5. Based on this new temperature, the refrigeration compression work can be saved by 10.00 %. The product of the chiller (chilled water) and the product of the air conditioner (air) will be colder.
6. The cooling water have been redistributed throughout the entire cooling tower water network systematically by trial and error to achieve the new total cooling water circulation rate 848.65 m^3/hr.
7. The new water circulation rate and the new temperature parameters with improved performance (work reduction) are shown in the Figure A5.3.

Chiller

Q_H remains fixed = 1105.800 kJ/s

Q_C increases from 873.000 kJ/s to 896.280 kJ/s.

The compression work reduces from 232.800 kW to 209.520 kW (10 %).

The cooling water flow been modified from 498.00 m^3/hr to 424.33 m^3/hr.

The cooling water inlet temperature (condenser) reduces from 28.30 ^{0}C to 26.30 ^{0}C.

The cooling water outlet temperature (condenser) reduces from 30.20 ^{0}C to 28.06 ^{0}C.

The chilled water temperature reduces from 9.71 ^{0}C to 9.64 ^{0}C.

Condenser (For Distillate Product A)

The cooling water flow remains fixed to 249.00 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.30 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.20 ^{0}C.

The heat load:

$Q = m \times Cp \times (T_{CW\ OUT} - T_{CW\ IN})$

$Q = (249.00\ m^3 / 1\ hr) \times (1000\ kg / 1\ m^3) \times (4.2\ kJ / 1\ kg.^0C) \times (30.20\ ^0C - 28.30\ ^0C)$

$Q = 1987020.00$ kJ/hr

T_B = The boiling point for product A (can be pure or mixture)

$Q = m_A \times Cp_A$ (Vapor) $\times (100.00\ ^0C - T_B) + m_A \times H$ (Latent Heat) $+ m_A \times Cp_A$ (liquid) $\times (T_B - 98.00\ ^0C)$

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

$Q = m_A \times [Cp_A$ (Vapor) $\times (100.00\ ^0C - T_B) + H$ (Latent Heat) $+ Cp_A$ (liquid) $\times (T_B - 98.00\ ^0C)]$

$Q = m_A \times$ [Constant]

$Q = v_A \times$ Density $\times$ [Constant] $= v_A \times$ [Constant A]

Q is proportional to volumetric flow rate of A.

Initially the flow rate of product A, **$v_A = 100.00\ m^3/hr$**.

The cooling water inlet temperature (condenser) reduces from 28.30 ^0C to 26.30 ^0C.

The cooling water outlet temperature (condenser) remains fixed 30.20 ^0C.

$Q' = m \times Cp \times (T_{CW\ OUT} - T_{CW\ IN})$

$Q' = (249.00\ m^3 / 1\ hr) \times (1000\ kg / 1\ m^3) \times (4.2\ kJ /1\ kg.^0C) \times (30.20\ ^0C - 26.30\ ^0C)$

$Q' = 4078620.00$ kJ/hr

$v'_A = v_A \times Q' / Q$

$v'_A = (100.00\ m^3 / 1\ hr) \times (4078620\ kJ / 1\ hr) / (1987020\ kJ / 1\ hr)$

$v'_A = 205.26\ m^3/hr$

% Increase in production (area) = (205.26 – 100.00) / 100.00 x 100 % = **105.26 %** (maximum)

Air Conditioner

Q_H remain fixed = 551.650 kJ/s

Q_C increases from 463.460 kJ/s to 472.280 kJ/s.

The compression work reduces from 88.190 kW to 79.370 kW (10 %).

The cooling water flow been modified from 249.00 m^3/hr to 175.33 m^3/hr.

The cooling water inlet temperature (condenser) reduces from 28.30 ^0C to 26.30 ^0C.

The cooling water outlet temperature (condenser) reduces from 30.20 ^0C to 28.57 ^0C.

The air temperature reduces from 17.00 ^0C to 16.85 ^0C.

To calculate the inlet cooling water return to the cooling tower temperature:

Assume the reference temperature = T_{ref} = 0.00 ^{0}C

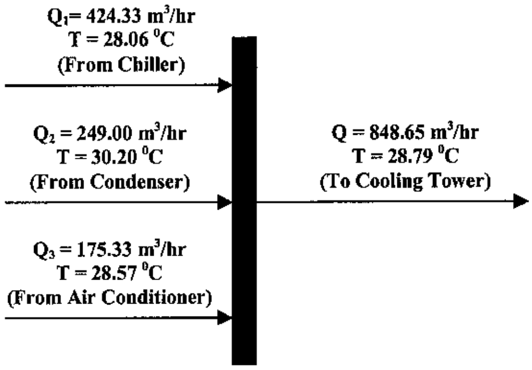

Figure A5.2: Cooling Tower Water Network Mass Balance

Q_1 x Density x Cp x (28.60 ^{0}C – T_{ref}) + Q_2 x Density x Cp x (30.20 ^{0}C – T_{ref}) + Q_3 x Density x Cp x (28.57 ^{0}C – T_{ref}) = Q x Density x Cp x (T – T_{ref})

T = 28.79 ^{0}C

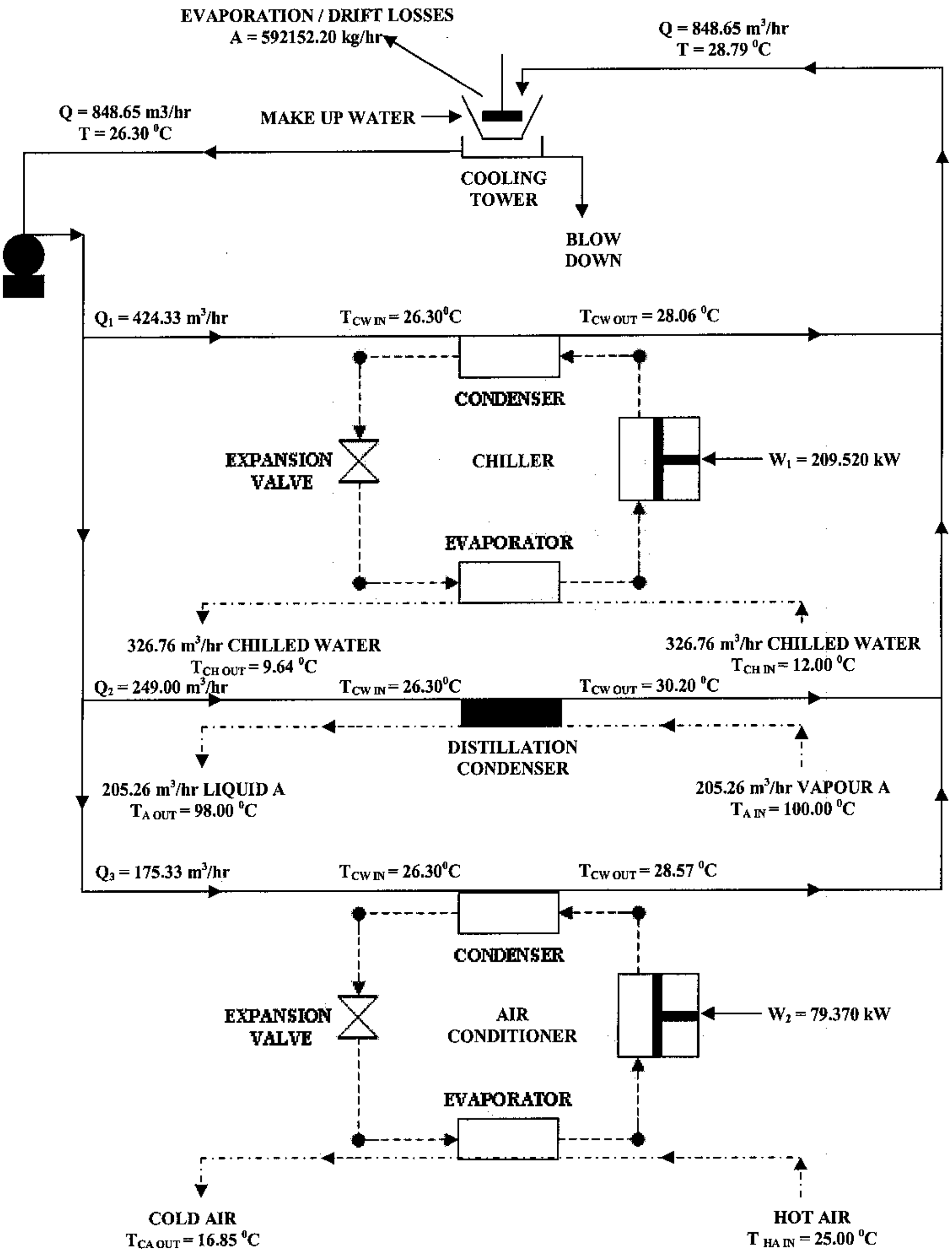

Figure A5.3: Cooling Tower System with One Chiller, One Distillation Condenser and One Air Conditioner after Redistribution of Cooling Water to Achieve 2.00 ^{0}C Reduction in Cooling Water Temperature

APPENDIX 6

CALCULATION
CASE 4: COOLING TOWER SYSTEM
(THREE CONDENSERS)

APPENDIX 6

CALCULATION

CASE 4: COOLING TOWER SYSTEM

(THREE CONDENSERS)

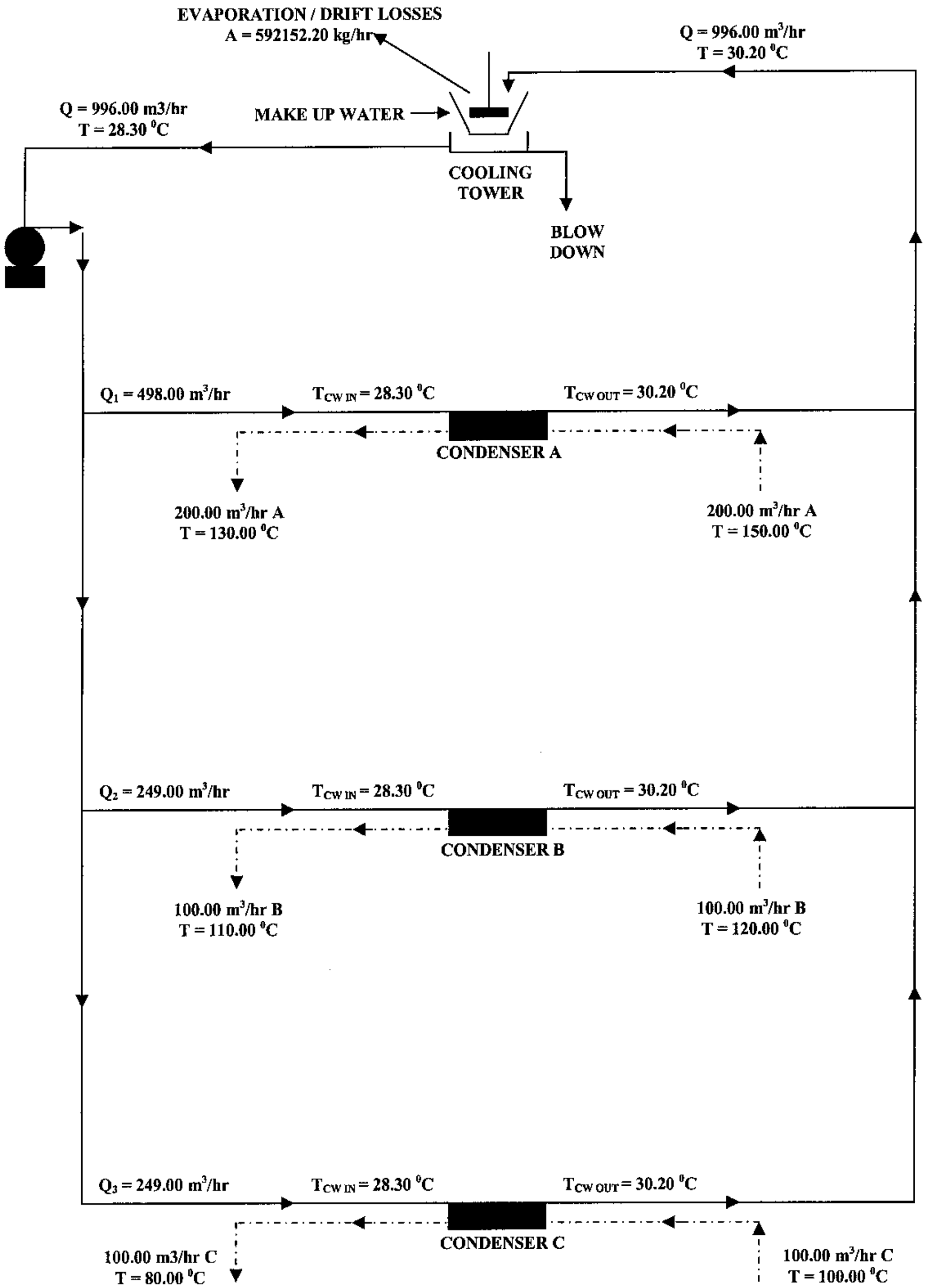

Figure A6.1: Cooling Tower System with Three Condensers

Assumptions

1. All assumptions and calculations are similar to Appendix 3, Appendix 4 and Appendix 5.
2. The same model of the cross flow cooling tower and its performance curve (Appendix 1) been used to determine the thermal demand and the L/G ratio.
3. Based on the Table A4.3: The Cooling Tower L/G ratio and The Thermal Demand, the target temperature of the cooling water is 26.30 ^{0}C from 28.30 ^{0}C previously (set by the engineer) by improving the performance of the cooling tower system. The new cooling water circulation rate is 848.65 m^3/hr to achieve 2.00 ^{0}C reduction in temperature.
4. The cooling water have been redistributed throughout the entire cooling tower water network systematically by trial and error to achieve the new total cooling water circulation rate 848.65 m^3/hr.
5. Lower cooling water temperature increases higher driving force creating more room for the product to be condensed in the condenser.
6. The condenser cooling water outlet temperature remains fixed at 30.20 ^{0}C.
7. The new water circulation rate and the new temperature parameters with improved performance are shown in the Figure A6.2.

Condenser A (For Distillate Product A)

The cooling water flow been modified from 498.00 m^3/hr to 448.88 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.30 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.20 ^{0}C.

The heat load

$Q = m \text{ x } Cp \text{ x } (T_{CW\,OUT} - T_{CW\,IN})$

$Q = (498.00\ m^3 / 1\ hr) \text{ x } (1000\ kg / 1\ m^3) \text{ x } (4.2\ kJ / 1\ kg.^0C) \text{ x } (30.20\ ^0C - 28.30\ ^0C)$

Q = 3974040.00 kJ/hr

T_B = The boiling point for product A (can be pure or mixture)

$Q = m_A \text{ x } Cp_A$ (Vapor) x $(150.00\ ^0C - T_B) + m_A$ x H (Latent Heat) + $m_A \text{ x } Cp_A$ (liquid) x $(T_B - 130.00\ ^0C)$

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

$Q = m_A \text{ x } [Cp_A$ (Vapor) x $(150.00\ ^0C - T_B)$ + H (Latent Heat) + Cp_A (liquid) x $(T_B - 130.00\ ^0C)]$

$Q = m_A$ x [Constant]

$Q = v_A$ x Density x [Constant] = v_A x [Constant A]

Q is proportional to volumetric flow rate of A.

Initially the flow rate of product A, $\mathbf{v_A = 200.00\ m^3/hr}$.

The cooling water inlet temperature (condenser) reduces from 28.30 ^{0}C to 26.30 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.20 ^{0}C.

$Q' = m$ x Cp x $(T_{CW\ OUT} - T_{CW\ IN})$

Q' = (448.88 m^3 / 1 hr) x (1000 kg/m^3) x (4.2 kJ / 1 kg.^{0}C) x (30.20 ^{0}C – 26.30 ^{0}C)

Q' = 7352654.40 kJ/hr

$v'_A = v_A$ x Q' / Q

v'_A = (200.00 m^3 / 1 hr) x (7352654.40 kJ / 1 hr) / (3974040 kJ / 1 hr)

$\mathbf{v'_A = 370.03\ m^3/hr}$

% Increase in production (area) = (370.03 – 200.00)/ 200.00 x 100.00 % = **85.02 %** (maximum)

Condenser B (For Distillate Product B)

The cooling water flow been modified from 249.00 m^3/hr to 199.88 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.30 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.20 ^{0}C.

The heat load

$Q = m$ x Cp x $(T_{CW\ OUT} - T_{CW\ IN})$

Q = (249.00 m^3 / 1 hr) x (1000 kg / 1 m^3) x (4.2 kJ / 1 kg.^{0}C) x (30.20 ^{0}C – 28.30 ^{0}C)

Q = 1987020.00 kJ/hr

T_B = The boiling point for product B (can be pure or mixture)

$Q = m_B$ x Cp_B (Vapor) x (120.00 ^{0}C – T_B) + m_B x H (Latent Heat) + m_B x Cp_B (liquid) x (T_B – 110.00 ^{0}C)

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

$Q = m_B$ x [Cp_B (Vapor) x (120.00 ^{0}C – T_B) + H (Latent Heat) + Cp_B (liquid) x (T_B – 110.00 ^{0}C)]

$Q = m_B$ x [Constant]

$Q = v_B$ x Density x [Constant] = v_B x [Constant B]

Q is proportional to volumetric flow rate of B.

Initially the flow rate of product B, $\mathbf{v_B = 100.00\ m^3/hr}$.

The cooling water inlet temperature (condenser) reduces from 28.30 ^{0}C to 26.30 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.20 ^{0}C.

$Q' = m \times Cp \times (T_{CW\,OUT} - T_{CW\,IN})$

$Q' = (199.88\ m^3 / 1\ hr) \times (1000\ kg / 1\ m^3) \times (4.2\ kJ / 1\ kg.^{0}C) \times (30.20\ ^{0}C - 26.30\ ^{0}C)$

$Q' = 3274034.40$ kJ/hr

$v'_B = v_B \times Q' / Q$

$v'_B = (100.00\ m^3 / 1\ hr) \times (3274034.40\ kJ / 1\ hr) / (1987020\ kJ / 1\ hr)$

$v'_B = 164.77\ m^3/hr$

% Increase in production (area) = (164.77 – 100.00)/ 100.00 x 100.00 % = **64.77 %** (maximum)

Condenser C (For Distillate Product C)

The cooling water flow been modified from 249.00 m^3/hr to 199.88 m^3/hr.

Initially the cooling water inlet temperature (condenser) = 28.30 ^{0}C.

Initially the cooling water outlet temperature (condenser) = 30.20 ^{0}C.

The heat load

$Q = m \times Cp \times (T_{CW\,OUT} - T_{CW\,IN})$

$Q = (249.00\ m^3 / 1\ hr) \times (1000\ kg / 1\ m^3) \times (4.2\ kJ / 1\ kg.^{0}C) \times (30.20\ ^{0}C - 28.30\ ^{0}C)$

$Q = 1987020.00$ kJ/hr

T_B = The boiling point for product C (can be pure or mixture)

$Q = m_C \times Cp_C$ (Vapor) $\times (100.00\ ^{0}C - T_B) + m_C \times H$ (Latent Heat) $+ m_C \times Cp_C$ (liquid) $\times (T_B - 80.00\ ^{0}C)$

The process parameters (temperature) cannot be changed because it affects the quality of the product. The parameters remain fixed.

$Q = m_C \times [Cp_C$ (Vapor) $\times (100.00\ ^{0}C - T_B) + H$ (Latent Heat) $+ Cp_C$ (liquid) $\times (T_B - 80.00\ ^{0}C)]$

$Q = m_C \times$ [Constant]

$Q = v_C \times$ Density $\times$ [Constant] $= v_C \times$ [Constant C]

Q is proportional to volumetric flow rate of C.

Initially the flow rate of product C, **$v_C = 100.00\ m^3/hr$**.

The cooling water inlet temperature (condenser) reduces from 28.30 ^{0}C to 26.30 ^{0}C.

The cooling water outlet temperature (condenser) remains fixed 30.20 ^{0}C.

$Q' = m \times Cp \times (T_{CW\,OUT} - T_{CW\,IN})$

$Q' = (199.88\ m^3 / 1\ hr) \times (1000\ kg / 1\ m^3) \times (4.2\ kJ / 1\ kg.^0C) \times (30.20\ ^0C - 26.30\ ^0C)$

$Q' = 3274034.40$ kJ/hr

$v'_C = v_C \times Q' / Q$

$v'_C = (100.00\ m^3 / 1\ hr) \times (3274034.40\ kJ / 1\ hr) / (1987020\ kJ / 1\ hr)$

$v'_C = 164.77\ m^3/hr$

% Increase in production (area) = (164.77 – 100.00)/ 100.00 x 100.00 % = **64.77 %** (maximum)

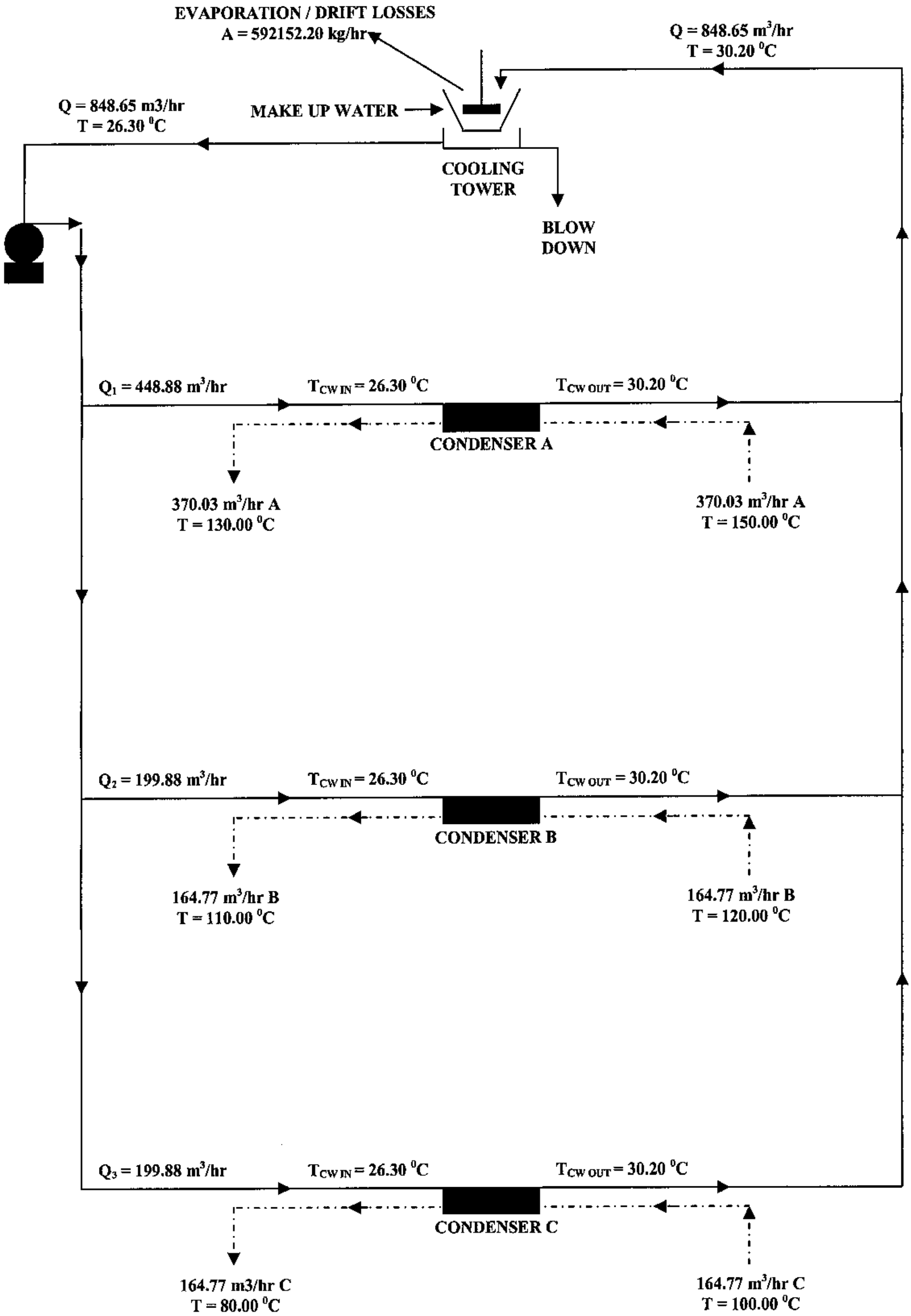

Figure A6.2: Cooling Tower System with Three Condensers after Redistribution of Cooling Water to Achieve 2.00 ^{0}C Reduction in Cooling Water Temperature