INDIVIDUAL DESIGN PROJECT: HEAT EXCHANGER DESIGN

EMS623U: Heat Exchange and Waste Minimization

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Introduction

A heat exchanger enables energy transfer by allowing two fluids of different temperatures to come into contact with one another. This system allows the transfer of heat from the hotter fluid to the cooler fluid, leading to the cooling of the hot fluid and the heating of the cold fluid [1]. The primary type is heat exchangers is shell-and-tube heat exchangers, also known as single-pass, cross-flow heat exchanger. They involve tubes within a cylindrical shell, through which one fluid passes the tubes and another flows around them. Heat is exchanged between the fluids, resulting in the cooling of one and the heating of the other [2]. Shell and tube heat exchangers are widely utilised in industrial applications due to their superior efficiency in transfer of heat, even under extreme pressure and temperature conditions. They are adaptable, accommodating diverse fluids, expandable for different system dimensions, and simple to maintain. Their compact design maximises area efficiency, making them flexible in sectors such as electricity generation and chemical processing. However, they have disadvantages, including a substantial footprint, complicated design, and elevated manufacturing expenses. Maintenance is essential to prevent leakage and corrosion, which can diminish efficiency. Furthermore, their efficiency declines with minimal temperature differentials between fluids [2].

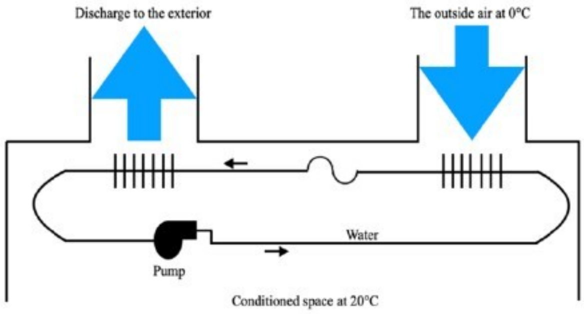

Figure 1. Diagram of the air-conditioning plant for which the heat exchanger is being designed [3].

This report presents the design of a heat exchanger for a run-around air-conditioning system in an office building. To minimize heating costs and prevent substantial heat loss, the system recovers heat from the discharged stale air (90 m^3/min) and uses it to preheat the incoming fresh air, transferring a total of 10 kW of thermal energy. The focus of this project is the design of a shell-and-tube heat exchanger with fins on the air side, positioned on the discharge side of the system. The goal is to optimize the exchanger's geometry and size to maximize heat recovery efficiency. The Stale air at 20°C heats the water, while the water heats the fresh air entering at 0°C as seen in Figure 1. To enhance heat transfer, fins are added to the shell side, improving the shell side heat transfer coefficient, as air is a poor thermal conductor. This study relies on several assumptions in order to simplify study. This requires the assumption that the heat exchanger operates in steady state and that the tubes are completely insulated. These assumptions facilitate the calculations required to determine the tube characteristics and the physical properties of water and air. The fins were presumed to enhance the shell side coefficient by a factor corresponding to the ratio of the surface area of the finned tube to that of the unfinned tube [3].

A comparative analysis also took place where heat transfer was calculated for a tube in tube and also for the shell and tube configuration.

Methodology

To find the optimum heat exchanger size, the physical properties of water and air were initially specified. In addition to this a material was selected for the exchanger and it was selected to be stainless steel AISI 304 [4]. Following this, the outlet temperature of the air was determined using Equation (1) [3].

$$Q = \dot{m}_a c_{p_a}(T_{a\,in} - T_{a\,out}) = \dot{m}_w c_{p_w}(T_{w\,out} - T_{w\,in}) \quad (1)$$

The $\dot{m}_a$ was subsequently calculated by applying Equation (2), employing the given volumetric flow rate and the density of air at the initial temperature [10].

$$\dot{m} = \rho\dot{V} \tag{2}$$

The temperature of the water flowing out towards the inflow side was calculated using the physical parameters of water. Since the heat transfer will be at its best at this velocity without the tube experiencing excessive pressure, the water's velocity was considered to be 1. Equation (2) was also used to establish the tube's starting dimensions, in order to compute the water's mass flow rate.

$$\Delta T_{LM} = \frac{(T_{a\,in} - T_{w\,out}) - (T_{a\,out} - T_{w\,in})}{\ln\left(\frac{(T_{a\,in} - T_{w\,out})}{(T_{a\,out} - T_{w\,in})}\right)} \tag{3}$$

Equation 3 was used to get the Log Mean Temperature Difference (ΔT_{LM}) between the air and water temperatures [8]. The Correction Factor Graph for a single pass cross flow exchanger (Figure 2) with both fluids unmixed had the greatest value, therefore this was used to determine the correction factor (F).

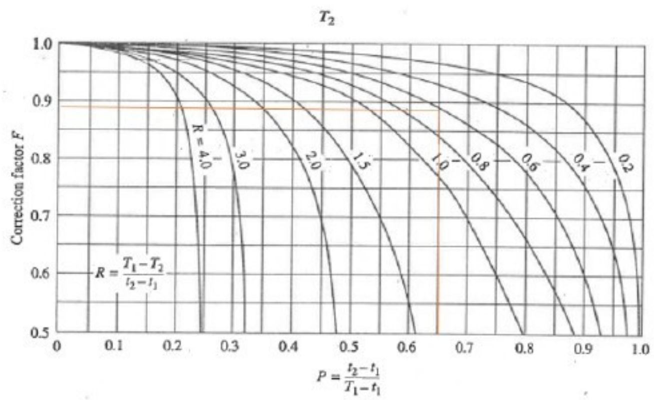

Figure 2: Correction Factor Graph for a single pass cross flow exchanger [3].

R and P were determined using Equations 4 and 5 below and in order to develop a heat exchanger design that could be utilised as a reference for the real heat exchanger, a total heat transfer coefficient of 200 W/m2K was initially considered [2].

$$R = \frac{T_1 - T_2}{t_2 - t_1} \qquad P = \frac{t_2 - t_1}{T_1 - t_1} \tag{4\&5}$$

An assumed area of the heat exchanger was obtained with Equation 6 .

$$Q = FUA\Delta T_{LM} \tag{6}$$

Equation 7 was then utilised to get the number of tubes, which was subsequently applied to the bundle diameter.

$$N_t = \frac{A}{\pi d_o L_t} \tag{7}$$

The tube side coefficient a_t and shell side coefficient A_s of the heat exchanger need to be calculated in order to determine the overall heat transfer coefficient of the initial design. Equation 10 illustrates the calculation process. Reynolds and Prandtl numbers for the tube side must be determined in order to get them; this was done using equations 8 and 9 [9][11]. It was discovered that the Reynolds number was 3225.1, indicating turbulent flow. Then equation 8 and 9 were used again when equivalent diameter was solved.

Subsequently, jh was used to find pressure drop fir shell-side as shown in equation 11. It was also calculated that $\frac{\mu}{\mu_w} = 1$.

$$P_r = \frac{c_p \mu}{k} \qquad R_e = \frac{\rho u d}{\mu} \qquad a_t = 0.023 \times {R_e}^{0.8} {P_r}^{0.33} \times (\frac{\mu}{\mu_w})^{0.14} = \frac{h_i d}{k_f} \qquad (8,9\&10)$$

$$j_h = \frac{h_t d}{k_f R_e {P_r}^{0.33}\ 1^{0.14}} \qquad (11)$$

The baffle spacing l_b becomes the tube's length because it was believed that there would be no baffles. Then, using equation 12, the cross-flow area A_s was determined and D_s was solved using equation 13. A thickness of 5 mm was taken [4]. Subsequently, equation 14 was then used to get the velocity.

$$A_s = \frac{(p_t - d_o) D_s l_b}{p_t} \qquad D_s = d_e - material\ thickness u_s = \frac{\dot{m}}{A_s} \qquad (12,13\&14)$$

$$G_s = \frac{\dot{m}_a}{A_s} \qquad u_s = \frac{G_s}{\rho} \qquad (15\&16)$$

Equation 15 and 16 were then used to find G_s and u_s to be used to find diameter D_e as shown in equation 17.

$$D_e = \frac{1.1}{d_0}(p_t^2 - 0.917 d_t^2) \qquad (17)$$

$$R_e = \frac{G_s D_e}{\mu} \qquad N_u = \frac{a_s D_e}{k_f} = J_k R_e P_r^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad (18\&19)$$

Then, the alpha ideal coefficient a_s can be solved using equation 19. Following this the pressure drop ΔP_s can be solved in equation 20. Following, equation 21 was used to get the total heat transfer coefficient. The k_{metal} of stainless steel AISA 304 was taken.[5]

$$\Delta P_s = 8 j_h \frac{D_s}{d_e} \frac{L}{L_b} \frac{\rho u_s^2}{2} \left(\frac{\mu}{\mu_w}\right)^{0.14} \qquad \frac{1}{U} = \frac{1}{a_s} + \frac{d_o \ln(\frac{d_o}{d_i})}{2 k_{metal}} + \frac{d_o}{d_i} \times \frac{1}{a_t} \qquad (20\&21)$$

Finally, the new optimized area was found as shown below and the steps were repeated.

$$new\ area = \frac{Q}{UF\Delta T_{lm}} \qquad (22)$$

To work out the heat transfer coefficients the 2 equations below were used.[6]

$$h_{fins\ known\ dimentions} = \frac{0.134 R_e^{0.681} P_r^{0.33} \left(\frac{P_f - w_f}{L_f}\right)^{0.2} \left(\frac{P_f}{W_f}\right)^{0.1134} k}{d_s} \qquad (23)$$

$$h_{fins\ unknown\ dimentions} = \frac{0.28\ R_e^{0.6} P_r^{0.33}\ k}{d_s} \qquad (24)$$

Results

	Shell (Air) T, C	Tube (Water) t, C
Inlet	19	10.61
Outlet	13.49	13

Table 1. Temperatures used in calculations

LMTD	4.251	Volumetric Flow Rate	1.5	m^3/s
P	0.285	Water Density	1000	kg/m^3
R	2.305	Dynamic Viscosity of Water	0.001	Pas
Initial area	13.072	C	0.023	
Optimized area	4.702	Thickness of tube	0.005	m

Properties	Initial area	Optimized area
	set 5	
Tube side Diameter	0.018	
Tube Length	2.5	
Pitch	0.023	
Area of Tube	0.071	
No of Tubes without spacing	185	67
Tube Spacing	0.005	
Area of Tube Spacing	0.106	
Final number of Tubes	124	45
Final number of Banks	41.095	14.781
Bundle Area	0.318	
Velocity	0.001	0.004
Reynolds number	3225.076	8966.389
frictional factor	0.012371910	0.009274369
pressure drop	0.005719186	0.033138815
Fan power consumption	0.006497123	0.037646435
Total power consumption	0.006984408	0.040469917
Heat Transfer Coefficient	200	404.4133

Table 2: best results achieved for tube in tube and shell and tube heat exchanger

The findings from the heat exchanger design study disclose the precise dimensions and performance characteristics of both exchangers, showing the benefits of some design decisions over others. Initially as seen on Table 1, the initial inlet temperature is taken as 19 degrees instead of 20 as stated in the project guide. This is to take into account the heat loss due to factors like thermal conduction, convection, and radiation. This means that by the time the fluid reaches the actual inlet point, its temperature may have decreased. Accounting for heat loss acknowledges the real-world factors that can affect system performance, such as ambient temperature variations and material properties of the pipes or ducts.

The optimised heat exchanger design lowered the initial heat transfer area from 13.072 m^2 to 4.702 m^2, a 64% reduction. The considerable drop is mainly due to the optimised design's increased heat transfer coefficient of 404.41W/m^2·K. A greater heat transfer coefficient indicates the heat exchanger transfers more heat per unit area per degree of temperature difference. Increased flow velocity and turbulent flow disturb the thermal boundary layer next to the tube walls, improving convective heat transfer and increasing the coefficient. More effective heat transfer per unit area reduces the surface area needed for heat exchange, reducing material and making the design more compact.

In the optimised design, flow velocity increased from 0.001 to 0.004m/s and Reynolds number from 3,225 to 8,966. Having fewer tubes reduces the cross-sectional area available for flow, which increases flow velocity at the same volumetric flow rate. Above 4,000 Reynolds [11] numbers imply turbulent pipe flow from a transient flow, indicating a change from laminar to turbulent flow. Turbulent flow improves convective heat transmission by mixing fluid and disturbing the thermal boundary layer. Although flow velocity increased and turbulent flow began, the friction factor reduced from 0.0124 to 0.0093. In turbulent flow regimes, tube surface roughness affects friction factor more than Reynolds number. Reduced friction means the optimised design may use tubes with smoother interior surfaces or materials. Techniques might have reduced flow resistance. Even when flow velocities rise, the decreased friction factor reduces energy losses and improves system efficiency.

In the optimised design, tubes dropped from 124 to 45 and banks from 41 to 15. The increased heat transfer coefficient of each tube improves heat transfer efficiency, causing this drop. For the same heat transfer rate, fewer tubes are required due to each tube's improved performance. The optimised design may also include a staggered tube structure to increase turbulence and heat transfer efficiency. The reduction in tubes and banks simplifies maintenance and lowers manufacturing and assembly costs, improving operations and finances.

The optimised heat exchanger pressure drop was 0.0331Pa, up from 0.0057Pa. This is predicted as flow velocity and turbulence increases. Pressure drop increases proportionally to flow velocity squared, therefore even a slight velocity increase may significantly increase pressure drop. Due to fewer tubes, fluid must flow via a smaller area, increasing resistance and pressure loss. Higher pressure drops require more energy to overcome, but they may improve heat transfer performance if they stay within the system's design limitations.

In the optimised design, the heat transfer coefficient rose to 404.41 $W/m^2 \cdot K$ compared to the initial assumed 200. The increase in flow velocity and turbulent flow are responsible for this enhancement. Turbulent flow breaks the thermal boundary layer at the tube surface, improving fluid-tube wall mixing and heat transmission. The smaller thermal boundary layer from higher velocities improves heat transmission. The optimised design may also include improved tube surface geometries or thermal conductivity materials to increase heat transfer coefficient. This greater coefficient lets the heat exchanger work thermally with less surface area.

Fins	Heat transfer coefficient
Known geometry	1286.391
Unknown geometry	1694.153

Table 3: Heat transfer coefficient for optimized design with the addition of fins

Fins are surfaces that extend from the tube in order to improve convection, which speeds up heat transfer. the greater heat transfer between the fluids is made possible by the larger area [12]. Allowing the performance to be achieved through improved heat transmission without requiring larger equipment. As seen in table 3, the heat transfer coefficient was solved for a known geometry, a pitch length of 3 mm, fin length of 15 mm, fin width of 1 mm, and a unknown geometry [6]. There are no extended surfaces on the heat exchanger's simple surfaces. Natural convection and conduction are only two variables that affect the heat transfer coefficient. The rate of heat transfer is comparatively small since the surface area is constrained to the base area. Adding fins with a known geometry increases the heat transfer coefficient by approximately 218% compared to having no fins. Adding fins with an optimized unknown geometry increases the heat transfer coefficient by approximately 319% compared to having no fins. This shows that a increase in the surface area has a huge effect on the heat transfer in a heat exchanger. We see a 27.36% difference between known and unknown geometry. The unknown geometry having a higher heat transfer coefficient means the known geometry is not optimized and has room to improve.

Conclusion
In conclusion, a more economical and efficient system is produced by optimising the heat exchanger design. The observed changes, including a reduction in heat transfer area, an increase in the heat transfer coefficient, and the decrease in the number of tubes and banks, are all related results of the design adjustments based on fluid dynamics and heat transfer theories. A smaller, more effective heat exchanger is made possible by the purposeful increase in flow velocity and the shift to turbulent flow, which improve convective heat transfer. Higher power consumption and pressure drops result from this, but these are reasonable trade-offs considering the significant improvements in thermal performance. The optimised design is a well-balanced strategy that maximises material use and operating efficiency while satisfying the required thermal criteria. Adding fins to a heat exchanger increases the heat transfer coefficient by 218% with known geometry and up to 319% with optimized geometry, demonstrating that increasing surface area significantly enhances heat transfer and that fin design can be further optimized for improved performance.

A number of improvements might be made to the heat exchanger design to make it more effective. By experimenting with various designs, it may be possible to optimise the fin geometry and increase heat transmission without significantly raising pressure drop. Efficiency may be increased by using materials with better heat conductivity, such copper or aluminium alloys. By offering more in-depth understanding of flow and thermal properties, computational fluid dynamics (CFD) modelling may assist in optimising design parameters. Lastly, constructing and testing a prototype would confirm the theoretical results and guarantee that the heat exchanger operates as planned under actual circumstances.

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Appendix

Properties	units	Initial area					Optimised area
		set 1	set 2	set 3	set 4	set 5	set 5
Tubeside Diameter	m	0.030	0.025	0.025	0.020	0.018	0.018
Tube Length	m	5	5.5	4.5	3.0	2.5	2.5
Pitch	m	0.038	0.031	0.031	0.025	0.023	0.023
Area of Tube	m^2	0.236	0.216	0.177	0.094	0.071	0.071
Number of Tubes req		56	61	74	139	185	67
Tube Spacing		0.008	0.006	0.006	0.005	0.005	0.005
Area of Tube Spacing	m	0.353	0.324	0.265	0.141	0.106	0.106
Number of Tubes	m^2	36.986	40.348	49.315	92.465	123.286	44.344
Number of Banks		12.329	13.449	16.438	30.822	41.095	14.781
Bundle Area		1.060	0.972	0.795	0.424	0.318	0.318
Velocity	m/s	0.014	0.012	0.008	0.002	0.001	0.004
Reynolds number		71668.364	66243.467	36282.109	6880.163	3225.076	8966.389
frictional factor		0.0059119	0.0059930	0.0067102	0.0099537	0.0123719	0.0092743
pressure drop	Pa	0.404877538	0.382523685	0.157035722	0.015705911	0.005719186	0.033138815
Fan power consumtion	W	1.533166433	1.327808330	0.445990222	0.023789688	0.006497123	0.037646435
Total power consumtion	W	1.648153915	1.427393955	0.479439488	0.025573914	0.006984408	0.040469917

Table 3: Full results

Sample calculation

Sample calculation 1

tube in tube

Inlet Shell temperature : 19°C

Outlet Shell temperature : 13.49°C

Inlet Tube temperature : 10.61°C

Outlet Tube temperature : 13°C

air

$\dot{m} = \rho \dot{V} = 1.2 \times 15 = 1.8$

C_p air = 0.718 kJ/kg k

C_p air = 1.005 kJ/kg k

$Q = \dot{m}_a C_{p_a} (T_{a_{in}} - T_{a_{out}}) = 10 \times 10^3 = 1.8 \times C_p \times 10^3 (T_{a_{in}} - T_{a_{out}})$

$\Delta T_A = \frac{10}{1.8 \times 0.718} = 7.7375\ °C$

$\Delta T_A = \frac{10}{1.8 \times 1.005} = 5.5279\ °C$

water

$\dot{m} = 1$ kg/s

$C_p = 4182$ J/kg k

$\Delta T = Q/\dot{m} C_p = 10^4 / 1 \cdot 4182 = 2.39°$ C

air

$Q = 10$ kw

$\mu = 1.806$ kg s^{-1}

$\rho = 1.204$ kg m^{-3}

$C_p = 1.005$ kJ/kg k

$P_r = \frac{C_p \mu}{k} = \frac{(1.005 \times 10^3)(0.001)}{0.6} = 1.675$

$U = 200$

$T = 0$

$a_t = C R_e^{0.8} P_r^{0.33} \left(\frac{\mu}{\mu_w}\right)^{0.14} = 0.023 \cdot (3725.076)^{0.8} \cdot (1.675)^{0.33} \cdot (\ 1\ )^{0.14}$

$= 17.47282$

$a_t = \frac{h_i\ de}{k_f} = 17.47282 = \frac{h_i\ (0.018)}{0.6}$ $\quad h_i = 582.42728$

$j_h = \frac{h_i\ d_e}{k_f\ R_e\ P_r^{0.33}\ (\ 1\ )^{0.14}} = \frac{582.42728 \cdot 0.018}{0.6 \cdot (3725.76) \cdot ((1.675)^{0.33} \cdot (\ 1\ )^{0.14}} = 4.582 \times 10^{-3}$

Shell

$A_s = \frac{(P_t - d_e)\ D_s\ l_b}{P_t}$ $\quad D_s = d_{e_{shell}}$ - material thickness

$A_{req} = 2\pi r h + 2\pi r^2 = 13.072$ $\quad$ Calculated $\frac{\mu}{\mu_w} = 1$

$r = 0.3862559$ $\quad r = -5.3862..$ x $\quad r = 0.3862559$ $\quad d_{shell} = 0.7725118$

Shell material : Stainless Steel AISI 304 $\quad$ thickness : 5mm

$D_s = 0.7725 - 0.005 = 0.7675$

$A_s = \frac{(0.0225 - 0.018)(0.7675)}{0.0225}(25 \times 0.4) = 0.13815\ m^2$

$G_s = \frac{\dot{m}_a}{A_s} = \frac{1.806}{0.13815} = 13.0727$

$u_s = \frac{G_s}{\rho} = \frac{13.0727}{1.2} = 10.8939$

$d_e = \frac{1.1}{d_o}\left(P_t^2 - 0.917 d_o^2\right) = \frac{1.1}{0.018}\left(0.0225^2 - 0.917(0.018)^2\right) = 0.012781$

$R_e = \frac{G_s\ d_e}{\mu} = \frac{13.0727 \cdot 0.012781}{1.8 \times 10^{-5}} = 9282.3432$

$N_u = \frac{a_s\ d_e}{k_f} = j_h\ R_e P_r^{0.33} (\ 1\ )^{0.14} = \frac{a_s \cdot 0.012781}{0.6} =$

$= 4.582 \times 10^{-3} \cdot 9282.3432 \cdot 1.675^{0.33} \cdot 1^{0.14}$ $\quad a_s = 2366.4769$

$\Delta P_s = 8 j_h \left(\frac{D_s}{de}\right)\left(\frac{L}{l_B}\right) \frac{\rho u_s^2}{2} (1)^{0.14}$

$= 8 (4.582 \times 10^{-3}) \left(\frac{0.7675}{0.012781}\right)\left(\frac{25}{1}\right) \frac{1.2 \cdot 10.8939^2}{2} (1)^{0.14} = 391.7372$

$\frac{1}{U} = \frac{1}{h_o} + \frac{d \ln\left(\frac{d}{d_o}\right)}{2k_{wall}} + \frac{d_o}{d_i a_i} = \frac{1}{2366.4769} + \frac{0.7725 \ln\left(\frac{0.7725}{0.7625}\right)}{2(16.2)} + \frac{0.7725}{0.7625 \cdot 582.42}$

$\frac{1}{U} = 2.47271785 8 \times 10^{-3}$

$U = 404.4133$

new area $= \frac{Q}{UF \Delta T_{lm}} = \frac{10^4}{404.4133 \cdot 1.237191 \cdot 4.2509} = 4.701766$

Fins calculation

known dimensions

$N_{u_{fins}} = 0.134\ R_{e_s}^{0.681} P_r^{0.33} \left(\frac{P_f - w_f}{l_f}\right)^{0.2} \left(\frac{P_f}{w_f}\right)^{0.1134} = \frac{h_f d_o}{k_{air}}$

P_f usually 2-4mm $\quad$ 3mm $\quad$ 0.003 m

l_f usually 12-16 mm $\quad$ 15mm $\quad$ 0.015m

w_f

fin width $\quad$ 0.001 $\quad$ 1 mm $\quad$ 0.001 m

$h_f = \frac{0.134\ R_{e_s}^{0.681} P_r^{0.33} \left(\frac{P_f - w_f}{l_f}\right)^{0.2} \left(\frac{P_f}{w_f}\right)^{0.1134} k}{d_o}$

$h_f = \frac{(0.134)(9282.3432)^{0.681}(1.675)^{0.33}\left(\frac{2}{15}\right)^{0.2}(3)^{0.1134}\ 16.2}{0.7625} = 1386.390993$

Unknown dimensions

$h_f = \frac{0.28\ R_e^{0.6}\ P_r^{0.33}\ k}{d_o}$

$= \frac{0.28\ (9282.3432)^{0.6}\ (1.675)^{0.33}\ 16.2}{0.7625} = 1694.153115$

∴ 27.3616 percentage difference