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A Brief Report on Heat Transfer Equipment Design

- **Compact Air-Cooled Condenser**
- **Plate Fin Heat Exchanger**

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MECHANICAL ENGINEERING, BUET

INDEX

The different segments of the report contain- *Page No.*

The completion of this project was very much complex and difficult than we conceived the idea of this project. We faced many problems as well as constraints in implementing it. We had to deal with some hardware components that needed help of professional experts from Machine shop and an engineering workshop from Wari. We are grateful to them.

We are also very thankful to our course teachers-**Md.Aminul Islam**, and **Md.Rakib Hossain Sir**. They provided us instructions, guidance and regularly kept track of our project .

It was very helpful to finish our project in time.

ACKNOWLEDGEMENT

Our classmates were very co-operative. They encouraged us to overcome various hurdles. Finally, our collective effort, mutual understanding, teamwork as well ensured our success. By the grace of Almighty, we were able to finish our project successfully in due time. We realized proper teamwork, sharing of knowledge and experience can help us achieve greater goals.

The main purpose of our project was to build compact air-cooled condenser which contains total 40 copper tubes and many tube in plate continuous plain fins made in Aluminum.

The heat exchanger follows crossflow mechanism and half-serpentine circuiting tube passes. It's a compact mini heat exchanger. We tried out utmost so that we can keep our condenser core size as minimum as possible. For that we followed many iterations process to minimize the core size for the most economical solution with **ASPEN.**

We have looked for the most accurate economic possible solution considering about the final targeted result according to the question and the economic balance. We have passed the hydraulic test and leakage test (pressure test) successfully in the lab and finally completed our project with proper teamwork.

Plate fin heat exchangers, because of their compactness, low weight and high effectiveness are widely used in aerospace and cryogenic applications. This device is made of a stack of corrugated fins alternating with nearly equal number of flat separators known as parting sheets, bonded together to form a monolithic block. Appropriate headers are welded to provide the necessary interface with the inlet and the exit streams. While aluminum is the most used material, stainless steel construction is employed in high pressure and high temperature applications.

The performance of a plate fin heat exchanger is determined, among other things, by the geometry of the fins. The most common fin configurations are - (1) plain (straight and uninterrupted) rectangular or trapezoidal fins (2) uninterrupted wavy fins and (3) interrupted fins such as offset strip, louver and perforated fins. The interrupted surfaces provide greater heat transfer at the cost of higher flow impedance.

The heat transfer surface area is increased by fins to increase the surface area per unit volume and there are many variations. The device is referred to a micro heat exchanger if the surface area density is above 10000 square meter/cubic meter.

Here I have designed continuous tube in plate fin(plain) heat exchanger. I have assumed some data and based on them I have designed heat exchanger.

The flowing fluid in heat exchanger is R134a and material of heat exchanger is Al fins and tubes are copper material. After designing the heat exchanger, rating is also necessary. R134a enters the condenser as a gaseous form and leaves the condenser as liquid at constant 37 degree Celsius thus condensing inside the condenser.

Flow Arrangement:

A plate fin heat exchanger is a form of compact heat exchanger consisting of a block of alternating layers of corrugated fins and flat separators known as parting sheets. A schematic view of such an exchanger is given in the Fig below. The corrugations serve both as secondary heat transfer surface and as mechanical support against the internal pressure between layers.

Steam exchanges heat by flowing along the passage corrugations between the parting sheets. The edges of the corrugated layers are sealed by sidebars. Corrugations and sidebars are brazed to the parting sheets on both sides to form rigid pressure-containing voids. The first and the last sheets, called cap sheets, are usually of thicker material than the parting sheets to support the excess pressure over the ambient and to give protection against physical damage.

Each stream enters the block from its own header via ports in the sidebars of appropriate layers and leaves in a similar fashion. The header tanks are welded to the sidebars and parting sheets across the full stack of layers.

A plate fin heat exchanger accepts two or more streams, which may flow in directions parallel or perpendicular to one another. When the flow directions are parallel, the streams may flow in the same or in opposite sense.

Thus, we can think of three primary flow arrangements – (I) parallel flow, (ii) counterflow and (iii) cross flow. Thermodynamically, the counterflow arrangement provides the highest heat (or cold) recovery, while the parallel flow geometry gives the lowest. The crossflow arrangement, while giving intermediate thermodynamic performance, offers superior heat transfer properties and easier mechanical layout.

Under certain circumstances, a hybrid cross – counterflow geometry provides greater heat (or cold) recovery with superior heat transfer performance. Thus, in general engineering practice, plate fin heat exchangers are used in three configurations: (a) cross flow, (b) counterflow and (c) cross-counter flow.

(a) **Cross flow:** In a crossflow heat exchanger, usually only two streams are handled, thus eliminating the need for distributors. The header tanks are located on all four sides of the heat exchanger core, making this arrangement simple and cheap. If high effectiveness is not necessary, if the two streams have widely differing volume flow rates, or if either one or both streams are nearly isothermal (as in single component condensing or boiling), the crossflow arrangement is preferred. Typical applications include automobile radiators and some aircraft heat exchangers.

(b) **Counter flow:** The counterflow heat exchanger provides the most thermally effective arrangement for recovery of heat or cold from process streams. Cryogenic refrigeration and liquefaction equipment use this geometry almost exclusively. The geometry of the headers and the distributor channels is complex and demands proper design.

(c) **Cross-Counter flow:** The cross-counterflow geometry is a hybrid of counterflow and cross flow arrangements, delivering the thermal effectiveness of counterflow heat exchanger with the superior heat transfer characteristics of the cross-flow configuration.

In this arrangement, one of the streams flows in a straight path, while the second stream follows a zigzag path normal to that of the first stream. Up to six such passes have been employed. While negotiating the zigzag path, the fluid stream covers the length of the heat exchanger in a direction opposite to that of the direct stream. Thus, the flow pattern can be seen to be globally counterflow while remaining locally cross flow.

Cross-counter flow PFHEs are used in applications similar to those of simple cross flow exchangers but allow more flexibility in design. They are particularly suited to applications where the two streams have considerably different volume flow rates or permit significantly different pressure drops. The fluid with the larger volume flow rate or that with 14 the smaller value of allowable pressure drops flows through the straight channel, while the other stream takes the zigzag path.

For example, in a liquid-to-gas heat exchanger, the gas stream with a large volume flow rate and low allowable pressure drop is assigned the straight path, while the liquid stream with a high allowable pressure drop flows normal to it over a zigzag path. This arrangement optimizes the overall geometry. Some plate fin heat exchangers are given below.

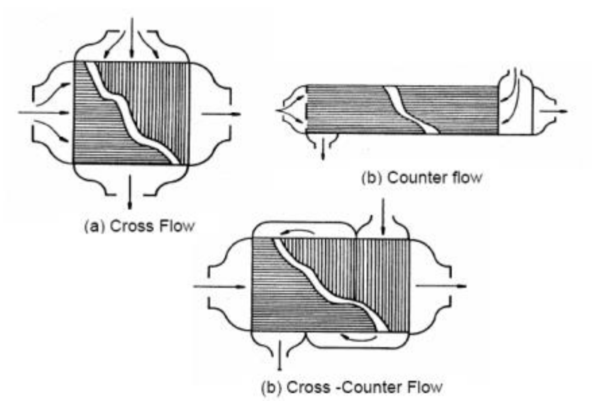
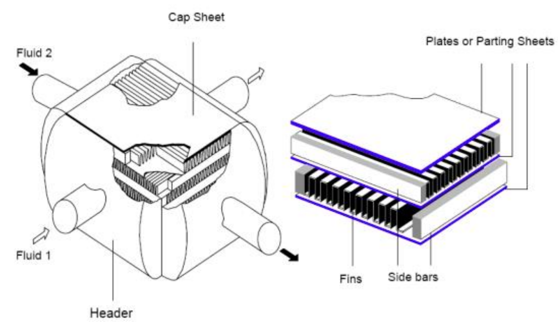
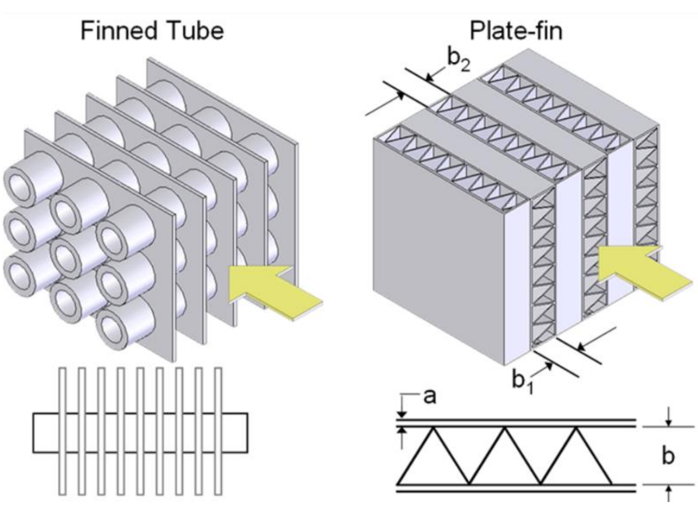
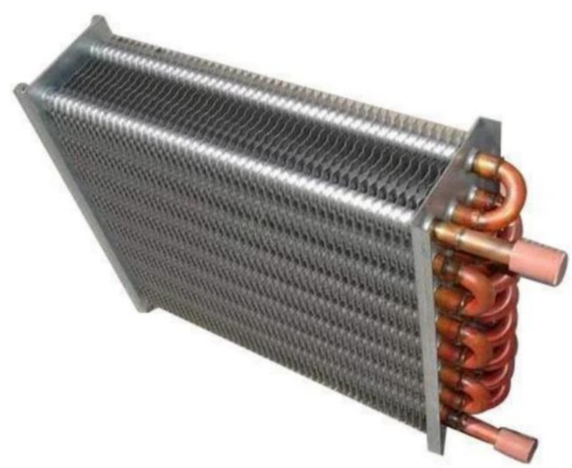


Figure 1.2: Heat exchanger flow arrangements







Design of a Compact Air-Cooled Refrigerant Condenser:

Specifications:

Cooling load (heat duty)	Q = 18kW
Refrigerant:	R-134A condensing inside tubes at Ts= 37°C (310 K)
Coolant:	Air Inlet temperature, Tc1 = 25°C Outlet temperature, Tc2 = 30°C Mean pressure P = 1 atm
Heat transfer matrix:	13-3/8T (Air Side Condition)

At least two different surfaces must be studied for thermal and hydraulic analysis and then compared; the primary attention must be given to obtaining the smallest possible heat exchanger. A parametrical study is expected to develop a suitable final design. The final report will include material selection, economic analysis (both hand and software analyses), thermo- hydraulic calculations (both hand and software), mechanical design, and technical drawings of the components and the assembly.

Plate-fin and tube-fin heat exchangers have found application in a wide variety of industries. Among them are air separation (production of oxygen, nitrogen and argon by low temperature distillation of air), petrochemical and syn-gas production, helium and hydrogen liquefiers, oil and gas processing, automobile radiators and air conditioners, and environment control and secondary power systems of aircrafts. These applications cover a wide variety of heat exchange scenarios, such as:

(1) exchange of heat between gases, liquids or both,

(2) condensation, including partial and reflux condensation,

(3) boiling,

(4) sublimation, and

(5) heat or cold storage

(6) Natural Gas liquefication

(7) Cryogenic air separation

(8) Ammonia production

(9) Offshore processing

(10) Nuclear engineering

(11) Syngas production

(12) Aircraft cooling of bleed air and cabin air

Our main objective is to make a compact air-cooled condenser that meets up the preceding characteristics mentioned in the question above. A R-134a refrigerant enters the condenser at a given temperature (37 degree Celsius) following the required latent heat pressure (9.33 bar) for condenser for that the compressor we used had the constant outlet pressure 9.33 bar. The refrigerant goes inside the tubes to the outlet and leaves the condenser as a liquid form.

The latent heat needed for condensing comes from the air which enters the condenser at our desired 25 degree Celsius and leaves at 30 degree Celsius in 1 atm pressure(theoretical and experimental) at a cross flow mechanism. An Induced fan does the job the perfectly.

After leaving the condenser, the refrigerant goes through the expansion valve to reduce pressure and after that to the evaporator at a low-pressure higher volume.

Evaporator again turns the liquid to gaseous form at a low pressure. Then gas goes to the compressor again, and the loop continues. That was the main objective of our project to fulfill our desired problem condition with economic consideration.

Many components we have used in our heat transfer equipment design project. The used components are-

1. Copper tubes (40 tubes,14-inch length)
2. Aluminum plain fins (13 fins/inch)
3. Sheet metal (Stainless steel)
4. Casing (Steel)
5. A Gasketed bar
6. Induced fan.
7. Motor
8. Wires
9. Nuts and Bolts

The basic principles of plate fin heat exchanger (Condenser) manufacture are the same for all sizes and all materials. The corrugations, sidebars, parting sheets and cap sheets are held together in a jig under a predefined load, placed in a furnace and brazed to form the plate fin heat exchanger block. The header tanks and nozzles are then welded to the block, taking care that the brazed joints remain intact during the welding process. Differences arise in the manner in which the brazing process is carried out.

The methods in common use are salt bath brazing and vacuum brazing. In the salt bath process, the stacked assembly is preheated in a furnace to about 550 0 C, and then dipped into a bath of fused salt composed mainly of fluorides or chlorides of alkali metals. The molten salt works as both flux and heating agent, maintaining the furnace at a uniform temperature. In case of heat exchangers made of aluminum, the molten salt removes grease and the tenacious layer of aluminum oxide, which would otherwise weaken the joints.

Brazing takes place in the bath when the temperature is raised above the melting point of the brazing alloy. The brazed block is cleansed of the residual solidified salt by dissolving in water, and then thoroughly dried. In the vacuum brazing process, no flux or separate pre-heating furnace is required.

The assembled block is heated to brazing temperature by radiation from electric heaters and by conduction from the exposed surfaces into the interior of the block. The absence of oxygen in the brazing environment is ensured by application of high vacuum (Pressure ≈ 10 -6 mbar). The composition of the residual gas is further improved (lower oxygen content) by alternate evacuation and filling with an inert gas as many times as experience dictates.

No washing or drying of the brazed block is required. Many metals, such as aluminum, stainless steel, copper and nickel alloys can be brazed satisfactorily in a vacuum furnace.

The main thermal hydraulic objective of our compact heat exchanger is to produce more efficient heat exchanger equipment for minimizing cost, which is to reduce the physical size of a heat exchanger for a given duty. The main goal is to obtain high heat transfer rates under the specified conditions for our problem. That's why we have increased the surface area by using plain fins and reducing the core size .Thus We ensured our desired heat transfer rate.

$$Q = hA(T_s\text{-}T_o)$$

We have used the heat transfer coefficient equations for air side and tube side conditions. There are various techniques used for heat transfer enhancement, which are segregated in two groups.(1) the active techniques that require external power to the surface(surface vibration, acoustic or electric field), and (2) the passive techniques, which is the case generally applied in designing compact hear exchangers, the heat transfer co-efficient may or may not be increased. The heat transfer coefficient of an enhanced surface, which is given by the **Colburn** modulus,**J**,and the frictional factor of the surface,**f**,are usually presented as a function of the Reynolds number(D_hG/u)

$J=(h/Gc_p)*Pr^{2/3}$

For air side condition, to measure the heat transfer coefficient and to determine friction factor we have used the graphs mentioned below.

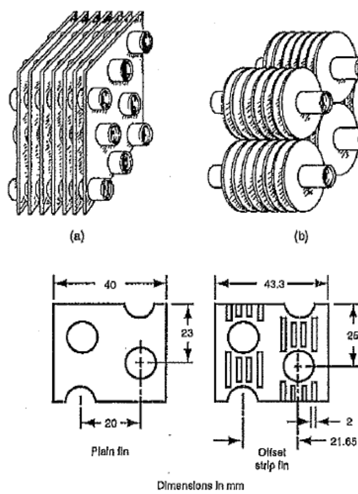
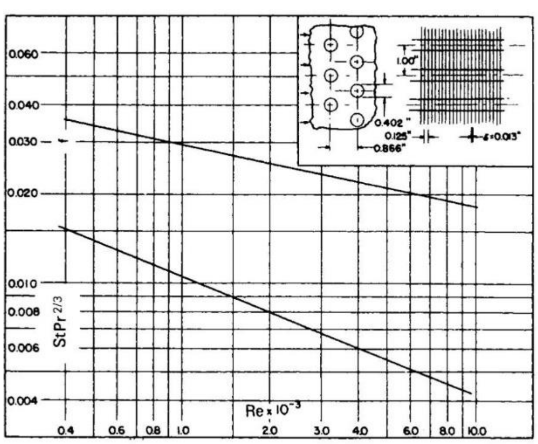


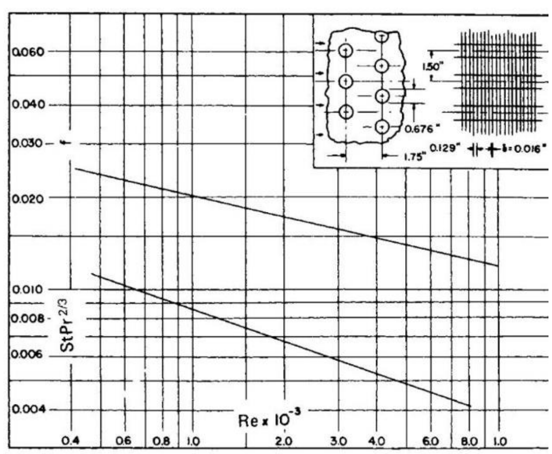
FIGURE 9.1
Finned-tube geometries used with circular tubes: (a) plate fin-and-tube used for gases; (b) individually finned tubes; (c) plain-fin and offset-fin. (From Webb, R. L., *Principles of Enhanced Heat Transfer*, John Wiley & Sons, New York, 1994.)

Fig. 10-91 Finned circular tubes, surface 8.0–3/8T.



Tube outside diameter = 0.402 in = 10.2×10^{-3} m
Fin pitch = 8.0 per in = 315 per m
Flow passage hydraulic diameter, $4r_h$ = 0.01192 ft = 3.632×10^{-3} m
Fin thickness = 0.013 in = 0.33×10^{-3} m
Free-flow area/frontal area, σ = 0.534
Heat transfer area/total volume, α = 179 ft^2/ft^3 = 587 m^2/m^3
Fin area/total area = 0.913
Note: Minimum free-flow area in spaces transverse to flow.

Fig. 10-92 Finned circular tubes, surface 7.75–5/8T.



Tube outside diameter = 0.676 in = 17.17×10^{-3} m
Fin pitch = 7.75 per in = 305 per m
Flow passage hydraulic diameter, $4r_h$ = 0.0114 ft = 3.48×10^{-3} m
Fin-flow area/frontal area, σ = 0.481
Heat transfer area/total volume, α = 169 ft^2/ft^3 = 554 m^2/m^3
Fin area/total area = 0.950
Note: Minimum free-flow area in spaces transverse to flow.

The graphs mentioned above are given from **HEAT EXCHANGERS SELECTION, RATING and THERMAL DESIGN by Sadik Kakac ,Hongtan Liu .**

For measuring tube passes, we have followed the condenser coil circuiting theorem.

circuiting means the number of tubes on any coil being fed by each header. Every coil has a specific number of rows, and a specific number of tubes within each row.

For example, in the illustration below we show a coil that is 8 tubes high and 4 rows deep for a total of 32 tubes (in another case, you could have a coil that is 24 tubes high and 8 rows deep for a total of 192 tubes). In this initial example, the “supply” enters from the right and the “return” exits on the left. (Note: Most fluid coils have the supply and return connections on the same end of the coil) .

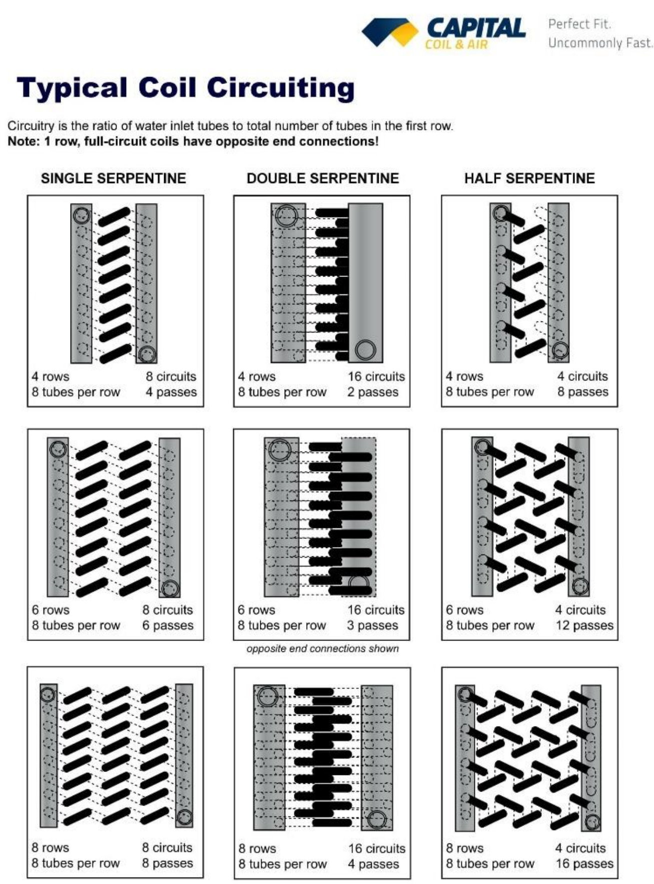
Determining the right number of circuits

To ensure proper heat transfer, fluid must travel through coils at the right speed. Some coils only have a small number of circuits while others have quite a few. If fluid travels too quickly through the coil, the heat transfer will be inefficient, the coil could have a high pressure drop, and this situation could cause tube erosion. If it travels too slowly, only a little heat transfer will occur. By selecting the appropriate number of circuits, you control the fluid speed and thus the heat transfer efficiency of the coil. Fewer circuits serve to speed up the fluid in the coil, while more circuits slow it down.

For example, feeding a larger number of tubes simultaneously spreads the flow of fluid to multiple circuits, which slows down the speed at which it flows through the coil. The number of circuits and the inside diameter of the tubes allows you to calculate total flow rate (in GPM) of the coil. If a coil has 8 tubes per row and is 4 rows deep, as shown in our example, then the circuiting is as follows:

- Half Circuit – 4 feeds
- Quarter Circuit – 2 feeds
- Full Circuit – 8 feeds
- Double Circuit – 16 feeds

Finally, here are 3 rules of thumb to follow regarding coil circuiting: 1. The number of tubes you feed must divide evenly into the number of tubes in the coil or you will have dropped tubes (tubes that aren't fed fluid). 2.The coil must have an even number of passes if you want connections to end up on the same end of the coil. With an odd number of passes, you will have fluid connections on opposite ends. 3.The number of circuits will determine the resulting pressure drop by determining the number of circuits and thus the fluid velocity and effective tube length for each circuit (acceptable fluid tube velocity is between 2 and 7 feet per second).



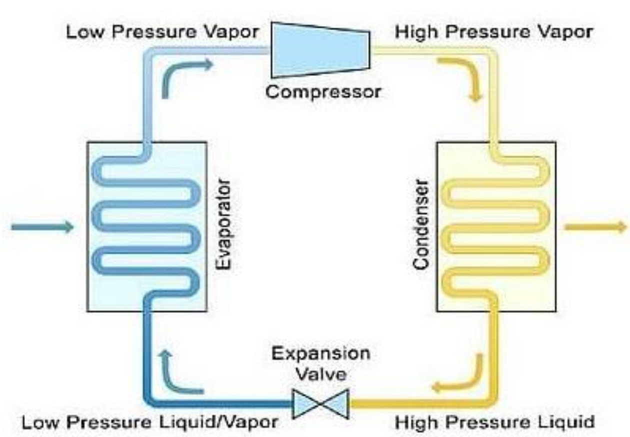
We have used half circuit criteria (half-serpentine) in our condenser. Design. That means 4 feeds were used. Total no of tubes =40.So our required tube passes are 40/4=10.

That's the no of passes we need in our condenser design for better heat transfer rate and to minimize the economic cost at all.

A refrigeration cycle's mission is heating absorption and heat rejection. As any HVAC instructor will tell you (emphatically), you can't make cold, you can just remove heat. The refrigeration cycle, sometimes called a heat pump cycle, is a means of routing heat away from the area you want to cool. This is accomplished by manipulating the pressure of the working refrigerant (air, water, synthetic refrigerants, etc.) through a cycle of compression and expansion.

That's not the full picture, of course, but that's the basic idea. Now, let's get into the equipment that helps execute that job. There are certainly other components in most loops, but most would agree the four fundamental elements of a basic cycle are as follows:

- The compressor
- The condenser
- The expansion device
- The evaporator



The compressor

Compression is the first step in the refrigeration cycle, and a compressor is the piece of equipment that increases the pressure of the working gas. Refrigerant enters the compressor as low-pressure, low-temperature gas, and leaves the compressor as a high-pressure, high-temperature gas.

Types of compressors

Compression can be achieved through several different mechanical processes, and because of that, several compressor designs are used in HVAC and refrigeration today. Other examples exist, but some popular choices are:

1. Reciprocating compressors

2. Scroll compressors

3. Rotary compressors

The condenser

The condenser, or condenser coil, is one of two types of heat exchangers used in a basic refrigeration loop. This component is supplied with high-temperature high-pressure, vaporized refrigerant coming off the compressor. The condenser removes heat from the hot refrigerant vapor gas vapor until it condenses into a saturated liquid state, a.k.a. condensation. After condensing, the refrigerant is a high-pressure, low-temperature liquid, at which point it’s routed to the loop’s expansion device.

The expansion device

These components come in a few different designs. Popular configurations include fixed orifices, thermostatic expansion valves (TXV) or thermal expansion valves (pictured above), and the more advanced electronic expansion valves (EEVs). But regardless of configuration, the job of a system’s expansion device is the same - create a drop in pressure after the refrigerant leaves the condenser. This pressure drop will cause some of that refrigerant to quickly boil, creating a two-phase mixture.

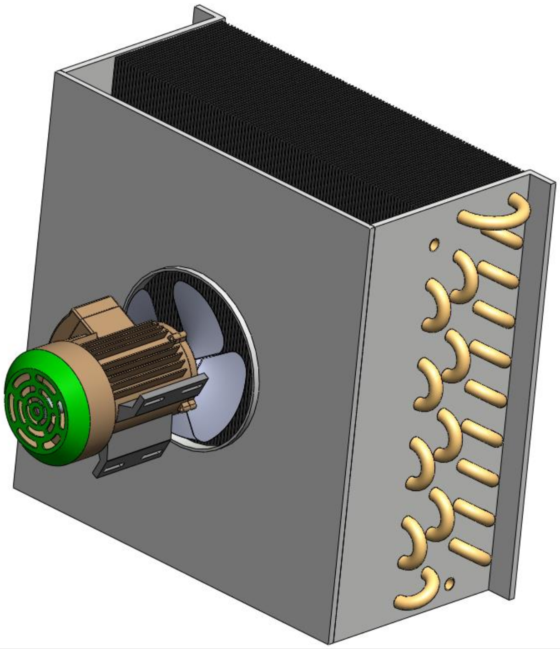
This rapid phase change is called *flashing,* and it helps tee up the next piece of equipment in the circuit, *the evaporator*, to perform its intended function.

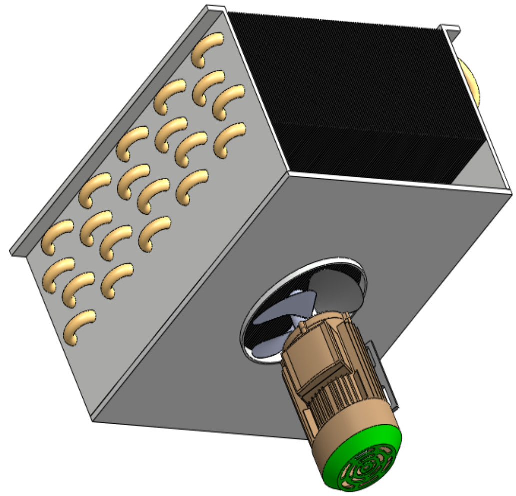
The evaporator

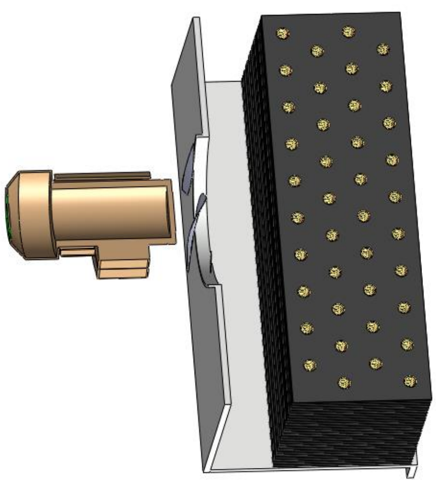
The evaporator is the second heat exchanger in a standard refrigeration circuit, and like the condenser, it’s named for its basic function. It serves as the “business end” of a refrigeration cycle, given that it does what we expect air conditioning to do – absorb heat.

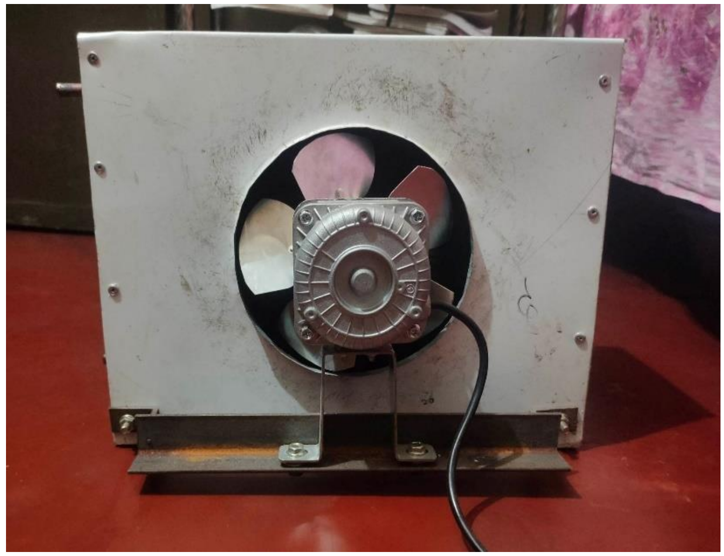
This happens when refrigerant enters the evaporator as a low temperature liquid at low pressure, and a fan forces air across the evaporator’s fins, cooling the air by absorbing the heat from the space in question into the refrigerant.

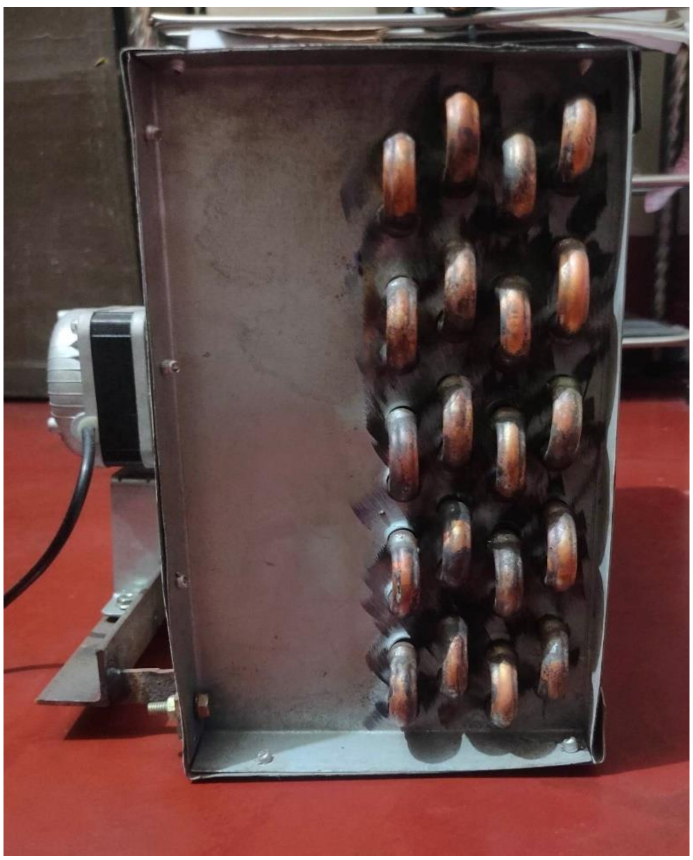
Solidworks Model











Code:

```
main_p =1; %main pressure

Tin_a = 298.15; %air in temp

Tout_a=303.15; % air out temp

Tr = 310.15;%refrigerant temp

Q = 18e3;%cooling load

%gc=

od = 9.525e-3;%outer dia

fin_pi = 13; %fin per inch

ntp = 10; %number of tube pass

fin_t = 0.7e-3; %fin thickness

nt = 40; %number of tubes

f_area = 0.06; %min flow area or frontal area

dh = 0.002; %hydraulic dia
```

```
t_area= 6.3; %heat transfer area/total area

p_t = 25e-3; %transverse pitch

p_l = 21.65e-3; %longitudinal pitch

%refrigeration side

t_thick = 0.00036; %tube thickness

single_f_area = 25*21.65*1e-6; %single tube frontal area

di = 8.81e-3; %internal dia

sig_w = ((pi/4)*di^2)/single_f_area; %heat transfer area/total area

hta_v = (pi*di)/single_f_area; %heat transfer area/ volume

t_film = (Tin_a+ Tout_a)/2; %film temperature

%At pm=1 atm
%Properties of air

rho_a = 1.18; %air density

mu_a = 1.84e-5; %air viscosity

cp_a = 1010;

pr_a = 0.7; %prandtl number
```

```
k_a = 0.0262;

%properties of refrigerant(r134a) at 310k

p_satr = 933000; %refrigerant saturation pressure

rho_rl = 1163; %liquid refrigerant density

mu_rl = 1.89e-4;

cp_rl = 1.497e3;

pr_rl = 3.91;

k_r = 0.071;

h_fg = 165.3e3; %latent heat of condensation

rho_rg = 44.45;

mu_rg = 0.125e-4;

k_alum = 232; %aluminium thermal conductivity

mf_r = round(Q/h_fg,2); %mass flow rate of refrigerant

mf_a = Q/(cp_a*(Tout_a - Tin_a)); %mass flow rate of air

Re_a = 7e3; %air reynolds number
```

```
G_a = (Re_a*mu_a)/(dh); %mass flux of air

% air side heat transfer coefficient

fr_a = 0.02; %friction factor of air

h_a = (0.0175*G_a*cp_a)/((pr_a)^(2/3)); %heat transfer coefficient
air

%fin effciency

l_fin = 0.00635; %fin length

ml_fin = sqrt(2*h_a/(k_alum*t_thick))*l_fin;

eta_fin = tanh(ml_fin)/ml_fin; %fin effciency

Af_At = 0.6;

toteta_fin = round((1-Af_At*(1-eta_fin)),2); %total fin effciency

%air side frontal area

Afr_a = mf_a/(G_a*f_area); %air side frontal area

%tube side heat transfer coefficient

qual_r = 0.8; %quality of refrigerant
```

```
vel_r = 0.05; %velocity of refrigerant


G_r = rho_rl*vel_r; %mass flux of refrigerant

Re_rl = (G_r*di*(1-qual_r))/mu_rl;

Re_rg = ((G_r*di*qual_r)/mu_rg)+ Re_a;

Re_eq = Re_rg*(mu_rg/mu_rl)*(rho_rl/rho_rg)^(0.589) + Re_rl +
Re_a;

h_r = 0.05*(Re_eq^(0.8))*(pr_a^(0.33))*(k_r/di);

U_over = ((t_area/G_a)*(1/h_a) + (od/(2*k_alum))*log(od/di) +
1/(toteta_fin*h_r))^(-1);

%air heat balance

C_min = mf_a*cp_a;

eff = (Tout_a-Tin_a)/(Tr-Tin_a);

C_max = Inf;

%C_min/C_max = 0;

NTU = 0.5;

Asurf_a = (NTU*C_min)/U_over;
```

```
%heat exchanger size

tot_vol = Asurf_a/(t_area+2);

req_totvol = tot_vol/ntp;


L_2 = req_totvol*Afr_a*10;

%A_fluid

A_fluid = mf_r/(rho_rl*(vel_r+2.3))*1e4;

Afr_f = (A_fluid/f_area)*1e-2;

L_1  = req_totvol/Afr_f;

L_3 = req_totvol/(L_1*L_2);


%pressure drop

%cross flow

%required power for motor to air flow

vel_maxa = 18;
```

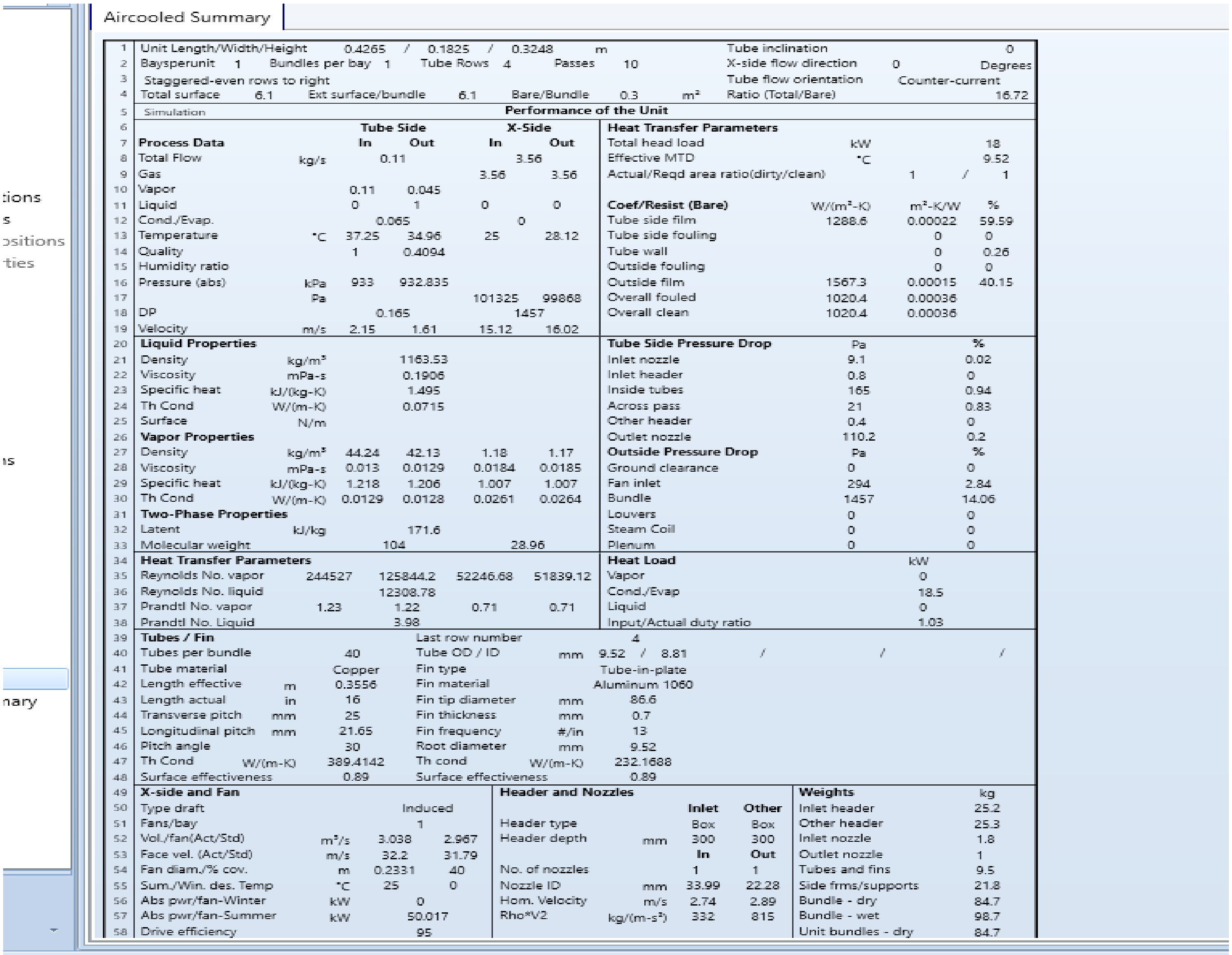
```
num_row = 4;

Re_ma = 4.8e4;

%from the graph

frp_a = 1.5;

del_pa = frp_a*0.5*rho_a*vel_maxa^(2)*num_row;

P_motor = del_pa*mf_a/rho_a;

P_motor = P_motor/746;

%considering 85% efficiency

P_motor_req = P_motor/0.85;

%power required to pump r134a

Re_mr = 1.8e5;

rough_cu = 1.5e-6;

%from moody diagram

frp_r = 0.05;

vel_maxr = 1.7;
```

```
del_pr = (frp_r*L_1/di)*0.5*rho_rg*(vel_maxr)^(2);

%compressor power needed

P_comp = (mf_r/rho_rg)* del_pr*100;
```

Workspace

Name	Value
A_fluid	0.4025
Af_At	0.6000
Afr_a	0.9225
Afr_f	0.0671
Asurf_a	1.9510
C_max	Inf
C_min	3600
cp_a	1010
cp_rl	1497
del_pa	1.1470e+03
del_pr	127.7399
dh	0.0020
di	0.0088
eff	0.4167
eta_fin	0.7010
f_area	0.0600
fin_pi	13
fin_t	7.0000e-04
fr_a	0.0200
frp_a	1.5000
frp_r	0.0500
G_a	64.4000
G_r	58.1500
h_a	1.4438e+03
h_fg	165300
h_r	1.2020e+03
hta_v	51.1361
k_a	0.0262
k_alum	232
k_r	0.0710
L_1	0.3504
L_2	0.2168
L_3	0.3094
l_fin	0.0064
main_p	1
mf_a	3.5644
mf_r	0.1100
ml_fin	1.1807
mu_a	1.8400e-05
mu_rg	1.2500e-05
mu_rl	1.8900e-04
nt	40
ntp	10
NTU	0.5000
num_row	4
od	0.0095
P_comp	31.6117

Name	Value
main_p	1
mf_a	3.5644
mf_r	0.1100
ml_fin	1.1807
mu_a	1.8400e-05
mu_rg	1.2500e-05
mu_rl	1.8900e-04
nt	40
ntp	10
NTU	0.5000
num_row	4
od	0.0095
P_comp	31.6117
p_l	0.0217
P_motor	4.6442
P_motor_req	5.4637
p_satr	933000
p_t	0.0250
pr_a	0.7000
pr_rl	3.9100
Q	18000
qual_r	0.8000
Re_a	7000
Re_eq	2.5540e+04
Re_ma	48000
Re_mr	180000
Re_rg	3.9787e+04
Re_rl	542.1180
req_totvol	0.0235
rho_a	1.1800
rho_rg	44.4500
rho_rl	1163
rough_cu	1.5000e-06
sig_w	0.1126
single_f_area	5.4125e-04
t_area	6.3000
t_film	300.6500
t_thick	3.6000e-04
Tin_a	298.1500
tot_vol	0.2351
toteta_fin	0.8200
Tout_a	303.1500
Tr	310.1500
U_over	922.5875
vel_maxa	18
vel_maxr	1.7000
vel_r	0.0500



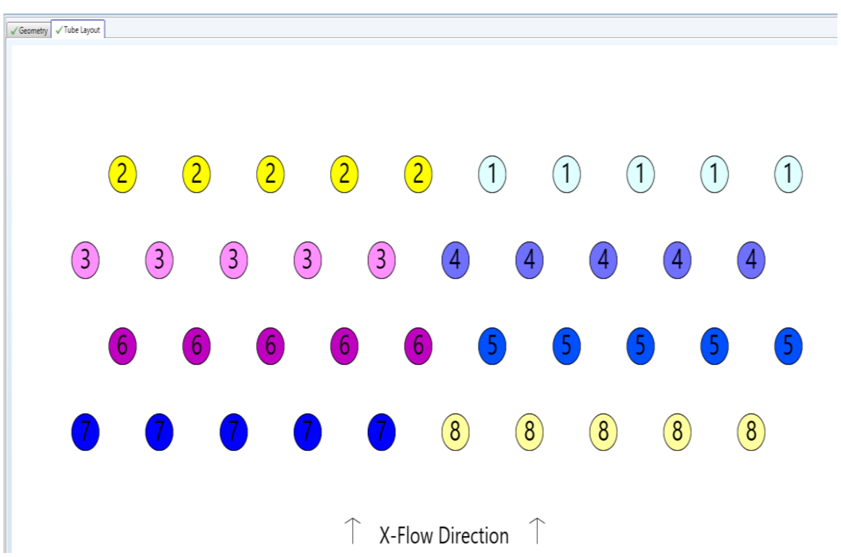
Aircooled Summary

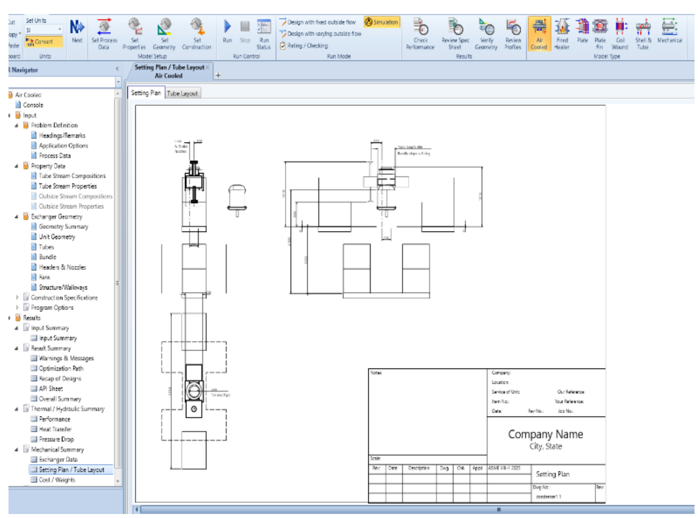
#											
1	Unit Length/Width/Height	0.4265	/	0.1825	/	0.3248	m	Tube inclination			0
2	Baysperunit	1	Bundles per bay	1	Tube Rows	4	Passes 10	X-side flow direction	0		Degrees
3	Staggered-even rows to right							Tube flow orientation	Counter-current		
4	Total surface	6.1	Ext surface/bundle	6.1	Bare/Bundle	0.3	m²	Ratio (Total/Bare)			16.72
5	Simulation				Performance of the Unit						

#			Tube Side In	Tube Side Out	X-Side In	X-Side Out					
6			Tube Side		X-Side		**Heat Transfer Parameters**				
7	**Process Data**		In	Out	In	Out	Total head load	kW			18
8	Total Flow	kg/s	0.11		3.56		Effective MTD	°C			9.52
9	Gas				3.56	3.56	Actual/Reqd area ratio(dirty/clean)		1	/	1
10	Vapor		0.11	0.045							
11	Liquid		0	1	0	0	**Coef/Resist (Bare)**	W/(m²-K)	m²-K/W		%
12	Cond./Evap.		0.065		0		Tube side film	1288.6	0.00022		59.59
13	Temperature	°C	37.25	34.96	25	28.12	Tube side fouling		0		0
14	Quality		1	0.4094			Tube wall		0		0.26
15	Humidity ratio						Outside fouling		0		0
16	Pressure (abs)	kPa	933	932.835			Outside film	1567.3	0.00015		40.15
17		Pa			101325	99868	Overall fouled	1020.4	0.00036		
18	DP		0.165		1457		Overall clean	1020.4	0.00036		
19	Velocity	m/s	2.15	1.61	15.12	16.02					
20	**Liquid Properties**						**Tube Side Pressure Drop**	Pa			%
21	Density	kg/m³		1163.53			Inlet nozzle	9.1			0.02
22	Viscosity	mPa-s		0.1906			Inlet header	0.8			0
23	Specific heat	kJ/(kg-K)		1.495			Inside tubes	165			0.94
24	Th Cond	W/(m-K)		0.0715			Across pass	21			0.83
25	Surface	N/m					Other header	0.4			0
26	**Vapor Properties**						Outlet nozzle	110.2			0.2
27	Density	kg/m³	44.24	42.13	1.18	1.17	**Outside Pressure Drop**	Pa			%
28	Viscosity	mPa-s	0.013	0.0129	0.0184	0.0185	Ground clearance	0			0
29	Specific heat	kJ/(kg-K)	1.218	1.206	1.007	1.007	Fan inlet	294			2.84
30	Th Cond	W/(m-K)	0.0129	0.0128	0.0261	0.0264	Bundle	1457			14.06
31	**Two-Phase Properties**						Louvers	0			0
32	Latent	kJ/kg		171.6			Steam Coil	0			0
33	Molecular weight		104		28.96		Plenum	0			0
34	**Heat Transfer Parameters**						**Heat Load**		kW		
35	Reynolds No. vapor		244527	125844.2	52246.68	51839.12	Vapor		0		
36	Reynolds No. liquid			12308.78			Cond./Evap		18.5		
37	Prandtl No. vapor		1.23	1.22	0.71	0.71	Liquid		0		
38	Prandtl No. Liquid			3.98			Input/Actual duty ratio		1.03		

#									
39	**Tubes / Fin**			Last row number		4			
40	Tubes per bundle		40	Tube OD / ID	mm	9.52 / 8.81	/	/	/
41	Tube material		Copper	Fin type		Tube-in-plate			
42	Length effective	m	0.3556	Fin material		Aluminum 1060			
43	Length actual	in	16	Fin tip diameter	mm	86.6			
44	Transverse pitch	mm	25	Fin thickness	mm	0.7			
45	Longitudinal pitch	mm	21.65	Fin frequency	#/in	13			
46	Pitch angle		30	Root diameter	mm	9.52			
47	Th Cond	W/(m-K)	389.4142	Th cond	W/(m-K)	232.1688			
48	Surface effectiveness		0.89	Surface effectiveness		0.89			

#										
49	**X-side and Fan**				**Header and Nozzles**				**Weights**	kg
50	Type draft		Induced				Inlet	Other	Inlet header	25.2
51	Fans/bay		1		Header type		Box	Box	Other header	25.3
52	Vol./fan(Act/Std)	m³/s	3.038	2.967	Header depth	mm	300	300	Inlet nozzle	1.8
53	Face vel. (Act/Std)	m/s	32.2	31.79			In	Out	Outlet nozzle	1
54	Fan diam./% cov.	m	0.2331	40	No. of nozzles		1	1	Tubes and fins	9.5
55	Sum./Win. des. Temp	°C	25	0	Nozzle ID	mm	33.99	22.28	Side frms/supports	21.8
56	Abs pwr/fan-Winter	kW	0		Hom. Velocity	m/s	2.74	2.89	Bundle - dry	84.7
57	Abs pwr/fan-Summer	kW	50.017		Rho*V2	kg/(m-s²)	332	815	Bundle - wet	98.7
58	Drive efficiency		95						Unit bundles - dry	84.7





Items	Price per Unit	cost(BDT)
Copper Tubes---X40	110Tk/feet	40*110 =4400
Fins (Al)----------X156	10Tk/piece	10x156 =1560
Fan--------------X1	400Tk/piece	400x1 =400
Motor (10W)	300Tk	300
Aluminum Shell	600Tk	=600
R 134a refrigerant	500Tk	=500
Assembly	1000Tk	=1000
Total		=8760

- Heat Duty =18 kW
- Tube OD =9.52 mm
- Core Length =0.4265 m
- Core Width =0.1825 m
- Core height = 0.3248 m
- Tube effective length =0.3556 m
- No of tubes = 40
- No of passes =10
- No of fins =13 fins /inch
- Fan power = 8.2 watt
- Pump Power =36 watt

Plate fin heat exchangers offer several advantages over competing designs.

(1) High thermal effectiveness and close temperature approach. (Temperature approach as low as 3K between single phase fluid streams and 1K between boiling and condensing fluids is common),

(2) Large heat transfer surface area per unit volume (Typically 1000 m^2/m^3),

(3) Low weight,

(4) Multi-stream operation (Up to ten process streams can exchange heat in a single heat exchanger.),

(5) True counter-flow operation (Unlike the shell and tube heat exchanger, where the shell side flow is usually a mixture of cross and counter flow.).

(6) Lower air turbulence and less fouling results in a lower air-side pressure drop, which is an important advantage for industries like Pulp & Paper, Food Processing, and Oil & Gas

(7) Plate style fins have up to 40% more secondary fin surface area

(8) More capacity than an equivalently configured coil.

ADVANTAGES AND LIMITATIONS:

The principal disadvantages of the plate fin geometry are:

(1) Limited range of temperature and pressure,

(2) Difficulty in cleaning of passages, which limits its application to clean and relatively non-corrosive fluids,

(3) Difficulty of repair in case of failure or leakage between passages, and

(4) Less efficient than plate heat exchangers.

Minimizing the cost is an important issue for our final product .We could have done our project more accurately if the fins were more uniform, uniform fin spacing ,uniform tube sizing, perfect brazing , more air tight .We have avoided the fouling factor in our project, which could cause some errors besides uniform flow rate inside the tubes should be ensured .Fin spacing should be accurate, outside pressure and air temperature should be stable .Anyway, we have always tried to do the project in a better way and give out best possible effort in this project.