



(RESEARCH ARTICLE)



## Optimization of water-cooled chiller in polyethylene plant

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### Abstract

This research was carried out to reduce the energy associated with process of cooling solvents for polymerization in Polyethylene (PE) plant. Cooling of cyclohexane to 10°C in a PE plant increased power consumption to 413.3kW and the optimization reduced the power compressor power input by 32.1 percent. Optimization was carried out with the use of Aspen HYSYS version 11.0, where Peng Robinson was used as the fluid package. To optimize, double stage centrifugal compressor was used. Optimum compression ratio of 1.73 and a corresponding intermediate pressure of 800kPa were obtained. A case study was made using variable First stage discharge pressure to determine compression ratio at each stage. A plot of compression ratio against first stage discharge pressure for each stage was used to determine the optimum compressor ratio and intermediate pressure. Another case study to determine the total compressor input at variable compression ratio was done and a plot of total compression power input against the compression ratio at each stage revealed that the total power input of the chiller was 280.5kW. The optimized chiller has an actual coefficient of performance (COP) of 6.24 and a theoretical COP of 8.64 while the efficiency of the chiller was 72.2%. The optimized chiller reduced the energy cost per day from 257.9USD to 154.8USD. The heat duty of the evaporator and condenser were 8377000kJ/h and 9865000kJ/h respectively while the flow coefficient of the J-T valve was 51.57gal/min. The power consumption of the first stage was 121.8kW while the power consumption at the second stage was 158.7kW. The compressor discharge was 1300kPa and 56.89°C. R134a was used as the refrigerant.

**Keywords:** Optimization; Chiller; Polyethylene; Plant

### 1 Introduction

In Polyethylene and Chemical plants, raw material preparation is a very important step in production. The process involves several unit operations which are affected by temperature and it makes temperature control a strategic activity in process plants. In cooling process, temperature of cooling water determines how far the process could be cooled. In Nigeria, cooling towers have limitation of not being able to cool to less than 20°C a reason why chiller is introduced to provide cooling as low as 10°C although it is associated with high energy consumption [1]. Several researches have been carried out on optimization of process for cooling process fluids.

[11] of Advanced Engineering Group, Engineering Services, Infosys Limited, India analyzed historical data from chiller plant identified operational difficulties in chiller that could lead to high operational cost of chillers. Tackling the difficulties identified means optimization of the chillers. Some of the factors identified include: incorrect sizing, low temperature difference, high condenser water inlet temperature, high temperature approach and inaccurate instruments. No optimization strategy was shown on the research work.

[14] Carried out research on “List of chilled water temperature reset strategies/system optimization” and explained that adjusting leaving chilled water temperature set point could improve chiller efficiency and optimize water-cooled

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chiller by reducing power consumed by compressor as a result of increase in water temperature. He added that chilled water temperature could be adjusted based on: outdoor air conditions and air handling unit coil position. Author concluded that a degree rise in temperature of chilled water can save 0.5 to 0.75% in kW/Tr in fixed speed chiller and 2-3% in variable frequency drive chillers. Only temperature reset approach was the only strategy used in chiller optimization. He centered on cooling of atmospheric air but not chiller for process cooling applications.

Chiller optimization could also be done by using variable pumping system especially the primary pump, changing the configuration of the chiller valve (from 3-way valve to 2-way valve) and integrating optimization into the chiller control system, integrating water side economizer with chilled water result, where mechanical cooling is eliminated. Water economizer strategy is not suitable for Nigerian climate where outdoor temperature is as high as 35°C.

Adaptive Neuro-fuzzy Interference system (ANFIS) and Genetic Algorithm (GA) was used to optimize chiller loading in 2015 [8]. They modeled power consumption of chiller within a short period using ANFIS while GA was used to optimize chiller loading. It was found that power consumption was reduced by 6.32 – 18.96% with partial load ratio of 0.6 – 0.95. They used part load ratio to develop power consumption model. They concluded that the power consumption model developed using ANFIS could also increase the convergence speeds with average processing period of 2mins.15 seconds. They used Heuristic model for comparison and optimization without the use of Aspen Hysys. 5 chillers were used in series but not double stage chiller used in this study. The type of chiller(s) used was not meant for process cooling application.

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## 2 Materials and Methods

### 2.1 Materials

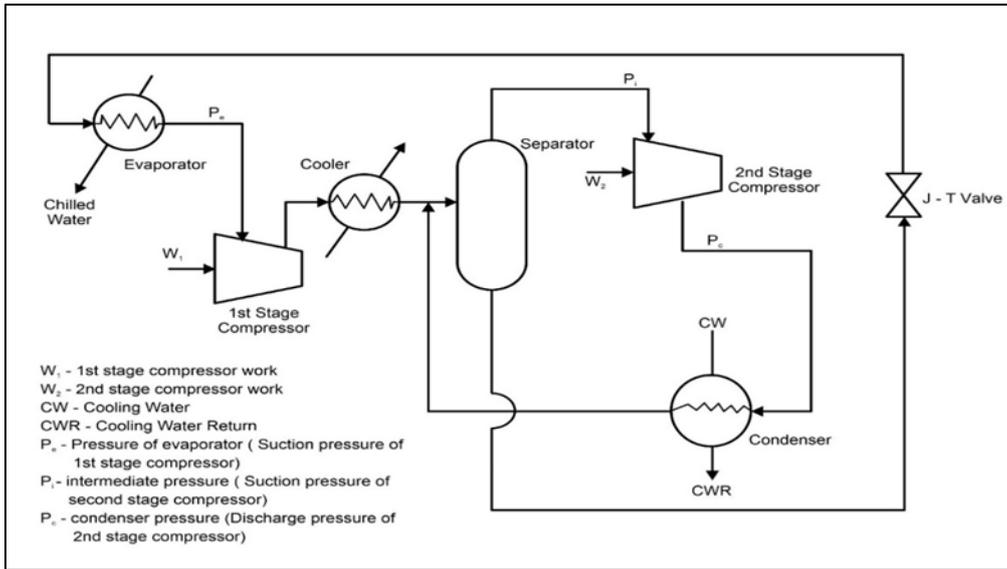
The materials used for the research work includes Aspen HYSYS version 11.0, data from PE plant distributed control system (DCS), ASHRAE handbooks, P-h diagram for R134a refrigerant among others.

**Table 1** Input data for the simulation from PE Plant DCS

Parameter (Unit)	Value
Flow rate of cyclohexane (T/h)	100.00
Temperature of cyclohexane (°C)	35.00
Pressure of cyclohexane (bar)	7.00
Inlet temperature of cooling water (°C)	29.00-30.00
Cooling water return temperature (°C)	32.00-34.00
Pressure of cooling water (bar)	3.50

### 2.2 Methods

This research project was executed by using Aspen HYSYS version 11.0. The optimization process was done as shown on Figure 1. Multi-stage compression is one of the ways for optimization of a refrigeration system [5]. Since in a multi-stage compression system, choice of intermediate pressure is a major consideration and this should do such that a minimal energy requirement is achieved by ensuring that the pressure ratio at each stage is equal, a 2-stage compressor was used and the compressor ratio of the two stages was made equal in other to optimize the chiller [9].



**Figure 1** Flow sheet for optimization of the simulated chiller

The fluid package used for the simulation was Peng Robinson while the refrigerant used was R134a. A spreadsheet was developed to determine the total compressor energy of the two stages. After converging, a case study was made at various discharge pressure (100-1000kPa) of the first stage compression and corresponding compression ratio at the respective stages were obtained. Plot of first stage discharge pressure against the compression ratio at each stage was plotted and the optimum pressure ratio was obtained from the graph.

Similarly, another case study was done using various compression ratio to obtain the total power consumption of the combined compression stages. A graph of compressor ratio against the total compression power was plotted and the corresponding total power consumption at optimum compression ratio was obtained.

Since Aspen HYSYS software has fluid package embedded, the energy requirement at the stages could be expressed as shown below considering the process to be a reversible adiabatic process [6]:

$$PV^\gamma = \text{Constant} \tag{1}$$

were  
 P = Pressure  
 V = Volume  
 $\gamma$  = Ratio of heat capacities

$$W_1 = \frac{P_{2\text{stage } 1} V_{2\text{stage } 1}^{-P_{1\text{stage } 1}} V_{1\text{stage } 1}}{1-\gamma} \tag{2}$$

were,

'w<sub>1</sub>' is the work done on stage 1 compressor and the other terms are as defined in equation 1.

$$W_2 = \frac{P_{2\text{stage } 2} V_{2\text{stage } 2}^{-P_{1\text{stage } 2}} V_{1\text{stage } 2}}{1-\gamma} \tag{3}$$

2.2.1 Specification of the Optimized Chiller and Cost Analysis

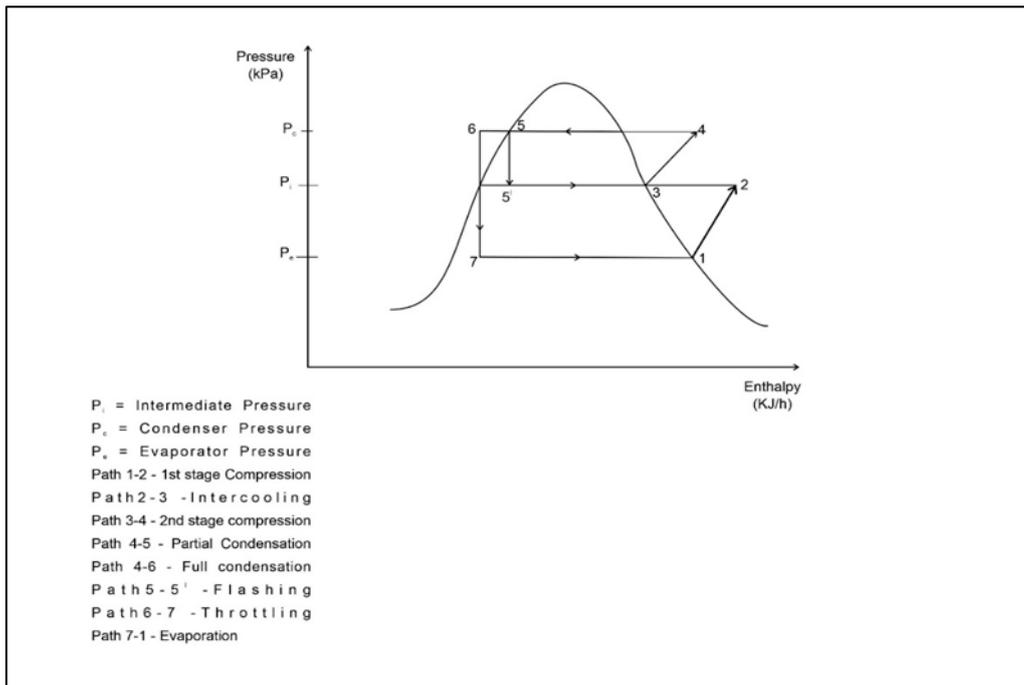


Figure 2 Pressure – enthalpy diagram for the optimized chiller

The design specification of the optimized chiller was done with the use of Aspen HYSYS. Each of the component of the chiller, compressors, evaporator, condenser, etc. was sized with the software. Peng Robinson was used as fluid package. The sizing could be done manually using the P-h diagram for R134a for a double stage compression system as shown in Figure 2. The first stage compression is path 1 2 while the second stage compression is shown in path 3 4. The intermediate pressure,  $P_i$ , correspond to the pressure at point 5 while path 7 1 shows the evaporation of the refrigerant and path 4 6 shows the path for full condensation of the refrigerant. The coefficient of performance (C.O.P) and the efficiency of the optimized chiller were also estimated.

2.1.1 Specification of the Optimized Chiller

The sizing of the four basic components of the chiller for economic analysis of the research were carried out as represented in this section.

Compressor Specification

The intermediate pressure of the design was chosen such that the pressure ratio in the two stages are equal. The intermediate pressure could be obtained as follows [5]

$$P_i = \sqrt{P_c \times P_e \times \frac{T_{\text{condenser}}}{T_{\text{evaporator}}}} \tag{4}$$

were,

$P_i$  = Intermediate pressure

$P_c$  = Condenser pressure

$P_e$  = Pressure of the evaporator

$T_{\text{condenser}}$  = Temperature of the condenser stage

$T_{\text{evaporator}}$  = Temperature of the evaporator stage

The isentropic (adiabatic) work done by the compressor could be expressed as follows [10]:

$$W_{\text{compressor}} = (h_{\text{discharge}} - h_{\text{suction}}) = \frac{\gamma}{\gamma-1} (P_{\text{discharge}}V - P_{\text{suction}}V) \tag{5}$$

were

$h$  = Enthalpy

$$\gamma = \frac{c_p}{c_v} \tag{6}$$

$\gamma$  = Isentropic index = ratio of specific heat capacities

For first stage, the compression work could be expressed as:

$$W_1 = V \frac{P_i - P_e}{1 - \gamma} = h_2 - h_1 \tag{7}$$

were

$P_i$  = Intermediate Pressure

$P_e$  = Pressure of the evaporator

$h_1$  = Enthalpy at the inlet of first stage compressor (Figure 2)

$h_2$  = Enthalpy of the at intermediate stage (point 2 on Figure 2)

$W_1$  = 1st stage work

For 2nd stage compression:

$$W_2 = V \frac{P_c - P_i}{1 - \gamma} = h_4 - h_3 \tag{8}$$

were

$P_c$  = Condenser stage pressure

$P_i$  = Intermediate Pressure

$\gamma$  = Ratios of heat capacities

$W_2$  = 2nd stage work

$h_3$  = enthalpy of the refrigerant at inlet of 2nd stage compressor

$h_4$  = Enthalpy of the refrigerant at the condenser stage

The total compression work of the chiller could be expressed as shown in equations 9, 10 and 11 respectively.

$$W_{\text{Total}} = W_1 + W_2 \tag{9}$$

Substituting equation 7 and equation 8 into equation 9:

$$W_{\text{Total}} = V \frac{P_i - P_e}{1 - \gamma} + V \frac{P_c - P_i}{1 - \gamma} \tag{10}$$

$$W_{\text{Total}} = V \frac{P_c - P_e}{1 - \gamma} = h_4 - h_1 \tag{11}$$

#### Specification of the Evaporator

The heat absorbed in the evaporator could be calculated with the use of Figure 2 as follows:

$$\text{Evaporator } Q = h_1 - h_7 \tag{12}$$

$$\text{But } h = mC_p dT \tag{13}$$

Equation 13 was substituted in equation 12:

$$\text{Evaporator } Q = mC_p(T_1 - T_7) \tag{14}$$

were,

$h_1$  = Enthalpy of the refrigerant at the outlet of the evaporator

$h_7$  = Enthalpy of the refrigerant at the inlet of the evaporator

$T_1$  = Suction temperature of the 1st stage compressor

$T_7$  = Temperatue of the refrigerant at evaporator inlet

$C_p$  = Specific heat capacity of the refrigerant

$m$  = mass flow rate of the refregerant

### Condenser Specification

The heat evolved could be calculated by taking energy balance at the condenser. With respect to Figure 2, the condenser duty,  $Q$ , is given as:

$$\text{Condenser } Q = h_6 - h_4 \quad (15)$$

### J-T Valve Specification

The J-T valve performance could be determined using equation 16 while the connection flange size was estimate using valve coefficient chart according to API 600 (2020).

$$Q = C_v \sqrt{\frac{\Delta P}{\text{Spgr}}} \quad (16)$$

### Coefficient of Performance (C.O.P) and Efficiency of the Optimized Chiller

The ideal coefficient of performance (ideal C.O.P) was calculated using equation 17 [12] and the actual C.O.P of the optimized chiller was estimated using equation 18 [13] while the efficiency of the optimized chiller was calculated using equation 19 [2].

$$\text{Ideal C. O. P} = \frac{\text{Evaporator Temperature (K)}}{\text{Condenser Temperature} - \text{Evaporator Temperature (K)}} \quad (17)$$

$$\text{Actual C. O. P} = \frac{\text{Chiller Cooling Capacity}}{\text{Total Power Input}} \quad (18)$$

$$\text{Chiller Efficiency} = \frac{\text{Actual C.O.P}}{\text{Ideal C.O.P}} \times 100 \quad (19)$$

## 2.2 Cost Analysis

The cost of energy of the chiller was estimated before and after optimization as follows:

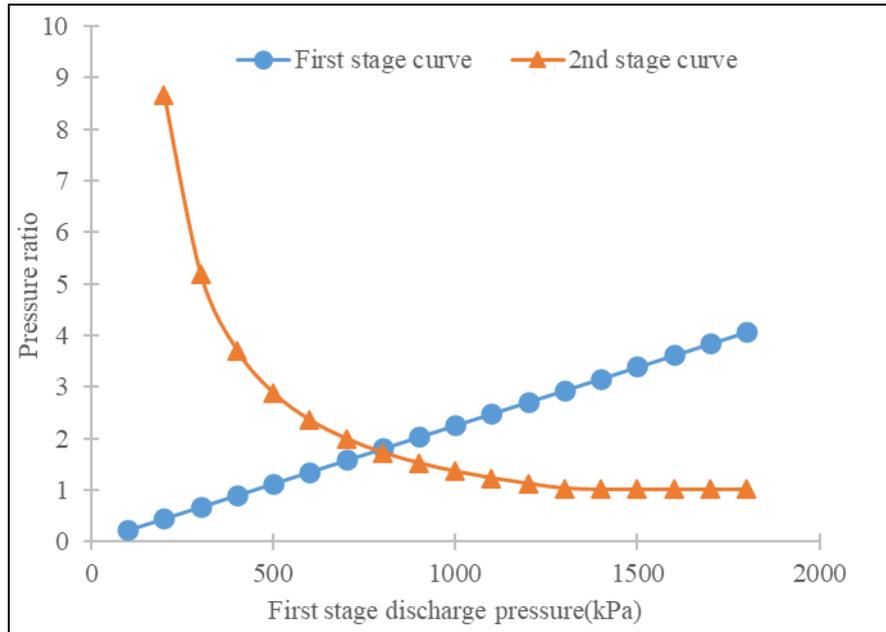
$$\text{Chiller energy consumption} = \text{Total compressor power} \times \text{operating hour} \quad (20)$$

$$\text{Chiller Energy cost (USD)} = \text{chiller energy consumption} \times 0.026 \quad (21)$$

where 0.026 is the cost of energy (in USD) per hour for Band A in Nigeria.

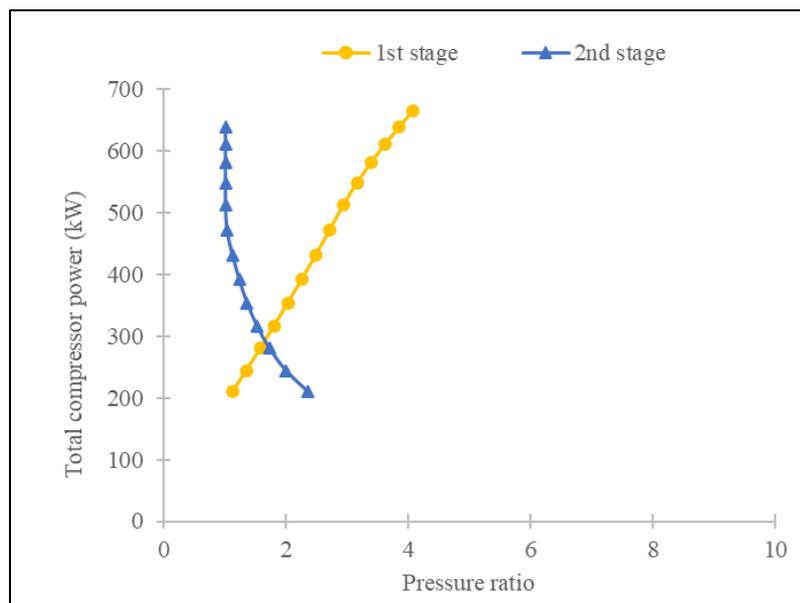
## 3 Results and Discussion

The results of the optimization of the chiller by increase of compression stage is presented in this section. It consists of the results of the optimum conditions, the effects of change of compression ratio and the impact of running the chiller at optimal conditions. Figure 3 specifies the optimum conditions for the multistage chiller, Figure 4 shows the power consumption at optimum condition while Table 2 shows the effect of running the chiller at these optimum conditions.



**Figure 3** A graph of 1st stage discharge pressure against compression ratio

Figure 3 predicts the effects of change of discharge pressure of the first stage on the compression ratio of both stages respectively. It shows that increased first stage discharge pressure increased the first stage compression ratio but reduced the compression ratio of the second stage. This is because in the double stage compression, the first stage discharge pressure became the suction pressure of the second stage and the suction pressure of the first stage was constant; compression ratio is defined as the ratio of the discharge pressure to suction pressure. The point of the intersection of the two curves is the optimal point and the corresponding compression ratio of 1.73 is the optimum compression ratio. The point of intersection represents the balance between the competing forces, compression ratio [5]. Figure 3 shows that this optimum pressure ratio could be achieved by maintaining a first stage discharge pressure of 800kPa while Figure 4 is a graph of the combined compressor power against the compression ratio at each respective stage.



**Figure 4** Effects of change in compression ratios on the total compression power

Figure 4 shows the effect of change in compression ratio on the combined power rating of the both compression stages. It shows that increased first stage pressure ratio increased the total compression power requirement of combined

stages while increased in second stage compression ratio reduced the combined compression power of the stages. The inverse relationship existed in the second stage curve because of the heat removal by inter stage cooling [4]. Cooled gases are denser than the gases from the first stage and this affects the flowrate making the second stage compressor to do less work. The optimal compression occurred when the compression ratio of was 1.733 as shown in Figure 2 and Figure 3 respectively. Figure 4 shows that the optimal combined compression power requirement of the chiller is 280.5kW.

### 3.1 Results of the specification of the optimized chiller and cost analysis

The results of the specification of the optimized chiller and cost evaluation is shown in this section. Table 2 compares the chiller before optimization and after optimization while Table 8 compares the operating cost of the chiller before and after optimization.

**Table 2** Specification of the chiller before and after optimization

Chiller Parameter (Unit)	Before Optimization	After Optimization
Compressor Power (kW)	413.30	280.50
Evaporator Duty (kJ/h)	8377000.00	8377000.00
Condenser Duty (kJ/h)	9865000.00	3048000.00
Valve Flow coefficient (Gal/min)	151.2	51.75
COP	4.23	6.24

Table 2 shows the effect of the optimization on the size of the major components of the chiller. The compressor power requirement reduced by 32.1 percent after optimization of the chiller and the condenser duty dropped by 69 percent. This happened because of the removal of heat at the intermediary state via the cooler. The J-T valve flow coefficient also dropped by 99.45gal/min due to reduction in differential pressure across the valve [7]. The evaporator duty remained unchanged since the chiller capacity (refrigeration effect) at the two scenario was kept constant [10] The COP of the optimized chiller increased to 6.24 due to the reduction in power input caused by inter-cooling. This value of COP conforms the acceptable COP value of 5.0 – 7.0 for highly efficient chillers [12].

#### 3.1.1 Specification of the optimized chiller

The size of each of the major components of the chiller as obtained from the software is presented in this sub section. They are as follows:

##### Compressor Specifications

Table 3 shows the designed properties of the optimized compressors of the chiller. It shows the design specifications of the first stage and second stage compressors.

**Table 3** Specification of the compressors

Parameter/unit	Design specification
Suction pressure of first stage compressor (kPa)	441.50
Intermediate pressure (kPa)	800.00
Discharge pressure of the 2 <sup>nd</sup> stage (kPa)	1300.00
Discharge temperature (°C)	56.89
Power consumption at the 1 <sup>st</sup> stage (kW)	121.80
Power consumption at 2 <sup>nd</sup> stage (kW)	158.70

The combined compression duty was 280.5kW and that was less than the duty when the chiller was not optimized. This was borne of the fact that energy was lost at intermediate stage. The discharge pressure of the compressor is less than the critical temperature of R134a which is 107°C and this makes the discharge pressure acceptable [5].

#### Evaporator sizing results

Table 4 shows the design specification of the evaporator. It gives the refrigerant and cooling water specifications and other major design parameters.

**Table 4** Evaporator specification

Parameter	Unit of measurement	Value
Heat duty	kJ/h	6.3 x10 <sup>6</sup>
Log mean temperature Difference (LMTD)	°C	6.800
Coefficient of heat transfer (U)	kJ/h-m <sup>2</sup> -C	1.62x10 <sup>4</sup>
Minimum Approach	°C	3.00
Heat transfer area	m <sup>2</sup>	75.40
Tube outer diameter	mm	20.00
Tube inner diameter	mm	16.00
Tube length	m	6
Shell diameter	mm	823.28
Tube side inlet temperature	°C	15.00
Tube side outlet temperature	°C	8.00
Tube side pressure	kPa	350.00
Shell side pressure	kPa	441.00
Refrigerant flow rate	m <sup>3</sup> /h	37.19
Chilled water flowrate	m <sup>3</sup> /h	278.2
Number of tubes		200

It shows that the chiller water must be cooled to 8°C in order for it to be able to cool the recycle cyclohexane to an optimal temperature of 10°C. To cool returned chiller water from 15°C to 8°C, 37.19m<sup>3</sup>/h of refrigerant (R134a) was needed while 6.3 x10<sup>6</sup>kJ/h is the evaporator heat duty. The coefficient of heat transfer was 16200kJ/h.m<sup>2</sup>.C while the area of heat transfer was 75.40 m<sup>2</sup> per shell pass. It shows that the flow rate of chiller water needed at this optimum condition was 278.2m<sup>3</sup>/h. The estimated number of tubes of the evaporator is 200 while the shell diameter is 823.28mm.

#### Condenser and expansion valve sizing results

The results of the condenser sizing are as shown on Table 5. It contains the specification of the refrigerant stream, the cooling water stream and other major design parameters.

**Table 5** Condenser Specifications

Parameter	Unit of measurement	Value
Heat duty	kJ/h	3.048x10 <sup>6</sup>
LMTD	°C	10.36
Coefficient of heat transfer (U)	kJ/h.m <sup>2</sup> .C	1858.00
Minimum Approach	°C	5.14

Tube outer diameter	mm	20.00
Tube inside diameter	mm	16
Tube length	m	6
Number of tubes		420
Heat transfer area	m <sup>2</sup>	158.30
Shell diameter	m	1181.64
Tube inlet temperature	°C	30.00
Tube outlet temperature	°C	45.00
Tube side pressure	kPa	350.00
Shell side pressure	kPa	1300.00
Chill water flow	m <sup>3</sup> /h	47.22

It shows that the condenser duty is 3048000kJ/h at a designed condenser temperature of 48.78°C. The overall heat transfer coefficient of the condenser was 1858kJ/h.m<sup>2</sup>.C while 47.22m<sup>3</sup>/h of cooling water was needed to remove 3.048x10<sup>6</sup>kJ/h of heat from the system. The area of heat transfer was 158.3m<sup>2</sup> while the number of tubes was 420.

#### Results of Sizing of Expansion Valve

Table 6 shows the design specification of the expansion valve.

**Table 6** Expansion valve specifications

Valve sizing Parameter	Unit of measurement	Value
Inlet Pressure	kPa	570.00
Valve opening	%	50.00-60.00
ΔP	kPa	350.00
Flow rate	Kg/h	46200.00
Flow coefficient C <sub>v</sub>	gpm	51.75
Vapour fraction		0.1826
Connection size	in	3.00

A thermostatic expansion valve was considered and it had a flow coefficient of 51.75gpm at 50-60% opening. The flow rate across the valve was 46200kg/h while a pressure drop of 350kPa was observed at 50% valve opening and a flange connection size of 3 inches.

#### Optimized Chiller Performance

The results of the determination of the performance of the optimized chiller is shown in this section.

**Table 7** Chiller Performance data

Chiller Performance Data	Value
Theoretical COP	8.64
Actual COP	6.24
Chiller Efficiency (%)	72.2

Table 7 shows the value of the theoretical COP, actual COP and the efficiency of the optimized chiller. The theoretical COP is less than the actual COP due to real-World losses and inefficiencies. Also, the theoretical COP is obtained when ideal conditions are assumed. The value of actual COP shows that the optimized chiller is efficient and energy cost will be minimized [5].

#### Cost Analysis

The cost of running the chiller before and after the optimization process is analysed in this section.

**Table 8** Cost of energy for the chiller

Equipment	Before Optimization		After Optimization	
	Energy consumed / Day (kWh)	Cost (USD)	Energy consumed/Day (kWh)	Cost (USD)
Chiller	9919.20	257.90	6732.00	154.80

Table 8 shows that the energy consumption of the chiller dropped by 66.6% after optimization. This is because optimization dropped the total power input of the compressor and this increased the actual COP of the chiller and its efficiency.

## 4 Conclusion

Water cooled chiller was optimized; the compressor power input of the chiller was reduced from 413.3kW to 280.5kw. Optimization was achieved by employing double stage compression where the compression ratio of both stages were maintained equal at 1.73. The optimization was done using Aspen HYSYS software. The optimized chiller could cool process fluid like recycled cyclohexane stream to 10°C with energy cost reduction of 66.6%.

The efficiency of the optimized chiller was increased to 72.2% and the daily operating cost of the chiller in Nigeria reduced from 257.9USD to 154.8USD because intercooling effect. The actual COP of the optimized chiller is 6.24 and it is within the acceptable range of 5.0-7.0 for efficient chillers.

It could be concluded that the intermediate pressure of 800kPa was maintained in other to optimize the chiller and power consumption at the first stage and second stage was 121.8kW and 158.7kW respectively. The condenser water flow was maintained at 47.22m<sup>3</sup>/h in other to achieve cooling. The expansion valve of the chiller had a flow coefficient of 51.57gpm with connection size of 3 inches while a pressure drop of 350kPa was maintained at 50-60% valve opening in other to achieve the optimum condition.

## Compliance with ethical standards

### *Disclosure of conflict of interest*

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

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