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Design of Industrial Water Cooled Chiller for Recycle Cyclohexane in Polyethylene Plant

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Abstract

The research was aimed at providing water for cooling recycled cyclohexane in polyethylene plant to 15˚C. This research could be redounding to the benefit of polyethylene plants that are using solution based polymerization technique. A chiller unit of 630 T, which has compressor power input of 378.33 kW and can provide chilled water capable of cooling recycle cyclohexane to 15˚C, was designed using Aspen Hysys version 7.1. Four different refrigerants were tested to know the best fit refrigerant for the design and the best among them was R134a. The designed chiller has a coefficient of performance of 6.3 and a capacity greater than that of the defective chiller (550 TR). Unlike the defective chiller, with the increased cooling capacity and corresponding increase in compressor power from 296 kW to 378.6 kW, it could discharge chilled water capable of cooling recycle cyclohexane from 38˚C to 15˚C without tripping of the unit. With this design, ethylene absorption rate could increase to 40 T/h. Evaporator and condenser were designed and duties were 2166.39 KJ/sec and 2544.72 KJ/sec respectively. A thermostatic expansion valve with flow coefficient of 46.17 gpm was designed. The designed suction and discharge pressure were 414 kpa and 1053 kpa respectively while condenser temperature of 40˚C was used for the design. The cooling of recycle cyclohexane from 38˚C to 15˚C using the chilled water supplied by the designed chiller was simulated using Aspen Hysys.

Keywords

Chiller, Compressor, Evaporator, Condenser, Expansion Valve

1. Introduction

Polyethylene production can be achieved using solution based polymerization technology [1], slurry method [2] or gas phase technology [3]. In solution based method, a solvent is used to absorb ethylene and this solvent does not take part in reaction but is however recovered and recycled in other to save cost. Example of such solvent is cyclohexane.

Ethylene absorption is highly exothermic and the temperature of the recycle cyclohexane affects the equilibrium absorption temperature [1]. In Eleme Petrochemicals, recycle cyclohexane at 7 Kg/cm^2 is cooled from 180˚C to about 35˚C using series of heat exchangers. Some of the heat exchangers use cooling water supplied at 32˚C. In a quest to further cool the cyclohexane so as to improve ethylene absorption, a chiller was introduced and this only helped to drop the recycle cyclohexane temperature to 20˚C. Efforts to cool further to 17˚C resulted in tripping of the chiller and the unit eventually broke down.

The aim of this research was to design a chiller capable of supplying water for cooling the recycle cyclohexane to 15˚C. A chiller is one of the heat transfer equipment and it employs the principles of vapour compression cycle.

The objectives of the project include: simulation of the cooling of cyclohexane from 38˚C to 15˚C, using Hysys, to get the chiller water requirements, test with different refrigerants to get the best fit refrigerant for the design, development of algorithm for the design and design of the proposed chiller based on the optimum chiller water parameter requirements and best fit refrigerant.

2. Materials and Methods

2.1. Materials

The materials used for this report includes P-h diagram for R134a, R125, R152a & R1270 respectively; ASHRAE handbooks, Handbook of Air Conditioning & Refrigeration; Process Technology manual, software (Aspen Hysys, version 7.1) and input data. The input data, which were gotten from the Plant, includes cyclohexane flow rate, temperature & pressure; cooling water pressure and temperature.

2.2. Methods

To achieve the aim of this research, Aspen Hysys software, version 7.1, which used the principles of mass & energy conservations, principles of heat transfer and Thermodynamics to generate results, was used. A design tolerance of 10% was used in this work and a steady state simulation and design procedure was applied. The following assumptions were made:

1) Potential and kinetic energy effect were negligible
2) Evaporator and condenser were at constant pressure
3) Evaporator outlet was saturated vapour while condenser outlet was saturated liquid
4) The vapour compression cycle was at steady state.
5) Adiabatic process occurs at the expansion valve
6) The chiller was considered to be an ideal vapour compression refrigeration cycle and thus, the compression process was isentropic.

Figure 1 is the chiller loop with cyclohexane cooling loop. The path "abcd" indicates the refrigeration loop. At evaporator, water used in cyclohexane cooling loss heat to the refrigerant and return to the plate heat exchanger as chilled water at a lower temperature for cooling the recycled cyclohexane while the refrigerant gain heat and vaporized. The compressor lifts the refrigerant to the condenser with increase in energy. At the condenser, the heat removed from the water at evaporator is rejected and the condensed refrigerant is sent back to the evaporator through metering device to complete the vapour compression cycle.

2.2.1. Chiller Water Process Parameter Requirement

The process of cooling 200 T/hr of cyclohexane from 38˚C to 15˚C was simulated using Aspen Hysys to determine the volume of water and the temperature of water going to chiller as shown in **Figure 1**. The sensible heat of the cooling process was estimated as follows [4]:

$$Q_s = m_s Cp_s \left(t_{so} - t_{s1}\right) \tag{1}$$

where, Q_s is the rate of heat loss by the hot cyclohexane, m_s is the mass flow rate of cyclohexane, Cp_s is the specific heat capacity of cyclohexane at mean temperature, t_{so} is the temperature of cyclohexane after cooling and t_{si} is the inlet temperature of cyclohexane.

In other to estimate the water requirement, energy balance on the plate heat exchanger was done:

$$Q_s = m_w Cp_w \left(t_{wo} - t_{wi}\right) \tag{2}$$

where, t_{wo} is the temperature of the water returned to the chiller evaporator after cyclohexane cooling, which is to be determined (as shown in **Figure 1**) and t_{wi} is temperature of the chilled water with a value not greater than 15˚C.

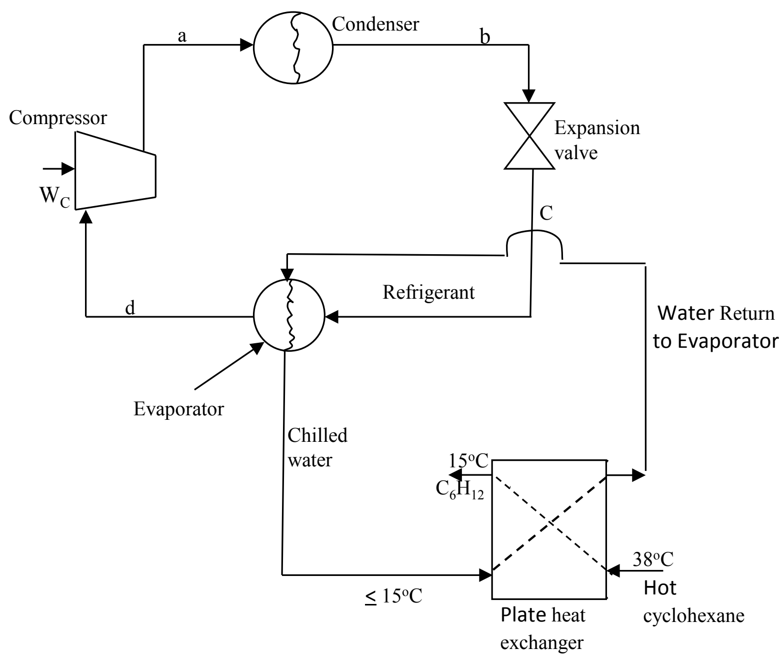

Figure 1. Chiller and cyclohexane cooling loop.

2.2.2. Selection of Refrigerant

Simulation of the vapour compression cycle and cyclohexane cooling were done at evaporator temperature of 5˚C, 8˚C, 10˚C, 12˚C, & 13˚C to respectively to determine the refrigerant that will require least energy input in the compressor. The refrigerants used were 1,1,1,2-tetraflouroethane (R134a), 1,1,1,2,2- pentaflouroethane (R125), propylene (R1270) & 1,1-diflouroethane (R152a). The volume of the refrigerant required, compressor power requirement and temperature of water returned to evaporator were noted.

2.2.3. Chiller Design

Using the cyclohexane cooling simulation results, the cooling load of the chiller was calculated using Equation (3) [5].

$$Q_{evap} = V\rho Cp_w\left(T_{in} - T_{out}\right) \tag{3}$$

The cooling load was converted to chiller capacity in refrigeration tons (TR) using a conversion factor of 1 TR is 3.5 kW [6].

The ideal P-h diagram for the chiller is shown in **Figure 2**. It was used to determine the thermodynamic properties of the refrigerant at stage 1 (compressor suction), stage 2 (compressor discharge), stage 3 (condenser outlet) and stage 4 (expansion valve outlet/ evaporator inlet.

1) Compressor Design

Since the compression process is isentropic, the discharge temperature was calculated using the temperature variation between stages in isentropic process.

$$T_d = T_S\left[r^{\left(\frac{\gamma-1}{\gamma}\right)}\right] \tag{4}$$

In this work, the discharge temperature does not exceed 107˚C because the discharge temperature should be less than the critical temperature of refrigerant [7].

Using **Figure 2**, the compressor energy input could be determined as follows

$$W_c = h_2 - h_1 \tag{5}$$

From the definition of enthalpy, Equation (5) could be transformed as [8]

$$W_c = \dot{m}C_{\Pr}\left(T_2 - T_1\right) \tag{6}$$

To determine the isentropic efficiency of the designed compressor, isentropic head was estimated as [9]

$$W_{isen} = Cp_R\left(T_2^{rev} - T_1\right) \tag{7}$$

To get T^{rev}, the compression was assumed reversible and the entropy of isentropic process was zero [4]

$$\Delta S = Cp\ln\left(\frac{T_2^{rev}}{T_1}\right) - R\ln\left(P_2/P_1\right) = 0 \tag{8}$$

The isentropic efficiency of the compressor was calculated using Equation (9) while the compressor power was estimated using Equation (10) [10].

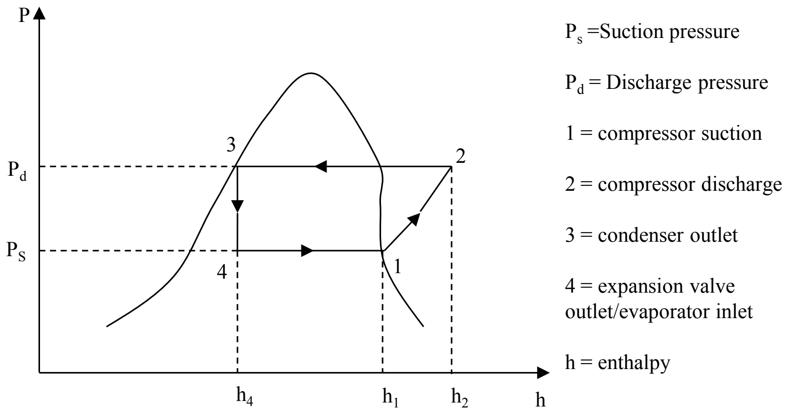

Figure 2. Ideal P-h diagram for the chiller.

$$\eta_{isen} = \frac{W_{isen}}{W_s} \tag{9}$$

$$\text{Power} = \dot{m} C_{\text{Pr}} \left(T_2 - T_1 \right) \tag{10}$$

In this study, an isentropic efficiency range of 70% - 85% was used [9].

2) Evaporator and Condenser Design

The chiller was designed using a condenser temperature of 40˚C and evaporating temperature of 10˚C. The type of evaporator designed was flooded water-cooled shell and tube heat exchanger [11]. The tube side fluid was water while the refrigerant, R134a, was the shell side fluid. The condenser was shell and tube.

Taking energy balance on the evaporator, the heat transfer rate can be estimated as shown on **Figure 2**.

$$Q_{evap} = m\left(h_1 - h_4 \right) \tag{11}$$

The area of evaporator required for heat transfer was estimated using Equation (12) while the number of tubes was estimated using Equation (14) [12].

$$Q_{evap} = UA\Delta T \tag{12}$$

The range of values of "U" in Equation (12) in this study was 1080 - 3600 kJ/h m^2·C (300 - 1000 W/m^2·K) for evaporator [13] and 3600 - 14,400 kJ/h m^2·C (1000 - 4000 W/m^2·K) for condenser [11].

The log mean temperature difference (LMTD) was evaluated as shown below [11].

$$\Delta T = \frac{TD_1 - TD_2}{\ln\left(\frac{TD_1}{TD_2} \right)} \tag{13}$$

$$N_t = \frac{A}{\pi d_o^2} \tag{14}$$

Using **Figure 2**, the heat transferred in the condenser can be established by taking energy balance at the condenser [14].

$$Q_{cond} = m(h_2 - h_3) \tag{15}$$

3) Metering Device

At the valve, the refrigerant exists as saturated liquid (f) and saturated vapour (g) due to the flashing action. The dryness fraction of the refrigerant was denoted by "X". To get the dryness fraction of the refrigerant at the valve outlet, the enthalpy of the refrigerant was obtained as [7]

$$h_4 = h_f + x_4(h_g - h_f) \tag{16}$$

where, h_f and h_g were gotten from chart (16)

From Figure 2, the coefficient of performance of the chiller can be determined as:

$$COP = \frac{h_1 - h_4}{h_2 - h_1} \tag{17}$$

For this study, the value of COP was considered to be greater than 5.7 since the Carnot COP of the chiller was 5.7.

2.3. Solution Technique

The chiller was designed using Aspen Hysys simulator. The following algorithm was developed as solution techniques (Figure 3).

3. Results and Discussion

3.1. Cyclohexane Cooling Simulation Results

The results of the simulation of the cooling of the cyclohexane are as shown in Table 1. The result suggests that the temperature of water returned to the chiller is 31.65˚C. To cool 200 T/h cyclohexane from 38˚C to 15˚C using 120 m^3/h of water, the water temperature is 15˚C. This is because the temperature approach of the heat exchanger is 0˚C. A water flow rate of 120 m^3/hr was used because the maximum rating of the pump supplying water to the chiller is 135 m^3/hr.

3.2. Choice of Refrigerant

3.2.1. Effect of Evaporator Temperature on Compressor Power

Figure 4 shows how the compressor power input vary with evaporator temperature for all the refrigerants used. It indicates that R134a was the best fit refrigerant for the design since it had the least compressor power input at each respective evaporator temperature.

Increased in evaporator temperature from 5˚C through 13˚C reduced the compressor power input for all the refrigerants due to drop in compression ratio which reduced the compressor work as shown in Figure 4.

This effect was maximum when R134a was used; thus, R134a is best fit for the design.

Evaporating temperature of less than 5˚C was not considered to avoid freezing of the chilled water inside the tubes of evaporator; while evaporating temperature greater than 13˚C was not considered to avoid temperature cross.

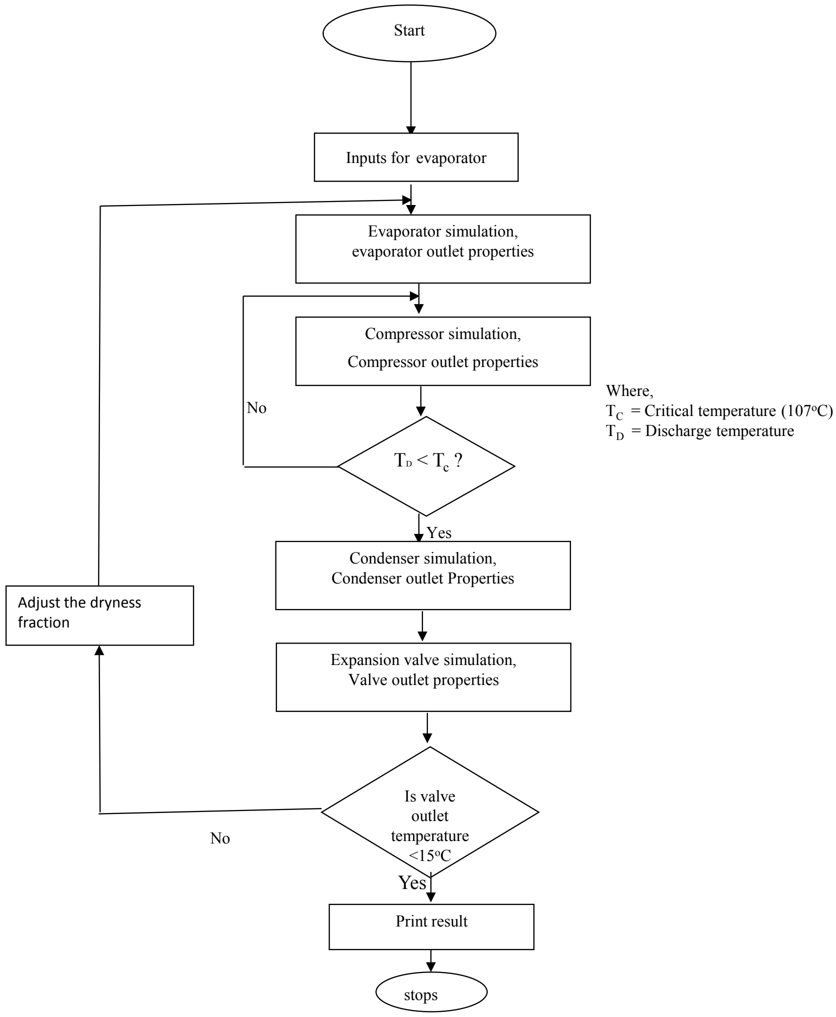

Figure 3. Algorithm of the chiller design.

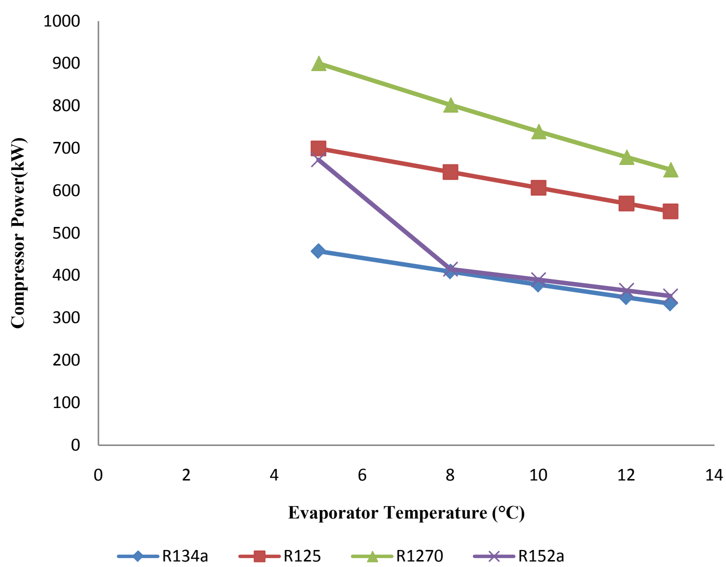

Figure 4. Effects of evaporator temperature on compressor power input for different refrigerants.

Table 1. Process requirement of chiller water.

Parameter	Existing	Simulation
Water Flow-rate (m^3/h)	103	120.00
Chilled-water temperature (˚C)	20.00	15.00
Water returned temperature (˚C)	27.30	31.65
Cyclohexane flow (T/h)	200.00	200.00
Cyclohexane temperature (˚C)	35.00	38.00

3.2.2. Effect of Compressor Duty on Coefficient of Performance

Figure 5 shows the effect of compressor duty on coefficient of performance for the different refrigerants used. The best fit refrigerant for the design was 1,1,1,2-tetrafluoraethane (R134a) since it required the least compressor power as shown on Figure 5. A chiller with less power input had high coefficient of performance and is cost effective [6]. R134a required a compressor power of 378.1 KW at 10˚C while R152a required 390 kW compressor power at 10˚C since the discharge temperature was higher when R152a was used.

Figure 5 also show that the coefficient of performance of a chiller increased with decreased in compressor power because coefficient of performance is the ratio of quantity of energy absorbed in evaporator to the energy input of the refrigeration cycle. R134a has the highest coefficient of performance of 6.49 since it had the least compressor input of 333.6 kW as shown in Figure 5.

3.3. Chiller Design Results

The summary of the results of chiller designed is shown in Table 2. The capacity of the designed chiller is 630 Refrigeration Tons. 7,799,004 kJ/h of heat could be removed when the chiller water was cooled from 31.65˚C to 15˚C at the evaporator using R134a. Electrical power of 379 kW is needed by the compressor to expand the 42.2 m^3/h R134a from 10˚C to a designed temperature of 49.53˚C.

The type of compressor is centrifugal because of the high volume of refrigerant that is needed in the vapour compression cycle.

3.3.1. Compressor Design Results

Table 3 shows the compressor sizing results gotten from Hysys. The power input of the designed compressor was 378.6 kw while it had adiabatic efficiency of 75%. The compressor had an adiabatic head of 1995 m. This high power consumption was as a result of increase in evaporator duty and refrigerant flow. The maximum discharge temperature of the designed compressor was satisfactory as it was less than the critical temperature of refrigerant which is 107˚C. The compressor was single stage because the compressor ratio was less than 4.0 [11].

3.3.2 Evaporator and Condenser Design Results

Table 4 shows that a design heat duty of 7,799,004 kJ/hr was required in the evaporator of this vapour compression cycle. The evaporator design temperature was 10˚C. This was because for a flooded water-cooled evaporator, the temperature approach, which is the difference between the evaporating temperature and the chilled water outlet temperature, should be 3.3˚C - 5.6˚C [6]. The tube side outlet temperature was 15˚C because the design was made at a worse case scenario of a temperature approach of the plate cooler, where the cooling of cyclohexane was done, at 0˚C.

The area of heat transfer was 288 m^2 as compared to that of defective chiller which is 244 m^2 (from vendor diagram) and the evaporator had 764 tubes. This area increase was as a result of the rise in heat duty.

Table 5 suggests that the condenser duty required for the design was 9,167,040 kJ/hr and a condensing temperature of 40˚C was used because as a rule of thumb, the condenser temperature is 10˚C - 15˚C above ambient temperature [15].

The overall heat transfer coefficient of the condenser was 3796 kJ/h-m^2 C and the area of heat transfer was 332.5 m^2 as compared to that of the defective chiller that was 319 m^2. The total number of tubes of the designed condenser was 882.

To avert the effect of corrosion, carbon steel was used for shell design.

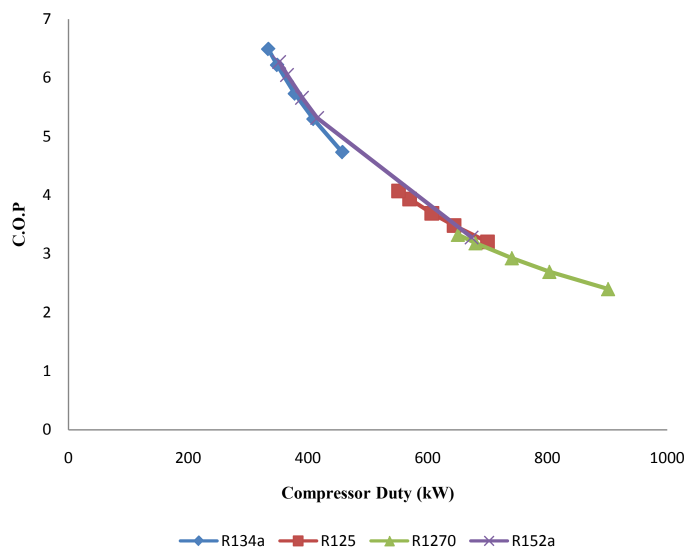

Figure 5. Effect of compressor duty on coefficient of performance for different refrigerants.

Table 2. New chiller specifications.

Designed Parameters and their Units	Values
Chiller Capacity (TR)	630
Cooling Load (kJ/h)	7,799,004
Power (kW)	378
Refrigerant flow (m^3/h)	42.2
Suction Temperature (˚C)	10
Discharge Temperature (˚C)	49.53
Refrigerant	R134a
Type	Centrifugal

Table 3. Compressor sizing results.

Parameter	Value
Power consumed (kW)	378.60
Adiabatic efficiency (%)	75.00
Adiabatic head (m)	1995.00
Suction pressure (kPa)	414.20
Discharge pressure (kPa)	1053.00
Flow rate (m^3/h)	42.070
Suction temperature (˚C)	10.00
Discharge temperature (˚C)	49.53
Compressor type/stages	Centrifugal/single stage

Table 4. Evaporator sizing results.

Parameter	Value
Evaporator duty (kJ/h)	7,799,004
Overall coefficient of heat	2495.00
Transfer (kJ/h-m^2-C)	
Area of heat transfer (m^2)	288.00
Number of tubes	764.00
Tube internal diameter (mm)	16.00
Tube external diameter (mm)	20.00
Tube length (m)	6.00
Tube thickness (mm)	2.00
Number of shell passes	1
Number of tube passes	2
Shell diameter (mm)	774 mm
Tube layout angle	Triangular (30 degrees)
Tube pitch (mm)	25.00
Baffle spacing (mm)	309.00
Tube side inlet temperature (˚C)	31.48˚C
Tube side outlet temperature (˚C)	15.00˚C
Tube side pressure (kPa)	350.00
Shell side pressure (kPa)	421.2
Water flow rate (m^3/h)	120.0
Refrigerant flow rate (m^3/h)	42.1
Material for tube construction	Copper

Table 5. Condenser sizing results.

Parameter	Value
Condenser duty (kJ/h)	9,167,040
Overall coefficient foe hat transfer (kJ/h-m^2-C)	3796.00
Area of heat transfer (m^2)	332.50
Number of tubes	882
Tube internal diameter (mm)	16.00
Tube external diameter (mm)	20.00
Tube material	Copper
Number of shell passes	1
Number of tube passes	2
Shell diameter (mm)	820
Material for shell construction	Carbon steel
Tube layout angle	Triangular (30 degrees)
Tube pitch (mm)	50.00
Baffle spacing (mm)	331.00
Tube length (m)	6
Shell side pressure (kPa)	1053
Tube side pressure (kPa)	350
Shell side inlet temperature (˚C)	49.53
Shell side outlet temperature (˚C)	40.00
Cooling water inlet temperature (˚C)	30.00
Cooling water outlet temperature (˚C)	36.51
Cooling water flow (m^3/h)	327.2

3.3.3. Expansion Valve Design Results

Table 6 shows the expansion valve sizing results. For this design, thermostatic expansion valve was considered. The valve has a flow coefficient (C_v) of 46.17 gpm at 50% opening.

The flow rate through the valve was 5.22×10^4 kg/h while the pressure drop of the valve was 597.1 kPa.

3.4. Comparison of the Propose Designed Chiller with the Existing Chiller

Table 7 gives a comparison of the proposed designed chiller with the old defective one. To cool recycle cyclohexane to 15˚C using the proposed designed chiller, the chiller capacity increased from 550 TR to 630 TR and compressor duty increases from 1,065,600 kJ/h to 1,361,988 kJ/h due to the drop in chiller water temperature which increased the evaporator duty.

Table 6. Valve sizing results.

Valve sizing parameter and units	Value
Inlet pressure (kPa)	1018.00
Valve opening (%)	50.00
ΔP (kPa)	597.1
Flow rate (kg/h)	5.224×10^{-4}
C_v (gpm)	46.17
Vapour fraction of refrigerant	0.2265

Table 7. Comparison of propose designed new chiller with the present chiller.

Element/Parameters	Present Chiller	Propose Designed chiller
Chiller Capacity (Tons)	550.00	630.00
Compressor Duty (kJ/hr)	1,065,600.00	1,361,988.00
Evaporator duty (kJ/hr)	5,245,200.00	7,799,004.00
Condenser duty (kJ/h)	6,310,800.00	9,167,040.00
Refrigerant flow (m^3/h)	33.00	42.2
Suction temperature (˚C)	10.00	10.00
Discharge temperature (˚C)	29.26	49.53
Chilled water flow (m^3/h)	103	120
Condenser cooling water flow (m^3/h)	257	372
Valve Coefficient (gpm)	26.7	46.17

The discharge temperature of the designed chiller increased from 29.26˚C to 49.53˚C due to rise in compressor work that occurred as a result of increase in refrigerant flow rate from 33 m^3/h to 42.2 m^3/h. Similarly, the condenser cooling water flow increased from 257 to 372 m^3/h because of the increased in the rate of heat rejection in the condenser from 6,310,800 kJ/h to 9,197,040 kJ/h.

The compressor power input in the new designed chiller was greater than that of the old chiller because of increased in compression ratio which was as a result of the rise in discharge temperature from 29.3˚C to 49˚C.

3.5. Effects of the Proposed Designed Chiller on Cyclohexane Temperature

Figure 6 shows the temperature shows the temperature profile of cyclohexane as flow of cyclohexane increased when these scenarios were considered; before chiller fault, when defective chiller was bypassed and when the proposed chiller would be in use.

The presence of the chiller caused a temperature drop of up to 23˚C in the cyclohexane used in ethylene absorption when cyclohexane flow was raised to 200 T/h. Cyclohexane was discharged to the ethylene absorption unit at 15˚C. This was because greater heat energy was removed from the cyclohexane by chilled water, which came from the chiller at a lower temperature of 15˚C.

Figure 6 suggests that when proposed chiller is used, it could give the least outlet temperature of cyclohexane.

3.6. Effect of Chiller Capacity

Figure 7 shows the effect of chiller capacity on ethylene absorption rate. The ethylene absorption rate increased from 20 T/h to 35 T/h when the chiller of 550 TR was used. By interpolation, it could be predicted that with the proposed chiller of 630 TR capacity, ethylene absorption rate would increase to 40 T/h as shown in Figure 7. This is because with 630 TR capacity, the cyclohexane would enter the absorption unit at 15˚C and this would reduce the equilibrium absorption temperature of ethylene [1].

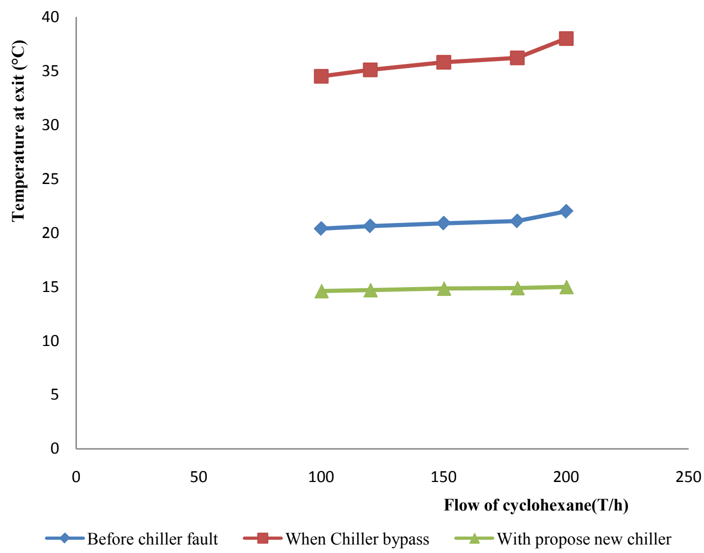

Figure 6. Temperature profile of cyclohexane before chiller fault, when chiller was by-passed, and when proposed chiller would be used.

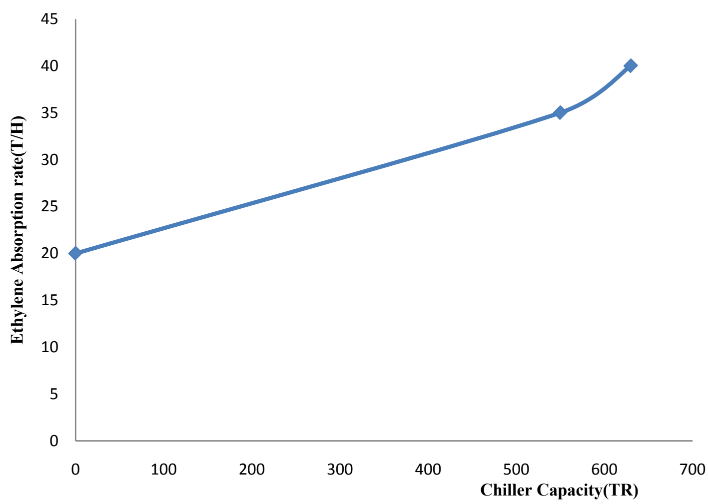

Figure 7. Effect of increase in chiller capacity.

4. Conclusions

A centrifugal water-cooled chiller of 630 tons of refrigeration was designed. The chilled water designed flow rate was 120 m^3/h at 15˚C. This chiller is capable of cooling 200 T/h of cyclohexane at 7 Kg/cm^2 from 38˚C to 15˚C at the same pressure.

Out of the four refrigerants (R134a, R125, R152a and R1270) used, R134a was the best fit refrigerant for the design. R134a had the least compressor power input requirement at 10˚C evaporating temperature and thus, gave a chiller with the highest coefficient of performance. R152a had a low compressor input but it is not suitable for evaporation temperature less than 8˚C; a blend of the R134a and R152a could be used in further studies to see the cooling effect.

The designed chiller had a coefficient of performance of 6.3 and a capacity greater than that of the defective chiller which was 550 TR. With the increased cooling capacity and corresponding increased in compressor power from 296 kW to 378.6 kW, it could discharge chilled water capable of cooling recycle cyclohexane from 38˚C to 15˚C without tripping of the unit. Optimization of the designed chiller could be done to further reduce the energy consumption in subsequent studies.

The 630 TR chiller needed a single stage centrifugal compressor with power input of 378.33 kW. The design suction pressure was 414.2 kPa, and discharge pressure was 1053 kPa, and temperature lift of the chiller was 39.53˚C, while the suction temperature was 10˚C. 42.2 m^3/h of R134a at 10˚C was needed to absorb 7,799,004 kJ/hr heat from 120 m^3/h chilled water at 31.48˚C. Similarly, 327.2 m^3/h of cooling water at 30˚C was needed to eject 9,167,040 kJ/hr heat at condenser with a condensing temperature of 40˚C. The maximum condenser water returned temperature was 36˚C. The heat transfer area requirement for the evaporator was 288 m^2 while that of condenser was 332.5 m^2 (as compared to that of the present defective chiller which is 244 m^2 and 319 m^2 respectively).

Algorithm was developed for the design procedure and this algorithm was self explanatory and easy to work with.

Finally, although the designed chiller could have cost implications, the chiller would increase the rate of ethylene absorption to 40 T/h thereby leading to increase in plant through-put and revenue. Further study can be done to cool the chilled water further and study the effects on the cyclohexane cooling and ethylene absorption rate.

Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

Nomenclature

Symbols/abbreviation	Meaning
h_f	Enthalpy of R134a in liquid phase

Continued

TR	Tons of refrigeration
h_g	Enthalpy of Refrigerant in gaseous phase
R134a	1,1,1,2-tetraflouroethane
R1270	Propylene
R125	1,1,12,2-pentaflouroethane
R152a	1,1-diflouroethane
T_d	Discharge Temperature
R	Compression ratio
γ	Ratio of heat capacities
T_s	Suction temperature
P	Pressure
H	Enthalpy
U	Internal energy
V	Volume
Rev	Reversible
η	Efficiency
W_c	Compressor work
W_s	Real work
$\dot{m}$	Mass flow rate
Δ	Change
C_p	Specific heat capacity t constant pressure
C_v	Specific heat capacity t constant volume
R	Universal gas constant
Isen	Isentropic
M_R	Mass flow of refrigerant
Q_{evap}	Evaporator duty
Q_{cond}	Condenser duty
S	Enthropy
TXV	Thermal Expansion valve
U	Coefficient heat transfer
A	Area
COP	Coefficient of performance
T/h	Tones per hour
X	Quality of refrigerant
Cv	Valve flow Coefficient
N_t	Number of tubes
1, 2, 3, 4	States on P-h diagram
ρ	Density
C_6H_{12}	Cyclohexane

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