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Analysis And Design of Heat Exchanger at PT. X For Process Optimization.

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ABSTRACT

Heat Exchanger is a device used to transfer heat between fluids to get the temperature operating conditions as needed. This research was conducted to obtain Heat Exchanger design calculations to solve problems and adjust the needs of PT X. The design of the Heat Exchanger is obtained by the Double Pipe Heat Exchanger type with a heat transfer area of 7.37 ft2. The hot fluid, Condensate Steam, will fill the inner pipe while the cold fluid in the form of refrigerant water will fill the annulus pipe. Superheated steam has a temperature higher than its boiling point at the same pressure. Steam condensate is steam that has cooled and condensed back into water after being used to do work or after losing heat.The length and number of hairpin heat exchangers are 10 ft and 1 hairpin. The height of the heat exchanger is 30cm. The designed heat exchanger uses turbulence flow type to optimize and streamline heat transfer. The performance and capability of the heat exchanger obtained Overall Dirty Coeficient Heat Transfer (Ud) 250 Btu/hr ft2 oF, Overall Clean Coeficient Heat Transfer (Uc) 558.027 Btu/hr ft2 oF, Fouling Factor (Rd) 0.002. The pressure drop values for both annulus and innerpipe are 0.99 psi and 2.321 psi, which do not exceed the safe limit. The friction values obtained on the innerpipe and annulus are 0.006 and 0.0105.

Keywords: Heat Exchanger, Double Pipe Heat Exchanger, Heat Transfer
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1. Introduction

Heat exchangers are widely used in the industrial world such as the chemical industry and others. Its use is very much needed and plays a very important role in managing heat against fluids to achieve the required operating conditions. Heat exchanger or commonly known as Heat Exchanger is a device that becomes a place for heat transfer from hot fluid to cold fluid with a temperature difference so that heat will exchange and cause heat loss from a fluid[1]. Many types of heat exchangers are widely used in industry depending on the needs reviewed by calculation. Commonly used heat exchangers are Double Pipe Heat Exchanger, Shell & Tube Heat Exchanger and others[2]. Double Pipe Heat Exchanger is currently commonly used in addition to shell & tube heat exchangers by consisting of two pipes of different diameters where the small pipe is inside the large pipe[3]. In this study, DPHE was chosen, because the flow of hot fluid and cold fluid flowing in separate pipes allows effective heat transfer between the two fluids, especially when counter-flow is used.

At PT.X there is a condensate water storage tank that has regulated operating temperature. PT.X efforts in regulating the temperature by adding ordinary water to keep stabilizing the operating temperature. In an effort to maintain its operating temperature, it always uses ordinary water and still cannot predict the exact quantity of condensate water. In addition, the use of ordinary water to add to the condensate water storage tank to stabilize the temperature as needed is considered a waste of ordinary water which can be used for other purposes. Therefore, the use of ordinary water needs to be stopped from mixing and a solution idea is given not to use ordinary water in direct contact.

Thus, the design of Heat Exchanger can be a solution idea in conditioning operations as needed where condensate water can reach its operating temperature and ordinary water can still be used for other purposes without mixing between condensate water and ordinary water to achieve these operating conditions, this design research is based on PT X data to condition the needs of the company.

1.1 Heat Exchanger

Heat exchanger is a device used to transfer heat between two or more fluids that have different temperatures to get outputs or products that have the desired temperature[4]. The flow direction of the heat exchanger can be adjusted and there are two types of flow, namely counter current flow and parallel flow. Counter current flow is an opposite type of flow where hot and cold flow enters through the opposite end or direction while parallel flow is a unidirectional flow type where hot and cold flow enters and aligns its flow to exit the heat exchanger[5][6]. The use of heat exchangers aims to overcome several problems such as fraction factors, maximizing effectiveness, heat transfer, and others[7].

Factors that affect the performance of heat exchangers are pressure drop, LMTD, fouling factor, and efficiency value in terms of Q material and coolant where the Q value is equal or almost the same, the efficiency is getting better[8].

1.2 Heat Transfer in Heat Exchanger

Heat transfer in heat exchangers can be more optimal and efficient by utilizing the turbulence mechanism in the flow in the heat exchanger[9]. In laminar flow, the fluid flows in a regular and smooth manner in parallel layers that do not mix with each other. This flow occurs at low speeds and high fluid viscosity. In turbulent flow, the fluid moves at high speeds, and the flow becomes chaotic with the movement of eddies and vortices that mix the fluid. This flow often occurs at high speeds or in fluids with low viscosity. The purpose of heat transfer in heat exchangers is to heat or cool a fluid to the desired temperature, save energy in the next process, and others[1]. The rate of heat transfer in heat exchangers is influenced by physical properties such as viscosity, thermal conductivity, heat capacity, surface properties, and others[10].

1.3 Heat Exchanger Design in General

In designing a heat exchanger, several general stages are needed which are the basis for designing to the selection of the type of heat exchanger that will be used later. Here are some general stages in heat exchanger design.

- Calculation of Ud Correction

The calculation of Ud (Overall Dirty Coeficient Heat Transfer) correction is intended to determine initially whether the heat exchanger is feasible to be established or not based on the accuracy between the selection of the Ud range in table 8, Kern according to the type of hot or cold substance with Ud correction produced using the equation. The calculation of Ud correction is influenced by the value of heat rate (Q), heat transfer area (A), and ΔT LMTD. Thus, it is necessary to do trial and error, especially on dimensional configurations that need to be seen based on Ud Correction until Ud Correction is correct or 1% tolerance based on field data.

$$Ud = \frac{Q}{A \times \Delta T \text{ LMTD}}$$

The selection of dimensions will affect the value of A and also Ud Correction, it is necessary to do trial and error with the help of a list of dimensional tables[5].

- Selection of Heat Exchanger Type

The selection of the type of heat exchanger is based on the value of A which is also an elimination of many types of heat exchangers. If the value of A is smaller than 200 ft2 then the most appropriate heat exchanger is a double pipe heat exchanger[5].

- Calculation of Heat Exchanger Capability and Evaluation

With the dimensions and type of heat exchanger that will be used based on the feasibility of Ud Correction, the performance test can be carried out, the performance test is carried out both on the inner and outer sides (annulus). The following are general performance tests: flow area, mass velocity, Reynold's number, hi (inner heat transfer coefficient), ho (outer heat transfer coefficient), hio (heat transfer coefficient between hi and ho), Uc (Overall Clean Coeficient Heat Transfer), Fouling Factor.

This calculation determines the feasibility of using the heat exchanger with the calculation of the Fouling Factor. If the calculation value of the fouling factor is greater than the limit value, then the heat exchanger is suitable for use and can be continued with the calculation of Pressure Drop[5].

- Calculation of Pressure Drop Heat Exchanger

This calculation is the last stage as far as the possibility of pressure drop.This calculation also calculates the inner and outer sides (annulus). The calculation contains the Fanning equation to determine the pressure drop (ft) and pressure loss (ft), as well as the HE flow velocity to density, to pressure drop in units of pressure[5]. Pressure drop is a pressure drop caused by friction against the flowing fluid where the pressure drop in the shell or tube is not allowed to exceed the allowable pressure drop limit[11].

2. Methodology of Research

Research was conducted at PT X to obtain information and understand how the working principle of a Plant explores and seeks information, analyzes and identifies problems, comes up with a solution idea, takes field data, performs calculations for problem solving with a literature study approach, completion to reach a conclusion. This Heat Exchanger design is adjusted to the needs of PT X where it is expected that condensate steam must be cooled to a temperature of 60oC. Therefore, it is necessary to observe and collect data and assemble a completion procedure diagram. Before being solved using a theoretical approach, it is necessary to understand the plant scheme to identify problems. In solving it, the following theoretical approach will be used.

2.1. Calculation of Q, Δ LMTD, A, and UD Correction

$$Q_{material} = Qh = Q_{out} - Q_{in}$$

$$Q = m\ (material) \times Cp \times \left(T_{\frac{out}{in}} - T_{ref}\right)$$

$$\Delta T\ LMTD = \frac{\Delta t_2 - \Delta t_1}{\ln \frac{\Delta t\ 2}{\Delta t\ 1}}$$

$$Ud = \frac{Q}{A \times \Delta T\ LMTD}$$

2.2. Selection of Heat Exchanger and dimensions

Selection of heat exchanger type if the value of heat transfer area is less than 200ft2 then double pipe heat exchanger type can be used. Pipe dimensions can be seen in Kern and Brownell[5][12]. DPHE is more suitable for applications with smaller or medium fluid flows. Compared to STHE, which is more ideal for large capacities and larger fluid flows, DPHE provides a more efficient solution for less large flow volumes.

2.3. Calculation of Heat Exchanger Performance and Capability

The performance and capability of heat exchangers include flow area, equivalent diameter, mass velocity that can be passed on the shell and tube, flow type, shell and tube heat transfer coefficient[5].

2.4. Calculation of Uc, Rd, Pressure Drop and Friction

The results of the performance and ability of the heat exchanger will get the Uc value to determine the value of the net heat transfer coefficient, pressure drop to determine the pressure drop caused by friction or friction[5].

$$Uc = \frac{hio \times ho}{hio + ho}$$

$$Rd_{(dep)} = \frac{Uc - Ud}{Uc \times Ud}$$

3. Result and Discussion

3.1 Condenser Calculation

Table 1. Operating conditions

Inlet mass (W)	:	1100 kg/hour	:	2425,08 lb/hour
Inlet material temperature (t_1)	:	95°C	:	194°F
Material exit temperature (t_2)	:	60°C	:	140°F
Inlet coolant temperature (T_1)	:	5°C	:	41°F
Cooling temperature out (T_2)	:	63,25°C	:	145,85°F
Ref temperature (T_{ref})	:	25°C	:	77°F
Cooling requirement (m)	:	564 kg/ hour	:	1243,406 lb/ hour
Heat carried by the coolant	:	130737,2 btu/ hour		
Heat acting on the system	:	130744,6 btu/hour		

Cp of water (5oC) = 1.009 Btu/lboF [13]

Cp of water (90oC) = 0,998 btu/lboF [13]

Value range Ud Water-BK =250-500 Btu/hour.ft2oF

Ud Selected = 250 Btu/hour.ft2oF [5]

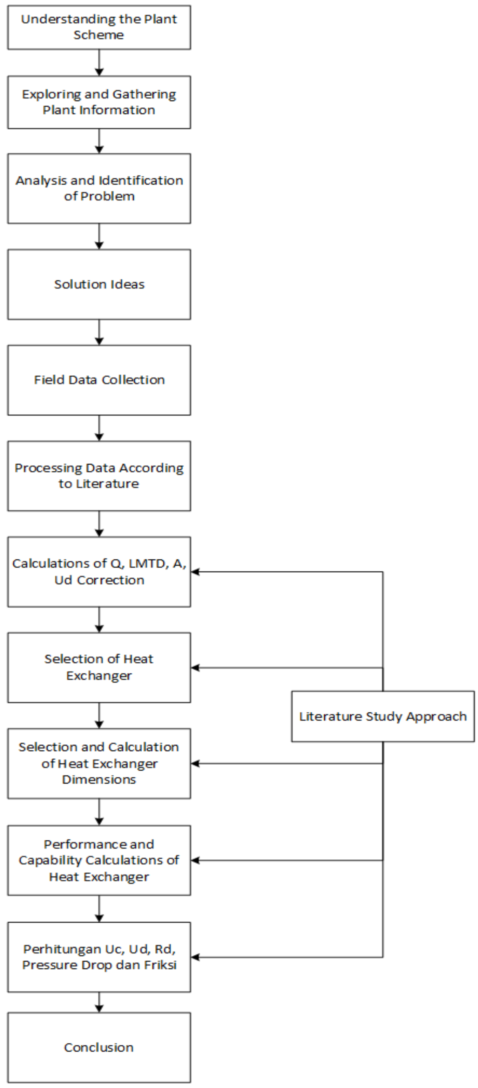

Figure 1. Research Flow Chart

Table 2. Calculating Δ LMTD

Hot Fluid (°F)	Category	*Cold Fluid* (°F)	(°F)	
194	*High Temp*	145,22	48,78	Δt 2
140	*Low Temp*	41	99	Δt 1
-70,95				Δt 2-Δt 1

$$\Delta T\ LMTD = \frac{\Delta t\ 2 - \Delta t\ 1}{\ln \frac{\Delta t\ 2}{\Delta t\ 1}}$$

$$\Delta T\ LMTD = \frac{-70{,}95}{\ln \frac{48{,}78}{99}}$$

$$\Delta T\ LMTD = 70{,}95°F$$

3.2 Calculating Q of materials and coolant

- Q material

  Qh = Q out – Q in

  Qout (Q outlet Cooler) = $m_{(BK)}$ x Cp x ΔT

  Q out = 2425,08 lb/hour x 0,998 btu/lb°F x (194°F - 77°F)

  Q out = 152526,7 Btu/hour

  Q in (Q inlet *Cooler*) = $m_{(cs)}$ x Cp x ΔT

  Q in = 2425,08 lb/hour x 0,998 btu/lb°F x (194°F-77°F)

  Q in = 283264 Btu/hour

  Qh = -130737 Btu/hour

  A negative Q value indicates an exothermic reaction where the heat of the water is released and transferred to the environment. [14].

  Qh = 130737,2 Btu/hour

- Q Coolant

  Q CW = m(CW) x Cp x ΔT

  Q CW = 1243.406 lb/hourr x 0.998 Btu/lboF x (145.22oF - 41oF)

  Q CW = 130744.6 Btu/hour

  A positive Q value indicates an endothermic reaction where water heat is absorbed and transferred to the system[14].

  In other words, the opposite Q value causes a mutual release and absorption reaction so that there is an element of heat transfer where the Q of the material will be absorbed by the Q of the coolant.

3.3 Calculating the area of heat transfer

$$A = \frac{Q}{Ud \text{ x } \Delta T \text{ LMTD}}$$

Heat Transfer Area, A (BK-CW)

$$A_{bahan} = \frac{130744{,}6 \ \frac{Btu}{jam}}{250 \frac{Btu}{jam}.ft^2\ {}^0F \text{ x } 70{,}95^0F}$$

$$A_{bahan} = 7{,}37 ft^2$$

$$A_{bahan} = \frac{130737{,}2 \frac{Btu}{jam}}{250 \frac{Btu}{jam}.ft^2\ {}^0F \text{ x } 70{,}95^0F}$$

$$A_{bahan} = 7{,}37 ft^2$$

3.4 Selection of Heat Exchanger and its dimensions

Based on information from the book Kern, if the heat transfer area is less than 200 ft2 (<200 ft2) then the type of heat exchanger that is good to design is Double Pipe Heat Exchanger (DPHE)[5]. In determining the dimensions of the DPHE, it is necessary to correct the value of Ud by adjusting the dimensions of the DPHE to ensure that the type used and the dimensions are continuously checked until Ud's target expectations are met. Double Pipe Heat Exchanger will be very impactful in changing heat transfer with the influence of the Reynold number so that it affects the increase in heat transfer rate and the higher the flowing flow discharge will accelerate the heat transfer rate[15].

Dimensions are measured based on the outermost and largest IPS values of Q and A in order to have the same length and be able to be at the largest possible Q especially when both loads (material and coolant) enter the HE.

Based on Kern's book, it is known that the flow area in the innerpipe is larger than the annulus, so the cold fluid whose mass is greater is placed in the annulus.

Table 3. Selection of Heat Exchanger and its dimensions

Exchanger, IPS		Schedule Number	Diameter Annulus		Diameter Inner Pipe		Surface per lin ft				Unit
Annulus	Inner		OD	ID (D2)	OD (D1)	ID (D)	Outside ft2/ft		Inside ft2/t		
1	1/2	40	1,320	1,049	0,840	0,622					Inch
			0,110	0,087	0,070	0,051	0,344	0,274	0,22	0,163	ft
			3,351	2,663	2,132	1,579					cm

Note :

OD = Outer Diameter

ID = *Inner* Diameter

- Total pipe length (ft) = A/Surface per lin ft

$$= \frac{7{,}37 \ ft^2}{0{,}344 \ \frac{ft^2}{ft}}$$

$$= 21{,}43 \text{ ft}$$

- Length of pipe to be designed = 10 ft
- Hairpin Calculation

One hairpin consists of 2 pipes (n = 2), then the number of hairpins = total L / n.L

Number of Hairpin

$$= \frac{21.43 \text{ ft}}{(2 \text{ x } 10 \text{ ft})}$$

$$= 1.07 = 1 \text{ buah hairpin}$$

With a designed pipe of 10 ft, there is a remaining 1.42 ft and will be used as the pipe to be bent.

- HE height = Bending height of the residual pipe

HE = 2 OD of annulus + Bending height.

Bending Height = half circumference of a circle

= (2 x L of remnant) / π

= (2 x 1.42) / π

= 27.68 cm

HE height = 2 OD of annulus + bending height

= 2 (3.35 cm) + 27.68 cm

= 34 cm

- Correction of Ud Value

Ud = Q/(A x ΔT LMTD)

= (130744.6 Btu/hr)/(7.37 ft^2 x 〖70.95〗^o F)

= 250 Btu/hr ft2 oF

Accuracy Ud = 100%

3.5 Evaluation of Heat Exchanger Performance & Capability

With the dimensions that have been selected, an evaluation of the performance and capabilities of the DPHE is obtained. It is known that the Flow Area in the Inner Pipe is larger than that in the Annulus, so the cold fluid whose mass is greater is placed in the Inner Pipe while the hot one in the Annulus because it has a smaller mass flow rate[5]. Thus Steam Condensate (SC) is in the Inner Pipe while Cooling Water (RW) is in the Annulus Pipe.

Table 4. Flow Area in Inner Pipe

D. Inner Pipe	:	OD (D1)	:	0,840 in	=	0,0700 ft
	:	ID (D)	:	0,622 in	=	0,0518 ft
D. Anulus	:	OD	:	1,320 in	=	0,1100 ft
	:	ID (D2)	:	1,049 in	=	0,0874 ft

- Correction Value hi(ho)c

$$hio\ x\ \text{hi x}\left(\frac{\text{IDp}}{\text{ODp}}\right)$$

$$= 1544{,}024\ \text{Btu/jam ft}^2\ \text{°F x}\left(\frac{0{,}0518\ \text{ft}}{0{,}0700\ \text{ft}}\right)$$

$$= 1143{,}313\ \text{Btu/jam ft2 oF}$$

- Calculating the Overall Net Heat Transfer Coeficient, Uc

$$Uc = \frac{hio\ x\ ho}{hio + ho}$$

$$\frac{1143{,}313\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}\ x\ 1090{,}067\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}}{1143{,}313\ \text{Btu/jam ft}^2\ ^\text{o}\text{F} + 1090{,}067\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}}$$

$$= 558{,}027\ \text{Btu/jam ft2 oF}$$

- Menghitung *Fouling Factor*, Rd

$$Rd_{(\text{abs})} = \frac{Uc\ -\ Ud}{Uc\ x\ Ud}$$

$$\frac{558{,}027\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}\ -\ 250\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}}{558{,}027\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}\ \ x\ 250\ \text{Btu/jam ft}^2\ ^\text{o}\text{F}}$$

$$= 0{,}00229\ (Rd_{\text{Deposited}})$$

$$Rd_{(\text{allowed})} = 0.001$$

Table 5. Performance & Designability Evaluation of DPHE, CS-RW

Inner Pipe (Hot Fluid –SC)	*Annulus* (Cold Fluid – RW)
1. *Flow Area*, Ap D = 0,0518 Ft Ap = π X (D^2/4) = π X (0,0518 Ft^2/4) = 0,00210 Ft^2 Diameter Ekivalen, De De = ($D_2^2 - D_1^2$) / D_1 $= \frac{(0{,}0874\ Ft)^2-(0{,}0700\ Ft)^2}{0{,}0700\ Ft}$ = 0,0391 Ft De *Annulus* = *Inner Pipe*	1. *Flow Area*, aa D_2 = 0,0874 ft D_1 = 0,0700 ft aa = ¼ π x ($D_2 - D_1$) = ¼ π x (0,0874 ft – 0,0700 ft) = 0,00215 ft^2 Diameter Ekivalen, De De = ($D_2^2 - D_1^2$) / D_1 $= \frac{(0{,}0874\ ft)^2-(0{,}0700\ ft)^2}{0{,}0700\ ft}$ = 0,0391 ft De *Annulus* = *Inner Pipe*
2. Mass Speed, Gp Gp = W/Ap $= \frac{2425{,}08\ lb/jam}{0{,}00210\ Ft^2}$ = 1150763,29 Lb/Jam Ft^2 Bilangan Reynold (Nre) Tc = ($T_1 + T_2$)/2 = (194°f+140°f)/2 = 167°f = 75°c μ = 0,3755 Cp, (Yaws, 1999) (1cp = 2,419088 Lb/Jam Ft^2) μ = 0,9082 Lb/Jam Ft^2 Nre = De X Gp/ μ $= \frac{0{,}0391\ Ft\ X\ 1150763{,}29\ Lb/Jam\ Ft^2}{0{,}9082\ Lb/Jam\ Ft^2}$ = 65646,40 (Turbulen)	2. Mass Speed, Ga Ga = m/aa $= \frac{1243{,}40\ lb/jam}{0{,}00215\ ft^2}$ = 578195,81 lb/jam ft^2 Bilangan Reynold (NRe) Tc = (T_1+T_2)/2 = (41°F+145,22°F)/2 = 93,11°F = 33,95°C μ = 0,7494 cP, (Yaws, 1999) (1cP = 2,419088 lb/jam ft^2) μ = 1,8128 lb/jam ft^2 NRe = De x Ga/ μ $= \frac{0{,}0391\ ft\ x\ 578195{,}81\ lb/jam\ ft^2}{1{,}8128\ lb/jam\ ft^2}$ = 12486,99 (Turbulen)
3. Calculating the Heat Transfer Coefficient, hi tc = 167°F J_H = 200 (Fig.24, Kern hal.834) Cp air = 0,998 Btu/lb°F (Yaws, 1999) k = 0,3578 Btu/(hr)(ft^2)(°F/ft) (Yaws, 1999) $hi = J_H\left(\frac{k}{De}\right)\left(\frac{Cp\ x\ \mu}{k}\right)^{\frac{1}{3}}$ hi = 1544,024 Btu/hour ft^2 °F	3. Calculating the Heat Transfer Coefficient, ho Tc = 93,11°F J_H = 70 (Fig.24, Kern hal.834) Cp air = 1,009 Btu/lb°F (Yaws, 1999) k = 0,3643 Btu/(hr)(ft^2)(°F/ft) (Yaws, 1999) $ho = J_H\left(\frac{k}{De}\right)\left(\frac{Cp\ x\ \mu}{k}\right)^{\frac{1}{3}}$ ho = 1090,067 Btu/hour ft^2 °F

Rd Allow is taken from the average type of water mixed in the kalimalang river based on the book Kern, (1950) p 108[5]. The Rd value obtained must be greater than Rd Allow (RdDeposited > Rd¬(allowed)). Rd values below the minimum limit will cause additional resistance to heat but if it is too large the Rd value can indicate the presence of scale or impurities as well [11]. Fouling can block the fluid flow path in the tube or heat exchanger surface. This reduces the fluid flow velocity and reduces the heat flow efficiency. When the flow is disturbed, the fluid becomes poorly distributed, which causes a decrease in the heat transfer rate.

The safe limit of pressure drop both inner pipe and annulus pipe is a maximum of 10 psi[5]. Pressure Drop is reviewed to determine the heat transfer performance where the smaller the pressure drop, the greater the value of the heat transfer coefficient in the heat exchanger[16]. A larger pressure drop causes a decrease in fluid flow under certain operating conditions, because the pump or compressor must work harder to overcome the resistance. This means more energy is required to pump or move the fluid through the system, leading to increased energy costs and decreased overall system efficiency.

Table 6. Pressure Drop and Friction Calculation

Inner Pipe (Hot Fluid – CS)	*Annulus* (Cold Fluid – CW)
1. Friksi, F Calculating Reynold's Number Re'p = (D x Gp)/μ $= \frac{0{,}0518 \text{ ft x } 1150763{,}29 \text{ lb/jam ft}^2}{0{,}9082 \text{ lb/jam ft}^2}$ = 65646,40 Friksi (per 3.7b hal 53) f = 0,0035 + (0,264/(DGp/μ)$^{0.42}$ = 0,0035 + (0,264/(65646,40)$^{0.42}$ = 0,006	1. Friksi, F Calculating Reynold's Number De' = D2-D1 = 0,0874 ft – 0,0700 ft = 0,0174 ft Re'a = (De' x Ga)/μ $= \frac{0{,}0174 \text{ ft x } 578195{,}81 \text{ lb/jam ft}^2)}{1{,}8128 \text{lb/jam ft}^2}$ = 5552,71 Friksi (per 3.7b hal 53) f = 0,0035 + (0,264/(De' Ga/μ)$^{0.42}$ = 0,0035 + (0,264/(5552,71)$^{0.42}$ = 0,0105
2. Fanning Equation, ft g = 9,8 m/s^2 = 4,25 x10^8 ft/hour^2 ρ = 61,202 lb/ft^3 $\Delta Fp = \frac{4xfxGp^2L}{2g\rho^2De}$ = 5,462 ft	2. Fanning Equation, ft g = 9,8 m/s^2 = 4,25 x10^8 ft/hour^2 ρ = 63,62 lb/ft^3 $\Delta Fa = \frac{4xfxGa^2L}{2g\rho^2De}$ = 2,245ft
3. *Pressure Drop*, psi ΔPp = (ΔFp x ρ)/144 = 2,321 psi	3. *Pressure Drop*, psi V = Ga/3600 x ρ $= \frac{578195{,}81 \text{ lb/hour ft}^2}{3600 \times 63{,}62 \text{ lb/ft}^3}$ = 2,524 ft/s ΔF_{loss} = 3 x V^2/2g = 3 x (2,524ft/s)^2/2(4,25 x10^8 ft/hour^2 = 2,25 x 10^{-8} ft ΔPa = (ΔFa+ΔF_{loss}) ρ)/144 = 0,99 psi

4. Conclusion

This research provides recommendations for heat exchanger design to be applied is the Double Pipe type with Counter Current Flow. In this heat exchanger analysis, it has been adjusted that the Q CS value is the same as Q CW although the difference is slight due to lack of accuracy in measuring flow rate and temperature. With this design, to cool the condensate steam using 5oC cooling water with cooling water output reaching 62.9oC. The length of the heat exchanger is 10 ft with a height of 34 cm. Inner Pipe diameters are OD and ID of 0.07 ft and 0.05 ft. Annulus diameter is OD and ID of 0.11 ft and 0.0874 ft. Uc (Overall Clean Coeficient Heat Transfer) amounted to 558.028 Btu/hour ft2 oF with an impurity factor (Rd) of 0.002.

5. Acknowledgments

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