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Research paper

Basic design of a cooling system for nylon 6, 6 process

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Abstract

The project concentrates on the basic design of a cooling system for rapidly cooling nylon 6, 6 polymer fibers using cold air. The ambient air after pre-treatment in the air-washer is available at 72°F all year round. Based on the company's throughput, it is required to supply (quench) air at 58°F. Nylon 6, 6 polymer after thorough polymerization is distributed through 16 quench cabinets and each quench cabinet requires approximately 530 ft3/min (cubic feet per minute, CFM) of air. The project concentrates on the basic design of a cooling system wherein air at the required mass flow rate is supplied at 58°F for the quenching process. A basic design of the refrigeration cycle and heat exchangers were considered in this work. In the development of the basic design for heat exchanger, performance charts were developed. Performance charts describe the performance of the heat exchanger in terms of fundamental dimensionless parameters. Using performance charts it was clearly seen that increasing the number of transfer units (NTU) doesn't necessarily increase the rate of heat transfer. Increasing the NTU beyond an optimum value is pointless and increases the capital cost of the heat exchanger. The preliminary design involves selection of appropriate NTU and capacity rate ratio for the heat exchanger. From the capacity rate ratio and NTU, it is fairly straight forward to extrapolate the detailed design for the heat exchanger. A cooling system model was developed for the design process and for the simulation of the cooling system.

Keywords: *Cooling System Design; Preliminary Heat Exchanger Design; Performance Charts.*

1. Introduction

Use of cooling systems are prevalent in all process and manufacturing industries for addressing various cooling requirements in the plant. Cooling requirements can be numerous; it can be for equipment cooling, or for air conditioning in buildings, or for the industrial process itself. Likewise, all large commercial buildings employ cooling systems for providing the required air conditioning in the buildings. It must be recognized that in large scale refrigeration systems, where substantial amount of cooling is required, a secondary refrigerant such as chilled water is employed. This is because in process plants and in commercial air conditioning systems, refrigeration systems are centrally located in one place and it is difficult to pipe the refrigerant to circulate over large distances. Therefore, a secondary refrigerant such as chilled water is often employed. Herein, the chilled water comes in contact with the medium that requires cooling. It is easy to circulate water over large distances. Leakages from water lines can be tolerated and water is non corrosive. Above all, water has high specific heat value that enables it to remove large quantities of heat. Therefore, chilled water systems are widely used in large-scale refrigeration systems.

This project considers the development for a chilled water cooling system for a nylon 6, 6 production unit. Nylon 6,6 polymers are widely used in fabrics, providing re-enforcement strength for car tires, plane tires, belts, ropes, etc. During the production of nylon 6, 6 polymer it is required to quench (rapidly cool) highly viscous polymer using pre-treated environmental air. Quenching process solidifies the viscous nylon 6, 6 polymer into fine threads before they are drawn into yarn for varying properties and requirements.

The environmental air after pretreatment in an air washer is available at 72°F at all times. The flow rate of air is 530 CFM per quench cabinet. There are 16 quench cabinets and therefore the requirement is to cool 8480 CFM of air from 72°F to 58ºF. Abundant space is available to erect the cooling system. Pre-existing air ducts connect the air washer with the quench cabinets. This design project aims in developing a basic integrated cooling system such that 8480 CFM of air is cooled from 72°F to 58ºF. As we can readily observe, this requires the development of a cross flow heat exchanger such that air exchanges heat with a cooling medium, presumably, chilled water. Likewise, it is essential for the cooling medium (chilled water) to exchange heat with a refrigeration system such as a chiller. Therefore, it is required to develop a refrigeration cycle and also a pumping system to circulate chilled water through the cross flow heat exchanger and through the refrigeration (chiller) unit. For brevity, the development of the pumping system is not discussed in this paper.

The major components of a vapor compression refrigeration cycle consists of an evaporator, a condenser, a compressor and a throttling device. This work shall also concentrate on the basic development of an evaporator and condenser equipment. Most chillers have water cooled condensers as water provides better heat transfer as compared to air and enhances the life span of the chiller unit. However, a water-cooled condenser will require an additional design of a cooling water circuit for the condenser equipment. For brevity, the design of a suitable pumping system for the cooling water and a selection of a cooling tower are not discussed herein.

There are numerous references available in the literature pertaining to heat exchanger performance modeling, and only the most pertinent are discussed. Kays and London [1] and Rohsenow [2] described both the logarithmic mean temperature difference and the $\varepsilon - NTU$ methods in order to size and predict the performance of a heat exchanger. Domingos [3] presented a general method of calculating overall performance and intermediate temperatures of complex crossflow heat exchangers using the concept of effectiveness and a local energy balance. Pignotti and Shah [4] and Shah and Pignotti [5] discussed the tools developed previously (such as Domingos' method, the Pignotti chain rule, etc.,) to determine the $\varepsilon - NTU$ relationship for highly complex heat exchanger flow arrangements. As compared with the present investigation, these studies pertained to quite different geometries such as cross flow and shell and tube heat exchangers. Furthermore, they did not address the design of optimal heat exchangers, which achieve the required task at the lowest cost while satisfying imposed constraints. Mott and Mills [6] and Genic et al. [7] are among those researchers who described optimization analysis based primarily on minimizing energy costs related to pumping of a fluid. Kovarik [8] described a technique to optimize a cross flow heat exchanger. The objective function which was employed included cost factors related to the heat exchanger size and pumping power, as well as the required heat transfer rate. Similarly Rao et al. [9] and Caputo et al. [10] proposed methods to minimize capital and operating costs for shell-and-tube heat exchangers while satisfying the required heat transfer duty. Silaipillayarputhur et al [11] developed a pumping system for a heat transfer fluid. Therein, the details for choosing an appropriate pump, head loss calculations, net positive suction head (NPSH) calculations are detailed and such concepts are applied in this work while designing pumping system for the heat exchanger.

2. Nomenclature

Symbol	Name
A	Heat exchanger surface area (m^2)
ADP	Apparatus dew point temperature ($°C$)
BP	Bypass factor (dimensionless)
C_{min}	Minimum capacity rate (W/K)
C_{max}	Maximum capacity rate (W/K)
COP	Coefficient of performance (dimensionless)
C_p	Specific heat (J/kg.K)
C_r	Capacity rate ratio (dimensionless)
h	Enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
n	Number of passes (dimensionless)
NTU	Number of transfer units (dimensionless)
Q	Volume flow rate (m^3/s)
$\dot{Q}$	Rate of heat transfer (W)
$\dot{W}_c$	Compressor work (W)
T	Temperature ($°C$)
U	Overall heat transfer coefficient (W/m^{2}K)
ΔT_m	Log mean temperature difference (K)
Symbols	
ρ	Density (kg/ m^3)
ϵ	Effectiveness (dimensionless)
η	Efficiency (dimensionless)
Subscripts	
hi	Hot fluid inlet
ho	Hot fluid exit
ci	Cold fluid inlet
co	Cold fluid exit
A	Minimum capacity rate fluid
B	Maximum capacity rate fluid
1	One shell pass, multiple tube passes (shell and tube heat exchanger)
Superscripts	
Per pass (cross flow heat exchanger)	

3. Basic layout of the cooling system

This project considers the design of a cooling system for cooling 8480 CFM of air from 72ºF to 58ºF. The cooling system is described in Figure 1. As described before, the cooling system consists of a cross flow heat exchanger where the air is cooled by chilled water, a refrigeration system for cooling the chilled water and a cooling water system for cooling the refrigerant in the refrigeration unit.

In this project, a basic design for the cross flow heat exchanger is considered. Therein, performance charts describing the performance of the heat exchanger in terms of significant dimensionless parameters are developed. The parameters are number of transfer units (NTU), capacity rate ratio (C_r), and heat exchanger effectiveness (ε). These three parameters completely describe the heat exchanger as they encompass physical, thermal, fluid, and material characteristics of the heat exchanger. Domingos approach [3] was employed in the development of performance charts. Using these charts engineers can quickly determine the required NTU and capacity rate ratio for the heat exchanger. Using these parameters, it is fairly straight forward to extrapolate the detailed design of the heat exchanger. The development of performance charts is described in the subsequent sections of this paper. Likewise, the development of a basic refrigeration cycle is also considered in the paper. The choice of refrigerant, capacities for evaporator & condenser equipment, and requirements for the compressor were all computed. Therein, it must be recognized that the evaporator and condenser equipment are basically shell and tube heat exchangers and performance charts for shell and tube heat exchangers were developed by using the explicit relations provided in [12]. Thereafter, the basic design for the evaporator and condenser equipment were developed as well.

The cooling system must be sized such that the required cooling is provided in the quench cabinets. The mass flow rate of air can be determined as follows.

$$\dot{m}_{air} = \rho Q \tag{1}$$

Where ρ is the density of air and Q is the flow rate of air. Therefore, the required amount of cooling in the cross flow heat exchanger can be readily computed as follows

$$\dot{Q}_{air} = \dot{m}_{air}\, c_{p,air}\left[T_{Ai} - T_{Ao}\right] \tag{2}$$

Here $c_{p,air}$ is the specific heat of air and T_{Ai} and T_{AO} are inlet and discharge temperatures of air respectively. From Equation (2), the required cooling load for the system can be readily determined.

Performance charts for cross flow heat exchanger

Consider a cross flow heat exchanger as described in Figures 2 and 3. Therein, the external fluid is in gaseous form and tube side fluid is in liquid state. The overall NTU of the heat exchanger can be described as follows [12]

$$NTU = \frac{UA}{C_{min}} \tag{3}$$

Most heat exchangers in process industries are configured such that the external fluid, the gas, is the minimum capacity rate fluid and therefore the NTU of the heat exchanger may be given as

$$NTU = \frac{UA}{\dot{m}_A\, Cp_A} \tag{4}$$

Herein, subscript A refers to the minimum capacity rate fluid. Likewise, the capacity rate ratio of the heat exchanger may be given as

$$C_r = \frac{C_{min}}{C_{max}} \tag{5}$$

Assuming that external fluid is the minimum capacity rate fluid

$$C_r = \frac{\dot{m}_A Cp_A}{\dot{m}_B Cp_B} \tag{6}$$

Where subscript B refers to the maximum capacity rate fluid.

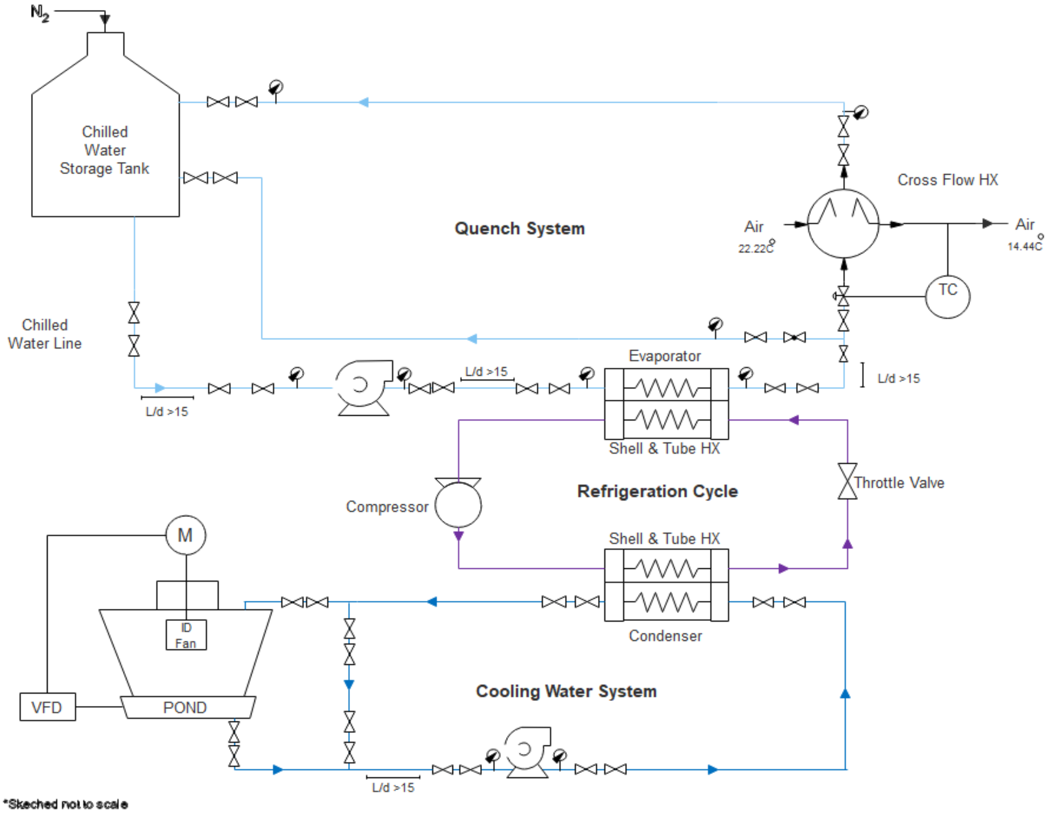

Fig. 1: Basic Layout of the Cooling System.

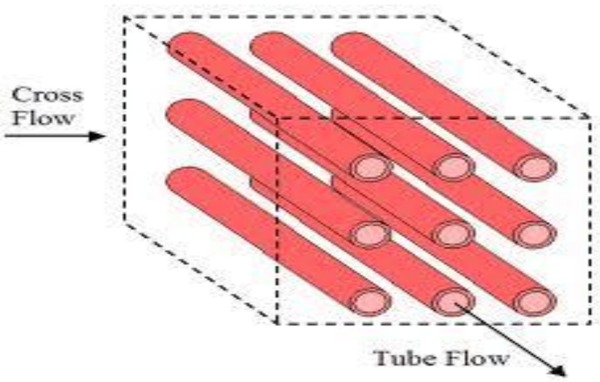

Fig. 2: Cross Flow Heat Exchanger.

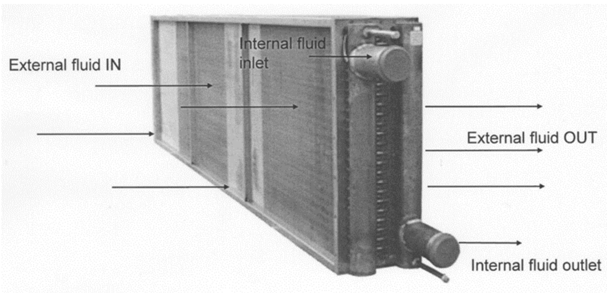

Fig. 3: Finned Cross Flow Heat Exchanger (Parallel Flow Configuration).

A cross flow heat exchanger can have several passes. A heat exchanger "pass" may be visualized as the number of times a tube side fluid particle travels through the entire length of the heat exchanger.

Figure 4 and Figure 5 depict a 2 pass cross flow heat exchanger in parallel and counter flow configurations.

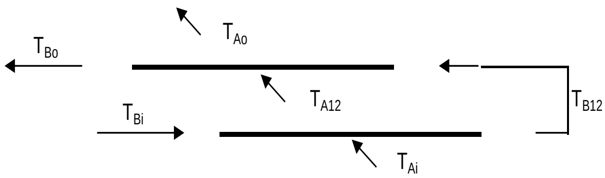

Fig. 4: Two Pass Parallel Cross Flow Heat Exchanger.

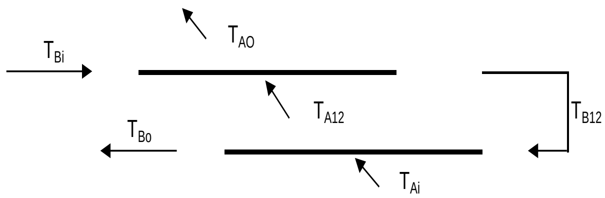

Fig. 5: Two Pass Counter Cross Flow Heat Exchanger

Considering a cross flow heat exchanger with n number of heat exchanger passes, the heat exchanger surface area may be assumed to be evenly divided among heat exchanger passes. Therefore, the NTU per pass:

$$NTU' = \frac{NTU}{n} \tag{7}$$

Note that n is the number of heat exchanger passes. In-addition, each pass encounters the full mass flow rate of the tube side and full mass flow rate of the external fluid. Therefore, capacity rate ratio per pass may then be given as

$$C_r' = C_r \tag{8}$$

The effectiveness per pass of a cross flow heat exchanger may be given as [12]

$$\varepsilon' = 1 - \exp\left[\left(\frac{1}{C_r'}\right) * NTU'^{0.22} * \left(\exp\left[-C_r' * NTU'^{0.78}\right]\right) - 1\right] \tag{9}$$

From Domingos [3], for a parallel flow heat exchanger, the overall effectiveness of the heat exchanger may be given as

$$\varepsilon = \frac{1 - \left[1 - \varepsilon'\left(C_r' + 1\right)\right]^n}{C_r' + 1} \tag{10}$$

And from Domingos [3], for a counter flow heat exchanger, the overall effectiveness of the heat exchanger may be given as

$$\varepsilon = \frac{n\varepsilon'}{1 + (n-1)\varepsilon'} \tag{11}$$

Performance charts for the parallel cross flow heat exchanger were developed by employing Equations (9) and (10) and whereas performance charts for counter cross flow heat exchanger were developed by employing Equations (9) and (11). The NTU of the heat exchanger was arbitrarily varied between 0.1 and 10, as this is the common working range of NTU seen in process industries. The capacity rate ratio is varied between the limits of 0 and 1.

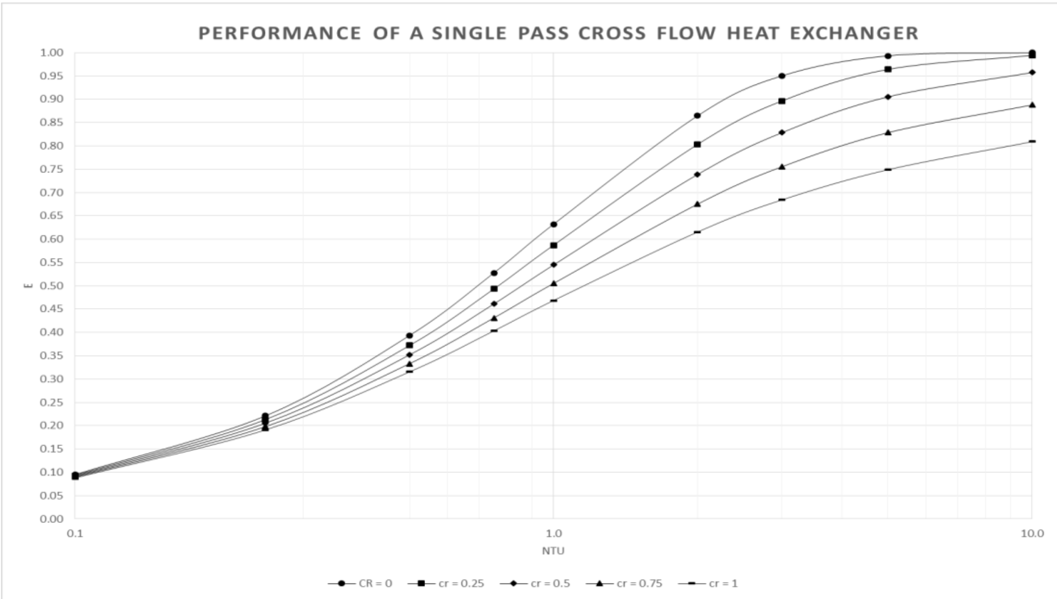

Fig. 6: Performance Chart – Single Pass Cross Flow Heat Exchanger.

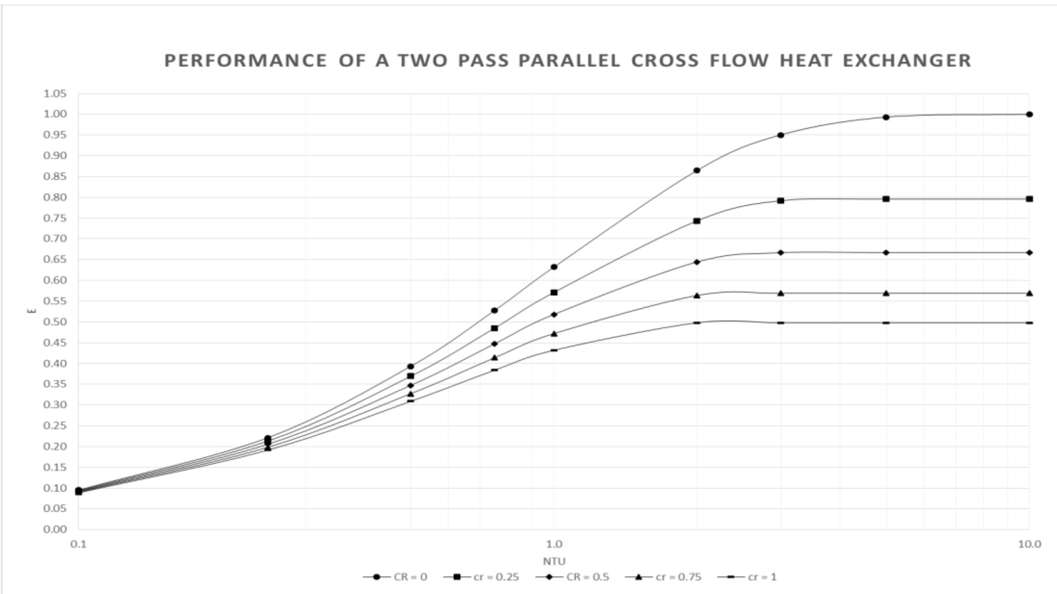

Fig. 7: Performance Chart – Two Pass Parallel Cross Flow Heat Exchanger.

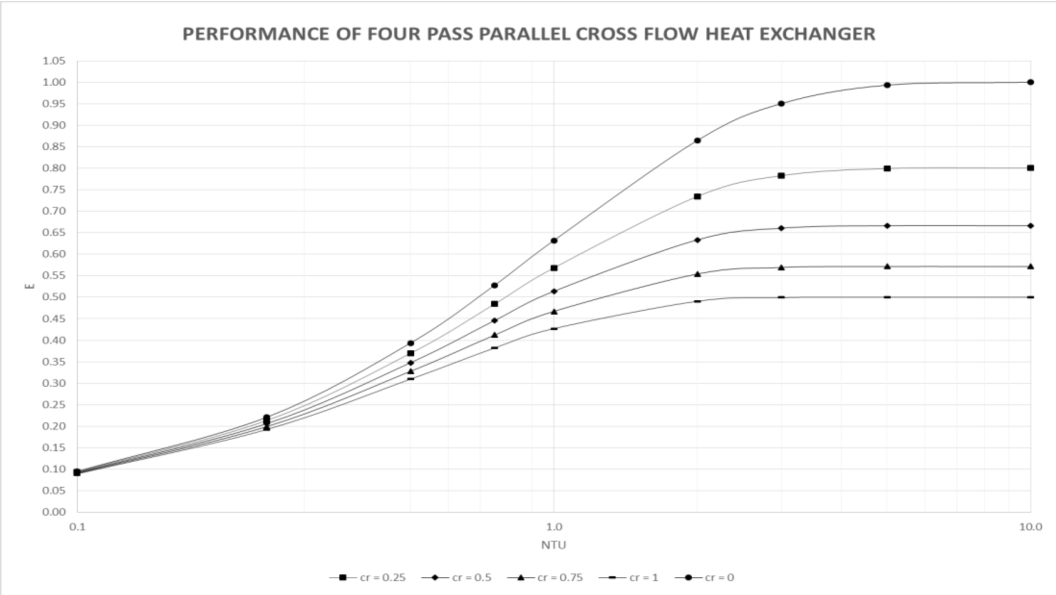

Fig. 8: Performance Chart – Four Pass Parallel Cross Flow Heat Exchanger.

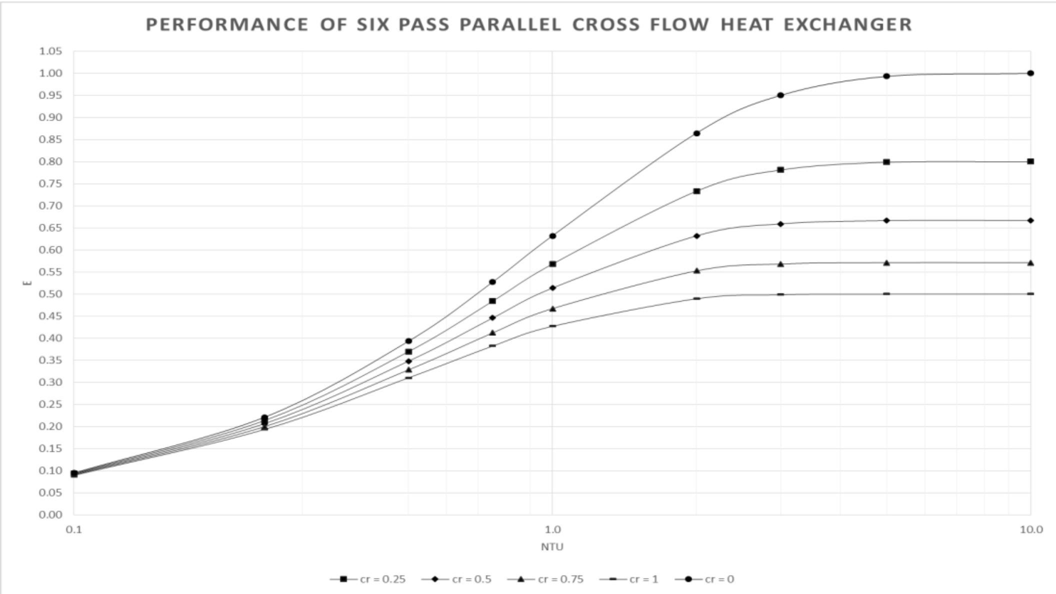

Fig. 9: Performance Chart – Six Pass Parallel Cross Flow Heat Exchanger.

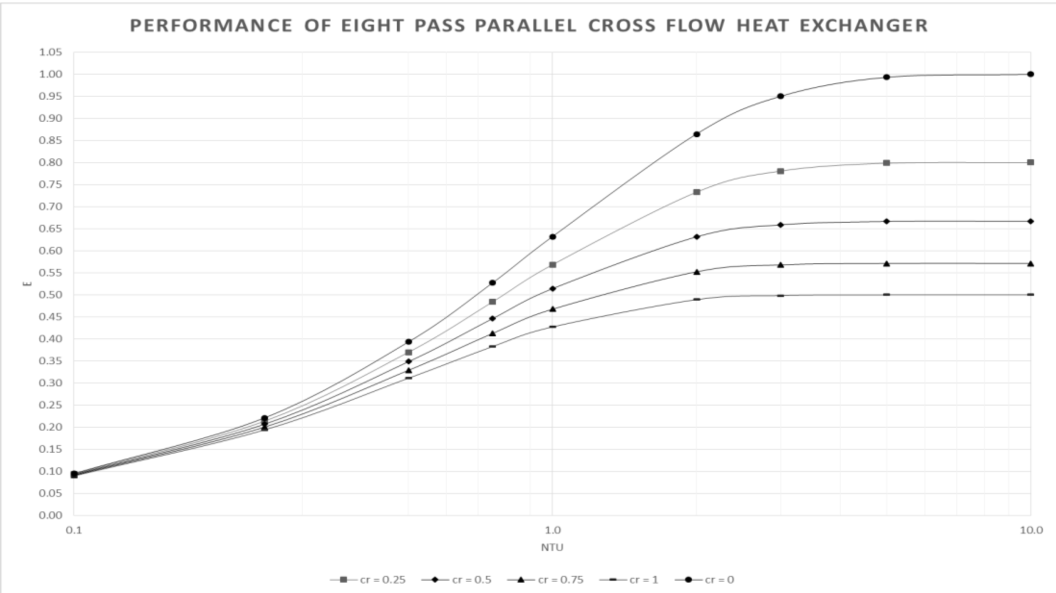

Fig. 10: Performance Chart – Eight Pass Parallel Cross Flow Heat Exchanger.

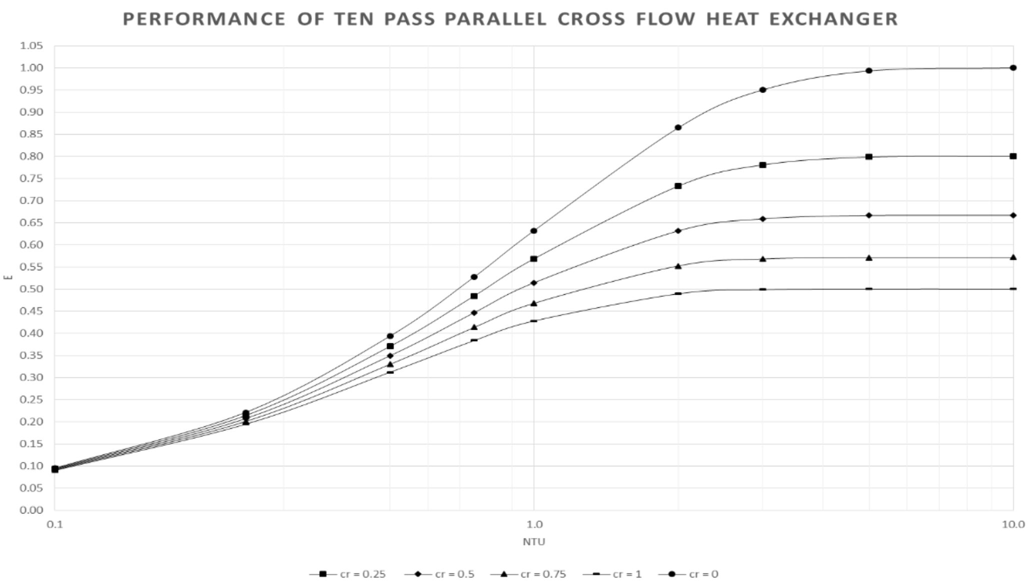

Fig. 11: Performance Chart – Ten Pass Parallel Cross Flow Heat Exchanger.

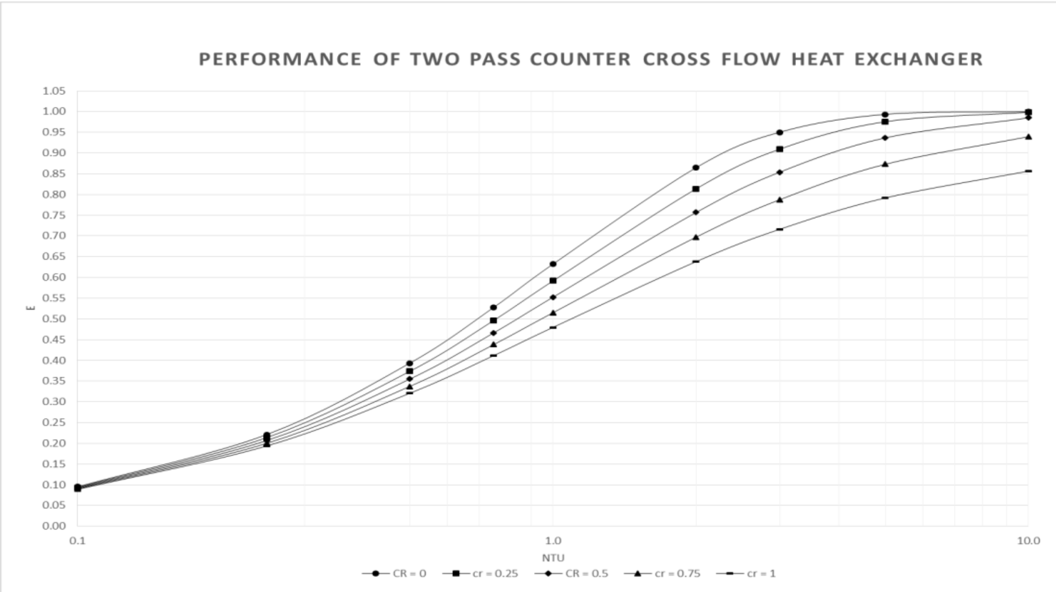

Fig. 12: Performance Chart – Two Pass Counter Cross Flow Heat Exchanger.

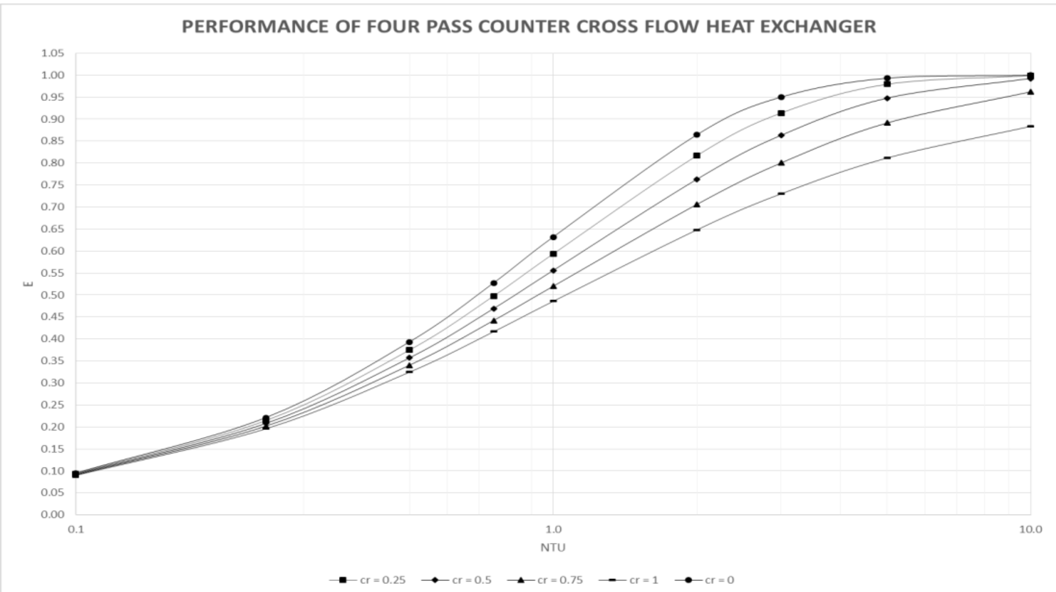

Fig. 13: Performance Chart – Four Pass Counter Cross Flow Heat Exchanger.

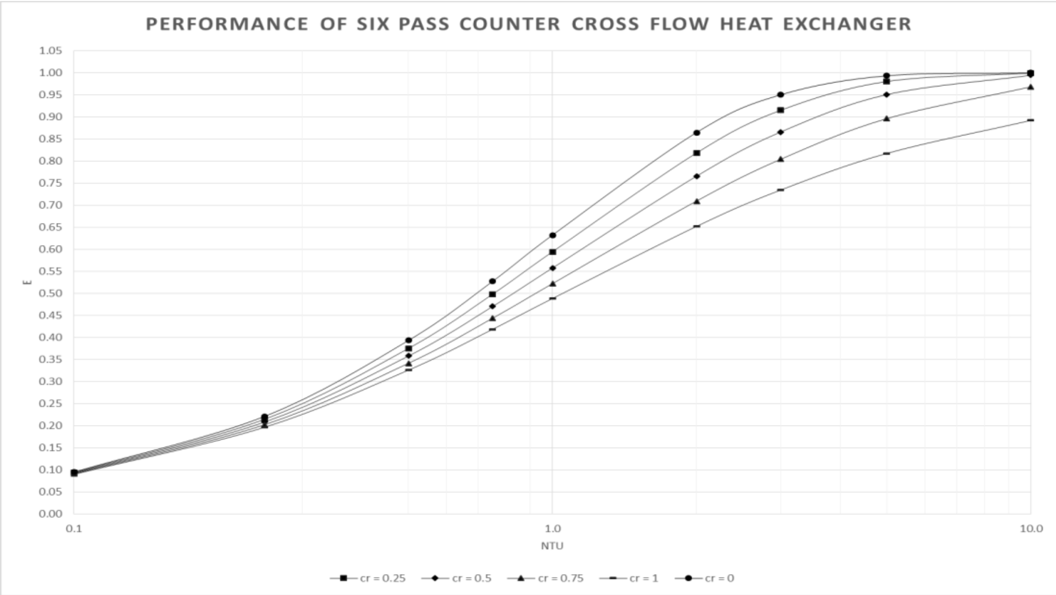

Fig. 14: Performance Chart – Six Pass Counter Cross Flow Heat Exchanger.

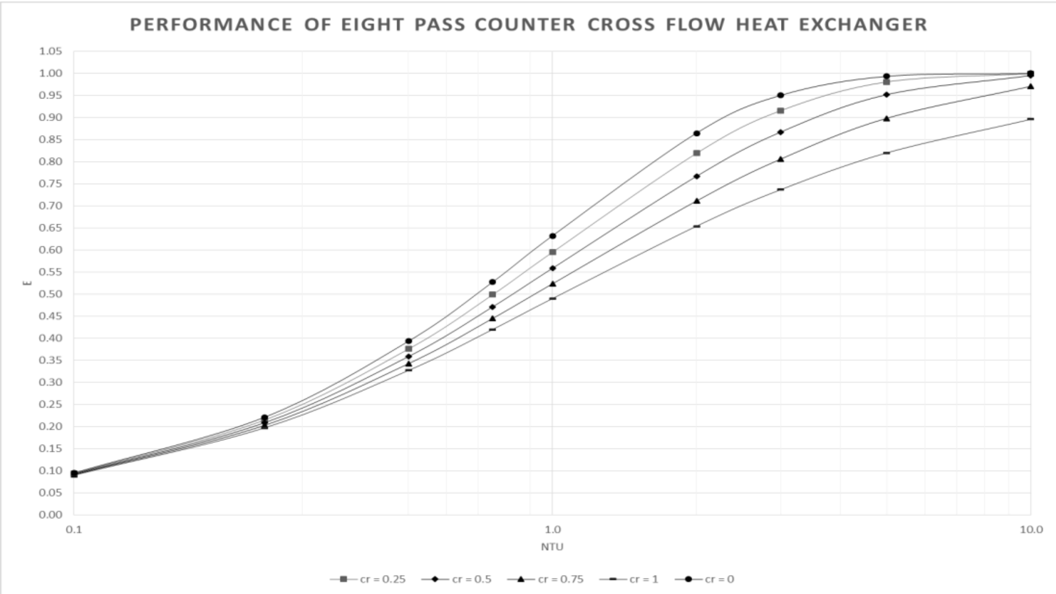

Fig. 15: Performance Chart – Eight Pass Counter Cross Flow Heat Exchanger.

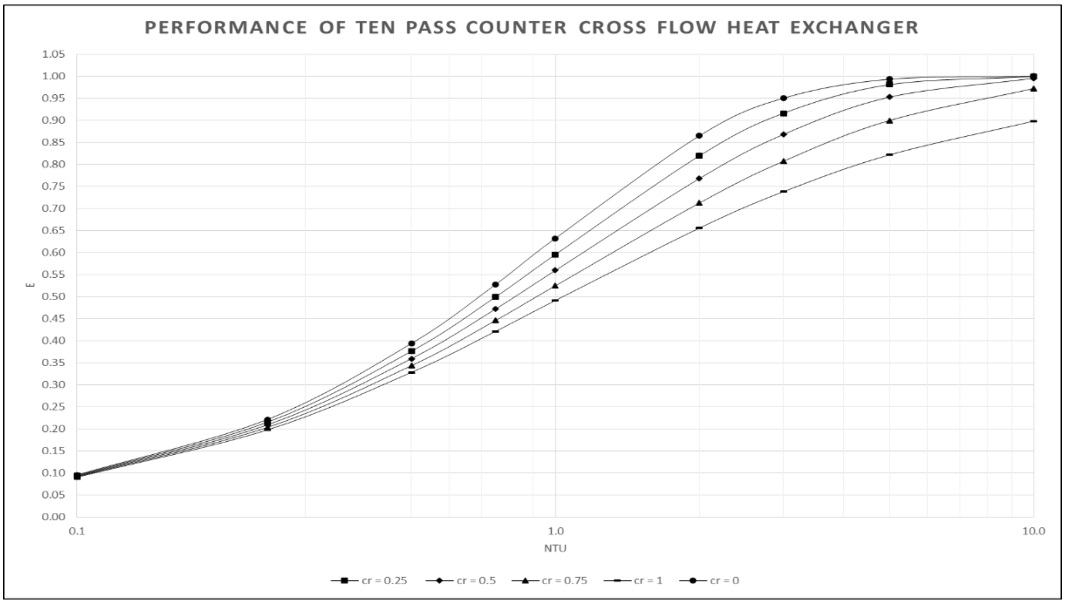

Fig. 16: Performance Chart – Ten Pass Counter Cross Flow Heat Exchanger.

Using the performance charts, the basic design of the cross flow heat exchanger can be ascertained. The basic design involves determination of the required NTU and capacity rate ratio for the cross flow heat exchanger. Heat exchangers are often characterized by NTU as they account for all the properties of the heat exchanger, size and configuration. These are described in the subsequent sections. From the basic design, it is easy to develop a detailed design for the cross flow heat exchanger. The basic design of the cross flow heat exchanger is described in Figures 17 and 18.

Inputs for the design of cross flow heat exchanger				
Air volume flow rate	8480	ft^3/min	4	m^3/s
Air density	1.2	kg/m^3		
Air supply tempreture	72	ºF	22.4	ºC
Required air discharge tempreture	56	ºF	14.6	ºC
Chilled water inlet temperature (assumed)	10	ºC	22.4	ºC
Specific heat of air $c_{p,air}$	1.005	kJ/kg.k	1005	J/kg.K
Specific heat of chilled water $c_{p,water}$	4.192	kJ/kg.k	4192	J/kg.K
Minimum capacity rate fluid - cross flow heat exchanger	Air			
Maximum capacity rate fluid - cross flow heat exchanger	Chilled water			

Fig. 17: Inputs for the Design of Cross Flow Heat Exchanger.

Cross flow heat exchanger design calculations						
Air mass flow rate	4.8	kg/s				
Cross flow heat exchanger load	37.6	kw	135458	kJ/hr	11	ton
Required heat exchanger effectiveness	0.63					
Selected HX configuration	counter					
Required NTU (chosen from performance charts)	1.2					
Required # of passes (chosen from performance charts)	4					
Required capacity rate ratio (chosen from performance charts)	0.25					
Capacity rate of air	4824	W/K				
Required capacity rate of chilled water	19296	W/K				
Required mass flow rate of chilled water	4.6	kg/s				
Anticipated chilled water discharge temperature	12.0	ºC				

Fig. 18: Cross Flow Heat Exchanger Basic Design Calculations.

4. Design of the refrigeration cycle

Design of the refrigeration cycle consists of the selection of the refrigerant, calculating mass flow rate of the refrigerant, determining the operating conditions for evaporator and condenser, and selection of a compressor and a throttling device. There are several refrigerants that can be employed for the refrigeration system. However, the most popular and the environmentally friendly refrigerant is R134a (Tetrafluoroethane) is chosen for this application. Consider Figure 19. describing the various components of the vapor compression refrigeration cycle.

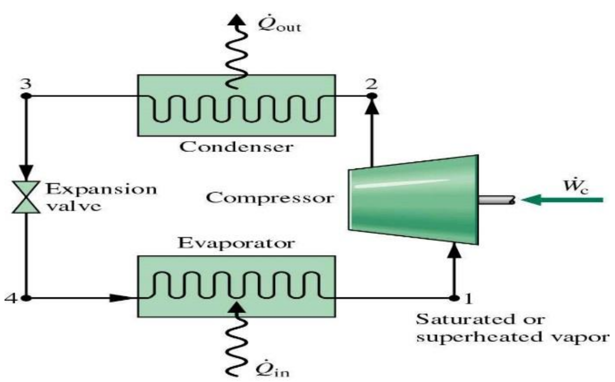

Fig. 19: Vapor Compression Refrigeration System.

The following are the assumptions employed in the development of the refrigeration cycle:

1) Evaporator and condenser operate at constant pressure.
2) Compressor efficiencies vary with compressor type, size, and throughput. They can only be determined by a compressor test, although compressor manufacturers can usually provide good estimates. For planning purposes, the compressor efficiency may be assumed as 0.85; (i.e.) $\eta_c = 0.85$ [13].
3) The by pass (BP) factor, the parameter that describes the percentage of air that is not cooled to the apparatus dew point temperature (the surface temperature of the evaporator coil) can be assumed as 0.2; (i.e.) BP = 0.2.
4) The refrigerant is a saturated vapor entering the compressor.
5) The refrigeration unit (the chiller) is assumed to be a water cooled chiller as described in Figure 1.
6) The condenser operating temperature and pressure is dependent on the condensing temperature of the refrigerant in the condenser.

The condensing temperature of the refrigerant is fixed based on the available cooling water temperature and the worst environmental conditions.

7) The refrigerant is a saturated liquid leaving the condenser.

Considering Figure 1, it can be clearly seen that the evaporator is a shell and tube heat exchanger wherein the refrigerant exchanges heat with the chilled water. As always, the refrigerant does not mix with chilled water and they are separated by means of a solid wall. Recall that the rate of cooling in the cross flow heat exchanger is known from Equation (2). Recognize that the cross flow heat exchanger effectiveness or the efficiency of the heat exchanger may also be described as the ratio of heat transfer to the maximum possible heat transfer, thus

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (12)$$

Or

$$\varepsilon = \frac{\dot{Q}}{C_{min}[T_{Ai} - T_{Bi}]} \quad (13)$$

By assuming a suitable inlet temperature for the chilled water, the required effectiveness of the cross flow heat exchanger can be readily computed. Thereafter, employing performance charts, a reasonable capacity rate ratio and the NTU for the heat exchanger can be determined.

The capacity rate ratio is described by Equation (6). By employing Equation (6), the required mass flow rate of chilled water can be readily computed. Applying energy balance to the chilled water yields

$$Q_{water,chilled} = \dot{m}_{water,chilled}\, c_{p,water}\left[T_{BO} - T_{Bi}\right] \quad (14)$$

Here, T_{Bi} refers to the chilled water supply temperature in the cross flow heat exchanger and T_{BO} refers to the chilled water discharge temperature from the cross flow heat exchanger.

The rate of heat transfer to the cooling water or the heat gained by the cooling water is equal to the rate of heat loss from air.

$$Q_{water,chilled} = Q_{air} \quad (15)$$

Employing Equations (14) and (15), the chilled water discharge temperature from the cross flow heat exchanger can be readily computed.

The by pass factor of the evaporator coil may be defined as

$$BP = \frac{T_{c,in} - ADP}{T_{c,out} - ADP} \quad (16)$$

Where $T_{c,in}$ refers to the temperature of chilled water entering the evaporator and $T_{c,out}$ refers to the temperature of chilled water leaving the evaporator. It must be recognized that the chilled water leaving the evaporator feeds the cross flow heat exchanger (chilled water supply) and the chilled water leaving the cross flow heat exchanger (chilled water return) enters the evaporator. Therefore Equation (16) may be rewritten as

$$BP = \frac{T_{BO} - ADP}{T_{Bi} - ADP} \quad (17)$$

Except ADP, all parameters in Equation (17) are known. This will help to determine the ADP and the working temperature and pressure of the refrigerant in the evaporator coil.

The condenser operating temperature and pressure is set based on condensing temperature of the refrigerant. The compressor work can be described as follows:

$$\dot{W}_c = \dot{m}_{ref}\left[h_1 - h_2\right] \quad (18)$$

The condenser, a shell and tube heat exchanger as well, is designed such that the refrigerant leaves the condenser as a saturated liquid. The condenser working pressure and temperature is chosen such that the cooling water provided by the cooling tower can condense the refrigerant to a saturated liquid. Likewise, the expansion in a thermostatic expansion valve can be assumed as an isoenthalpic process, such that

$$h_3 = h_4 \quad (19)$$

The required mass flow rate of the refrigerant can then be determined by applying energy balance to the evaporator

$$\dot{Q}_{evaporator} = \dot{m}_{ref}\left[h_1 - h_4\right] \quad (20)$$

Recognize that

$$\dot{Q}_{air} = \dot{Q}_{water,chilled} = \dot{Q}_{evaporator} \quad (21)$$

The above Equation describes that the rate of heat transfer from the air is equal to the rate of heat absorbed by the chilled water which is likewise equal to the rate of heat absorbed by the refrigerant in the evaporator. The coefficient of the refrigeration (COP) cycle is described as

$$COP = \frac{Q_{evaporator}}{\dot{W}_c} \quad (22)$$

The development of the refrigeration cycle is in Figures 20 and 21.

Inputs for the design of refrigeration cycle				
Refrigerant	R134a			
By pass factor (evaporator)	0.2			
Minimum capacity rate fluid - evaporator	Chilled Water			
Maximum capacity rate fluid - evaporator	Refrigerant (R134a)			
Compresser effeciency	0.85			
Al hasa worst case ambient temperature (DBT)	45	ºC		
Al hasa worst case relative humidity (RH)	70	%		
Wet bulb temperature (corresponding to the worst conditions)	39.39	ºC		
Condenser operating temperature (corresponding to WBT)	39.39	ºC		
Refrigerant (R134a) phase change (condensing) temperature	52.43	ºC		
Minimum capacity rate fluid - condenser	Cooling water			
Maximum capacity rate fluid - condenser	Refrigerant (R134a)			
Minimum cooling anticipated in cooling tower (ΔT)	4	ºC		

Fig. 20: Inputs for the Design of the Refrigeration Cycle.

Refrigeration cycle design calculations						
Evaporator - shell and tube heat exchanger						
Evaporator coil surface temperature (ADP)	9.513	ºC				
Evaporator operating temperature	8.93	ºC				
Evaporator operating pressure (corresponsing to temp)	4	bar				
Refrigerat (R134a) phase change temperature	8.93	ºC				
Capacity rate ratio of evaportaor	0					
Required heat exchanger effectiveness; evaporator	0.65					
Required NTU (chosen from performance charts)	1.75					
Condenser - shell and tube heat exchanger						
Al Ahsa worst ambient wet bulb temperature	39.39	ºC				
Lowest possible cooling water temperature provided by cooling tower under worst conditions	39.39	ºC				
Refrigerant (R134a) condensing temperature	52.43	ºC				
Condenser operating temperature	52.43	ºC				
Condenser operating pressure (corresponding to temp)	14	bar				
Capacity rate ratio of condenser	0					
Required heat exchanger effectiveness; condenser	0.31					
Required NTU (chosen from performance charts)	0.5					
Refrigerant properties and cycle calculations						
Enthalpy of refrigerant leaving the evaporator (sat. vapor) h_1	252.32	kJ/kg				
Entropy of refrigerant leaving the evaporator (sat. vapor) s_1	0.915	kJ/kg.K				
Entropy of refrigerant leaving compressor (assuming isentropic) s_{2s}	0.915	kJ/kg.K				
Enthaly of refrigerant leaving the compressor (assuming isentropic) h_{2s}	278.09	kJ/kg				
Enthalpy of refrigerant leaving the compressor (actual) h_2	282.63	kJ/kg				
Enthalpy of refrigerant leaving the condenser (sat. liquid) h_3	125.26	kJ/kg				
Enthalpy of refrigerant leaving the expansion valve h_4	125.26	kJ/kg				
Evaporator Load	37.6	kw	135458	kJ/hr	11	ton
Required referegrint mass flow rate of refrigerant (R134a)	0.30	kg/s	1066	kg/hr		
Required compressor power	8.98	kW	12.03	hp		
Condenser load	46.60	kW	167773	kJ/hr	13	ton
Cooling tower load	46.60	kW	167773	kJ/hr	13	ton
Required mass flow rate of cooling water	2.78	kg/s				
COP of the refrigeration system	4.2					

Fig. 21: Refrigeration Cycle Basic Design Calculations.

5. Development of performance charts for shell and tube heat exchanger

The evaportor and condenser equipment of the refrigeration cycle are shell and tube heat exchangers. These equipment need to be designed to provide the required heat transfer to the refrigerant such that the cycle operates per the requirements. During the basic design of the shell and tube heat exchanger, performance charts describing the variation of shell and tube heat exchanger's effectiveness with respect to NTU and capacity rate ratio are developed. Heat exchanger specifics and steady state performance can be described by three significant parameters. They are NTU, capacity rate ratio and thermal effectiveness. All these parameters are accounted for in the performance charts.

The heat exchanger effectiveness for a shell and tube heat exchanger having one shell pass and multiple tube passes may be given as [12].

$$\varepsilon_1 = 2\left\{1 + C_r + (1+C_r^2)^{\frac{1}{2}} * \frac{1+\exp\left[-(\mathrm{NTU})_1(1+C_r^2)^{\frac{1}{2}}\right]}{1-\exp\left[-(\mathrm{NTU})_1(1+C_r^2)^{\frac{1}{2}}\right]}\right\}^{-1} \quad (23)$$

And:

$$\mathrm{NTU}_1 = \frac{\mathrm{NTU}}{n} \quad (24)$$

Where n corresponds to the number of shell passes. Likewise, for a heat exchanger having mutiple shell passes and multiple tube passes the overall effectiveness may be given as [12]

$$\varepsilon = \left[\left(\frac{1-\varepsilon_1 C_r}{1-\varepsilon_1}\right)^n - 1\right]\left[\left(\frac{1-\varepsilon_1 C_r}{1-\varepsilon_1}\right)^n - C_r\right]^{-1} \quad (25)$$

Number of transfer units (NTU) is a dimensionless parameter that is widely used by process engineers and heat exchanger designers. NTU is a physically significant dimensionless parameter as it accounts for material characteristics, fluid characteristics, flow characteristics, theraml characteristics, heat exchanger size, fouling, etc.

Using Equations (23), (24), and (25), for a range of NTU varying from 0.1 to 10, the heat exchanger effectiveness was plotted as a function of capacity rate ratio. Capacity rate ratio is varied between 0 and 1. The NTU range selected for this study is the commonly seen working range in the process industries.

Consider Figures 22 through 24 describing the performance charts for shell and tube heat exchanger. With the available inputs and the desired output, the heat exchanger's required effectiveness can be readily computed. Thereafter, using the charts, a reasonable NTU and capacity rate ratio that would satify the required heat exchanger effectiveness can be selected. Subsequently, these parameters can be extrapolated using log mean temperature difference (LMTD) and/or ε-NTU approach to yeild the detailed design of the heat exchanger.

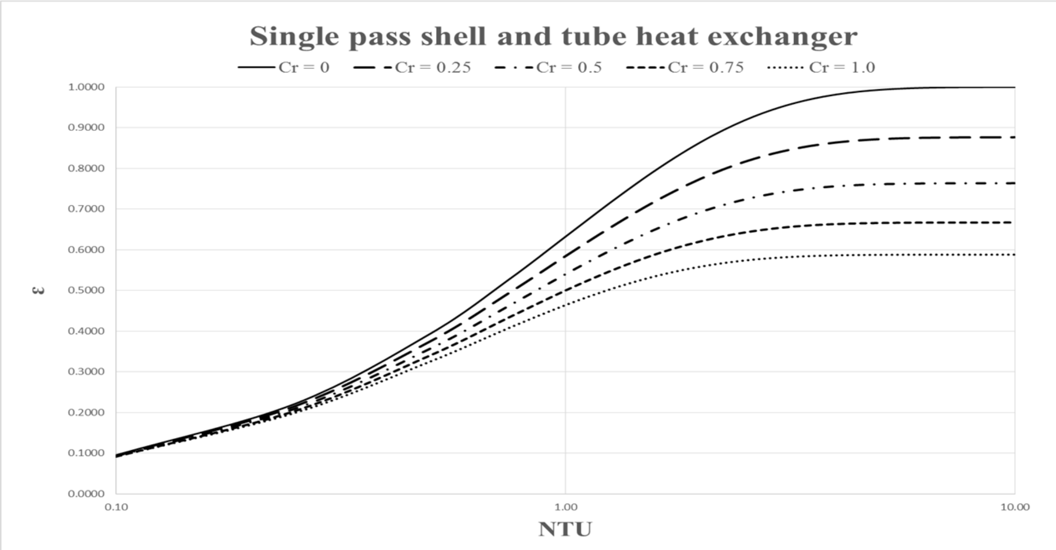

Fig. 22: Performance of A Single Pass Shell and Tube Heat Exchanger.

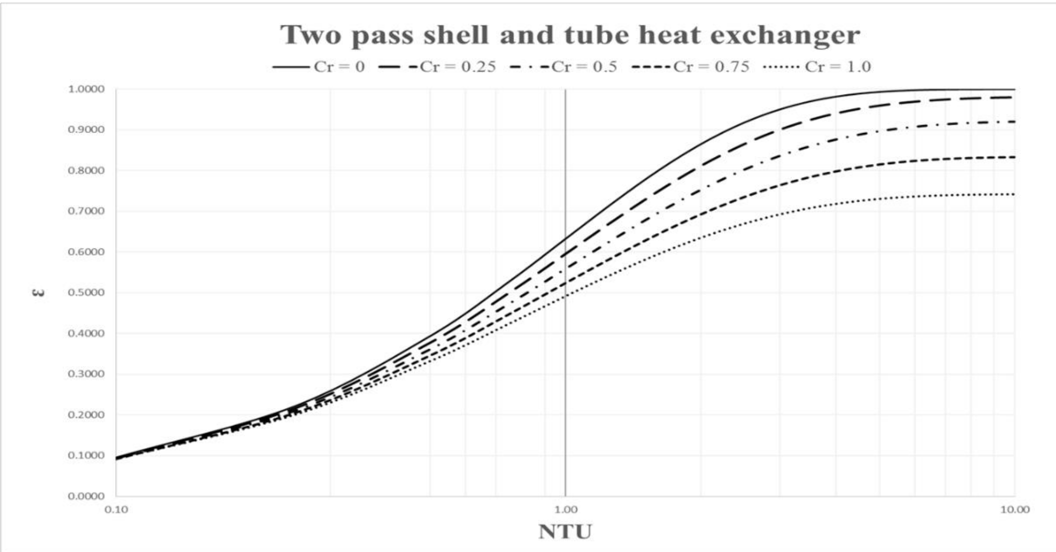

Fig. 23: Performance of A Two Pass Shell and Tube Heat Exchanger.

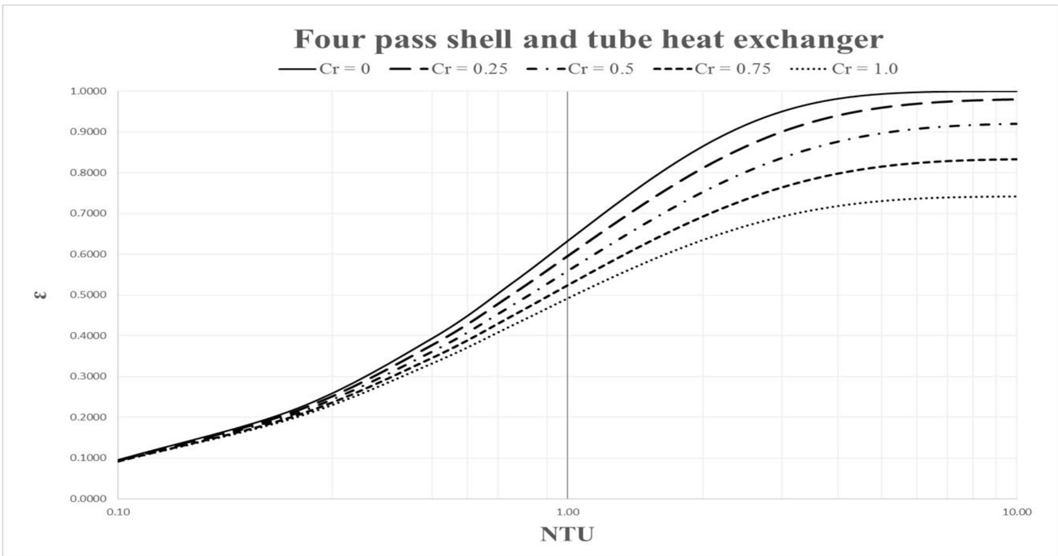

Fig. 24: Performance of A Four Pass Shell and Tube Heat Exchanger.

6. Design alternatives

The design alternatives must be considered while designing thermal systems.

7. Cross flow heat exchanger

Per the flow configuration, either parallel or counter flow can be considered for the analysis. Performance charts were developed for both parallel and counter cross flow heat exchangers. The difference in the requirements between the parallel and counter cross flow designs are described in Figure 25.

Parallel Cross Flow	Counter Cross Flow
ε = 0.63	ε = 0.63
NTU = 1.5	NTU = 1.2
$C_r = 0.25$	$C_r = 0.25$
# of HX passes = 4	# of HX passes = 4

Fig. 25: Requirements for Parallel and Counter Cross Flow Heat Exchanger.

From Figure 25, it can be clearly seen that both heat exchangers can deliver the heat transfer requirement. However, the counter cross flow heat exchanger requires a smaller NTU than that of a parallel cross flow heat exchanger. This means that the counter cross flow heat exchanger shall require a smaller surface area for heat transfer, which implies smaller space, less material, less weight and lower cost. Thus, counter cross flow heat exchanger is chosen for the project.

8. Refrigerant selection

Various hydrofluorocarbon refrigerants such as R134a, R404A and R41A0A are available in the market. Likewise, hydrochlorofluro-carbon refrigerants such as R22 and R408A are also available in the market. Out of all these refrigerants, the most popular, the readily available and the environmentally friendly refrigerant is R134a. Thus, R134a, Tetrafluoroethane refrigerant is selected for the project work.

9. Compressor selection

Different kinds of compressors can be used for operation of the refrigeration cycle. Centrifugal compressors are generally used for large scale applications, where the refrigeration capacities exceed 400 kW. Likewise, reciprocating compressors are not very suitable when there is moisture present in the refrigerant. Scroll compressors are suitable for producing refrigeration capacities around 15 kW. The selection of an appropriate compressor has enough merit to be considered as a stand alone paper and thus hasn't been described in this work.

10. Evaporator and condenser selection

Both evaporator and condenser equipment of the refrigeration cycle are heat exchangers. In the evaporator, refrigerant exchanges heat with the chilled water and whereas in the condenser, the refrigerant exchanges heat with the cooling water. Either a double pipe heat exchanger or a shell and tube heat exchanger is suitable for the given application. Since the required rate of heat transfer is large in both the evaporator and condenser a shell and tube heat exchanger is considered for the given application.

11. Basic design of the cooling system

The cooling system is intended to cool air by using chilled water. The basic design of a cross flow heat exchanger, the refrigeration cycle to provide the chilled water, the basic design of evaporator and condenser equipment were all considered in this work. Further, an analytical model was developed to partially simulate the functioning of the cooling system. In the development of the basic design of the heat exchanger equipment, performance charts were developed. Performance charts describe the performance of the heat exchanger in terms of significant dimensionless parameters. The charts report the heat exchanger effectiveness in terms of capacity rate ratio and number of transfer units. Capacity rate ratio and number of transfer units encompass the physical, the thermal and material characteristics of the heat exchanger. Likewise, they also account for flow configuration and fouling. In the design of any heat exchanger, it is very essential to develop the equipment such that it delivers the required heat transfer and yet is of light-weight, cheap, occupies less space, easy to manufacture, easy to maintain, consumes less material, etc. The parameter NTU accounts for all of the above-mentioned characteristics. Therefore, in the development of the heat exchanger, it is first essential to optimize the NTU and thereafter the detailed design of the heat exchanger can be developed.

Considering performance charts as described from Figures (6) through (16) and Figures (22) through (24), it can be clearly seen that increasing NTU indefinitely doesn't enhance the heat transfer, or in other words the heat exchanger effectiveness. From the basic principles, it could be assumed that increasing the surface area (or

NTU, as NTU is a function of heat exchanger surface area) increases the rate of heat transfer. However, by considering the performance charts, it can be clearly seen that increasing the surface area increases heat transfer only until a certain limit. Beyond that threshold limit, increasing the surface area or in other words, NTU is useless as the heat exchanger effectiveness becomes almost constant thereafter. A heat exchanger having a lower NTU than the required will not deliver the required heat transfer and likewise, a heat exchanger having a NTU more than the required will have more material, more weight and shall most certainly incur additional cost. Hence, selecting or optimizing the NTU is an important activity in the design of the heat exchanger.

This work considered the development of performance charts for both cross flow and shell and tube heat exchangers. For the cross flow heat exchanger, Domingos approach [3] was employed in the development of performance charts. Likewise, since evaporator and condenser equipment are shell and tube heat exchangers, performance charts were also developed for those heat exchangers by using explicit relations provided in [12].

Using the equations presented, an analytical model was developed in the design of the cooling system. The model helped to develop the basic design for the cross flow heat exchanger, the design of the refrigeration cycle, the basic design for the evaporator and the condenser equipment. Figure 26 describes the basic design of the cooling system.

BASIC DESIGN OF CROSS FLOW HX

Item	Description
Number of transfer units, NTU	1.2
Capacity rate ratio, C_r	0.75
Heat exchanger effectiveness, ε	0.63
Flow configuration	Counter
External fluid	Air
Tube side fluid	Chilled water
Mass flow rate of air	4.8 kg/s
Mass flow rate of chilled water	1.5 kg/s
Air inlet tempreture	22.4°C
Required air discharge tempreture	14.6°C
Chilled water inlet temperature	10°C
Chilled water discharge temperature	15.9°C
Cross flow heat exchanger load	37.6 kW

BASIC DESIGN OF REFRIGERATION CYCLE

Item	Description
Refrigerant	R134a
By pass factor, BP	0.2
Apparatus dew point temperature, ADP	9.025°C
Evaporator temperature	8.93°C
Evaporator pressure	4 bar
Condenser temperature	52.43°C
Condenser pressure	14 bar
Mass flow rate of refrigerant	0.30 kg/s
Compressor power	13 hp
Evaporator load	11 ton
Condenser load	13 ton
Cooling tower capacity	13 ton
Mass flow rate of cooling water	2.8 kg/s
COP	4.2

BASIC DESIGN OF CONDENSER

Item	Description
Number of transfer units, NTU	0.5
Capacity rate ratio, C_r	0
Heat exchanger effectiveness, ε	0.31
Flow configuration	NA
External fluid	R134a
Tube side fluid	Cooling water
Mass flow rate of refrigerant	0.30 kg/s
Mass flow rate of cooling water	2.8 kg/s
Condenser temperature	52.43°C
Condenser pressure	14 bar

BASIC DESIGN OF EVAPORATOR

Item	Description
Number of transfer units, NTU	1.75
Capacity rate ratio, C_r	0
Heat exchanger effectiveness, ε	0.78
Flow configuration	NA
External fluid	R134a
Tube side fluid	Chilled water
Mass flow rate of refrigerant	0.30 kg/s
Mass flow rate of chilled water	2.3 kg/s
Evaporator temperature	8.93°C
Evaporator pressure	4 bar

Fig. 26: Basic Design of the Cooling System.

12. Conclusions

The paper concentrates on the development of a basic cooling system to cool the “quench” air used in the production of nylon 6,6 polymer. The air is pretreated in an air washer and is available at 72ºF. The air is required to be cooled to 58ºF. The heart of the cooling system is the refrigeration cycle that produces chilled water. Chilled water is used as a secondary refrigerant and is used for cooling air to the desired level.

As described in Figure 1, the cooling system is sub divided into three subsystems, such as the quench system, the main refrigeration system and a cooling water system. The main component of the quench system is the cross flow heat exchanger. The basic design of the cross flow heat exchanger was developed in this work. Therein, performance charts using Domingos approach [3] were developed to come up with the basic design of the heat exchanger. Using performance charts, the required number of transfer units (NTU) and capacity rate ratio (Cr) were estimated. These two parameters can readily be extrapolated to complete design for the cross flow heat exchanger. This has been described in detail in [1]. During the basic design of the cross flow heat exchanger, both parallel and counter flow configurations were considered. Therein, it was seen that the required NTU for the counter cross flow heat exchanger was lower than that of a parallel cross flow heat exchanger. This means that the counter cross flow heat exchanger will require smaller surface area, which means lower capital cost, lighter weight, etc.

Likewise, a refrigeration cycle was developed to provide chilled water for the quench system. A standard, readily available and an environmentally refrigerant was chosen for the application. The operating parameters for evaporator, condenser and compressor were estimated. It must be recognized that evaporator and condenser are heat exchangers where the refrigerant exchanges heat with the other fluids. Typically, evaporator and condenser are all shell and tube heat exchangers. Therefore, in the design of evaporator and condenser equipment, performance charts were developed for shell and tube heat exchangers. Since the refrigerant undergoes phase change in both these equipment, the choice of flow configuration is not of importance. Using performance charts, the basic design of the evaporator and condenser equipment were estimated as well. From the basic design the detailed design can be devloped by using the techniques presented in [2] and [12]. As mentioned, the required pumping system for the cross flow heat exchanger and for the condenser cooling water are not detailed in this work. The techniques presented in [11] can be readily employed in the development of the pumping system.

The performance charts presented in this work will be valuable for design engineers during the initial development of the heat exchanger. Estimating NTU is a crucial activity during the development of any heat exchanger equipment. The process is usally time consuming as it requires detailed calculations. Performance charts will guide the engineers in quickly estimating the required NTU for the heat exchanger. Likewise, NTU is a direct function of heat exchanger surface area and it accounts for all the physical and thermal properties of the heat exchanger. Thus choosing the right NTU will help in optimizing the capital cost of the heat exchanger. Likewise, capacity rate ratio has a direct impact on the operating cost of the heat exchanger as it is a function of mass flow rate of the fluids. Therefore, these charts will certainly support in the design process of heat exchanger. In-addition all cooling systems must be developed such that it can handle varying refrigeration loads. When the refrigeration load varies, the operating set points of the heat exchanger will also vary. The field engineers can use performance charts as a guide and can estimate ahead of time whether the equipment is capable of handling such variations.

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