ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Design Analysis of Heat Exchanges Using FEM

Taranisen Barik
M.Tech (Thermal Engineering) Student
Department of Mechanical Engineering
Aditya Institute of Technology and Management,
Tekkali.

Dr. D. SREERAMULU
Professor
Department of Mechanical Engineering
Aditya Institute of Technology and Management,
Tekkali.

ABSTRACT

Refrigeration and air-conditioning systems play an important role in applications like institutions, human comfort and industries like chemical industries, cold storage, etc. in recent days. The main components of the refrigeration and air-conditioning systems are compressor, condenser, evaporator and expansion valve. Among all these, the compressor is the heart of the systems.

A calorimeter is designed to measure the heat transfer between the refrigerants and is a device used to test the compressor in terms of the parameters like cooling capacity, power consumption, operating voltage range and motor winding temperature of the compressors under the controlled load conditions. The main components of the calorimeter are shell & tube condenser, expansion valve, secondary evaporator pot and return gas control pot (Desuperheater). The existing calorimeter of 1.5 Ton is modified and designed as per the prerequisite refrigeration capacity of 2 Ton.

The present work deals with the design and analysis of water-cooled shell and tube condenser. The condenser is designed using theoretical procedures. The condenser is modeled in UNIGRAPHICS using the dimensions obtained from the design procedures. Then thermal analysis is carried out in ANSYS. The results obtained through the analysis are discussed in detail and compared with experimental values.

INTRODUCTION

Water-cooled shell & tube condenser is an important component of the refrigeration and air-conditioning systems. The condenser removes the heat from refrigerant carried from evaporator and added by compressor and convert the vapor into liquid refrigerant. It is a heat exchanger in which heat transfer takes place from high temperature vapor refrigerant to low temperature water, which is used as cooling medium. These condensers are always preferred where adequate supply of clean and inexpensive means of water disposal are available.

Shell & tube condensers are those in which heat transfer occurs between two fluid streams, which do not mix or physically contact each other. The fluids so involved are separated from one another by a tube as well as wall, which may be involved in the heat transfer path. Heat transfer will thus occur by convection from the hot fluid surface, by conduction through the solid and again by convection from the solid surface to the cooler fluid.

Many types of condensers have been developed to meet the widely varying applications. " Shell & Tube" arrangements are often used, where heat transfer effectiveness and reliability are important. The present industrial services require watercooled shell & tube condenser, as the quantity of heat to be transferred is large. These condensers occupy considerable ground area. A combination of space, cost and pressure drop limitations usually results in a preference for the more compact "shell & tube condensers". These condensers consist of a bundle of tubes through which one fluid passes and this is enclosed by the shell, which contains the other fluid. If one of the fluids is to be condensed or vaporized, it is generally introduced to shell side. The tube side fluid may make numerous passes of the bundle, but shell fluid flows through the shell only once and therefore it is said to have one shell pass. The condensers may have more than one shell pass depending upon the requirements.

The shell side fluid may be forced to follow a devious path over the outside surface of the tubes by cross and longitudinal baffles inserted among the tubes. Shell & tube condensers can be connected in series or parallel either or both the streams.

Objective of the Present Work

The existing condenser of 1.5 Ton is modified and designed as per the prerequisite refrigeration capacity of 2 Ton. The present work deals with the design of shell and tube condenser using mathematical procedures and it is thermally analyzed. For this analysis, whole condenser model is selected as the computational domain and meshed with SOLID92 and SHELL63. The material and boundary conditions are applied and the domain is solved.

This powerful tool along with faster and robust digital computers makes it possible to predict temperature distribution at any point in the condenser tubes and shell and also the thermal stresses induced in the condenser.

The results obtained from the analysis are compared with experimental values. The experimental values at different condensing temperatures and specifications of the condenser are obtained from ASHRAE (American Society of Heating, refrigerating and Air-conditioning Engineers) and ARI (Air-conditioning and Refrigeration Institute). Temperatures and thermal stresses at various points are measured in the analysis.

CALORIMETER DESCRIPTION

The secondary system calorimeter has the useful compressor capacity measurement range of 1750 Watt (6000Btuh) to 7320 Watts (25000 Btuh). The calorimeter is capable of conducting various tests for the air-conditioning and heat pump compressors.

The Calorimeter needs enough space for operation and maintenance, adequate electric supply, chilled water supply and the dry nitrogen connection. Given below are the details.

1. Locate the calorimeter in a room in which the ambient temperature is controlled within +/- 2oC. The ambient temperature preferably should be controlled to about 27 to 28oC. This will ensure faster stabilization of all the test conditions. Moreover, the control of various parameters will be more precise if the room ambient temperature is controlled within +/-2oC.

2. A distance of at least 2000 mm should be provided between the back and two sides of the calorimeter and the adjacent walls for inspection and regular maintenance

3.Connect the proper electric supply to the calorimeter. The calorimeter needs 400V 50Hz Three Phase Four Wire Supply. Use an Isolator Switch of at least 125A rating. Connect secure earthing to the calorimeter. The Constant Frequency Drive and Auto Dimmerstat for the compressor frequency and voltage respectively, are to be located outside the main calorimeter frame. The frequency drive has 3 phase input and single-phase output. Connect the cable between the compressor MCB (4F2), frequency drive, auto Dimmerstat and back to the control panel. The cable should be at least 6-sq.mm size with multi-strand copper conductor.

4.The calorimeter needs Dry Nitrogen Supply for the operation of the expansion valve. The pressure required at the ER3000 valve is 120psig. The ER3000 Valve is located inside the Evaporator Pot Chamber.

The consumption of the nitrogen or the air is very small.

COMPRESSOR APPLICATION

AW Series of compressors are being widely used in vapor compression systems for the following appliances:

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

• Air conditioners:
- Window units
- Spit Systems
- Packaged units
• Heat Pumps
• Dehumidifiers
• Walk-in-Coolers
• Others

These compressors are manufactured from the best quality material and with highest degree of precision and workmanship.

COMPRESSOR COOLING

Ironically major problem for refrigeration compressor is heating. Heat is generated in the compressor due to electrical losses in the motor winding, frictional losses due to moving parts and the heat of compression. This heat will be conducted and converted to all parts of the compressor refrigerant gas and lubricating oil.

A part of this will be compensated by the cool return gas and the remaining part of the heat is rejected to the atmosphere air through compressor shell. As a result of heat generation and transmission, the temperature of the compressor stabilizes at certain values. Winding temperature is the most critical parameter of the hermetic compressor. Insulation used for the motor winding in AW hermetic compressors can withstand a temperature of 130oC. However, it is recommended to limit the winding temperatures below 120oC from the safety point of view.

Hence for longer and better operation of AW hermetic compressors it is essential to limit the winding temperatures by minimizing the heat generation within the compressor and effectively rejecting it to the atmosphere. Therefore the control parameters are:

1. Electrical losses in the windings
2. Frictional losses in the moving parts
3. Heat of compression
4. Return gas temperature
5. Air flow over the compressor

ELECTRICAL DETAIL

Supply (entire equipment) : 400v 50 hz three phase four wire.
Maximum pawer drawn : 25 hp excluding the chiller
Supply point size required : 400v 50hz three phase four wire, 100a

COMPRESSOR ONLY ELECTRICAL POWER SUPPLY.

Supply Voltage, Nominal : 230V 50Hz
Max. Continuous Current : 15A
Voltage Range : 70V to 300V

DETAILS OF PRIMARY REFRIGERATION SYSTEM

Condenser : Water-Cooled, Tube-in-Tube.
Liquid Line Filter/Drier : Provided.
Sub-cooler : Encased in the pot-2.
Liquid Line Sight Glass : Provided at two places.
Expansion Valve : Pneumatically operated (automatic operation)
Evaporator : Immersed in the secondary pot-1.
Super heater : Encased in pot-3.
HP/LP cutout : Provided.

RATED CONDITIONS

The rated conditions of AW compressors are: (According to ASHRAE)

OPERATING CONDITIONS (R-22)	UNIITS		R-22	
	MKS	FPS	MKS	FPS
Evaporating Temperature	$^{o}C \pm 0.5$	$^{o}F \pm 1$	7.2	45
Condensing Temperature	$^{o}C \pm 1$	$^{o}F \pm 1.8$	55	131
Liquid sub cooled temperature	$^{o}C \pm 1$	$^{o}F \pm 1.8$	46	115
Return gas temperature	$^{o}C \pm 1$	$^{o}F \pm 1.8$	35	95
Ambient temperature	$^{o}C \pm 1$	$^{o}F \pm 1.8$	35	95
Suction pressure (Gauge)	Kgf/cm^2	Lbf/in^2	5.3	76
Discharge pressure (Gauge)	Kgf/cm^2	Lbf/in^2	21	300

Condenser Design:

DESIGN OF WATER-COOLED SHELL AND TUBE CONDENSER

Known Parameters: -
Refrigerant Capacity : 2 Ton

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Refrigerant Used : Freon-22
Condensing Temperature : 54.40oC
Inlet temperature of water, twi : 23.53oC
Outlet temperature of water, two : 33.96oC
Temperature rise of water, Δtw : 10.43oC
Water tube outer diameter, dwo : 15.875 mm
Water tube inner diameter, dwi : 14.875 mm

a) Finding condensing heat transfer coefficient for the vapor condensing on outside of the horizontal tubes, h_o: -

We know that,

$h_o = 0.725 [(k_f^3 \rho_f^2 g h_{fg})/(N d_O \mu_f \Delta t)]^{0.25}$

Where, h_o = Condensing film coefficient of heat transfer.
k_f = Thermal conductivity of the condensing film.
ρ_f & μ_f = Density and Viscosity of the condensate film.
Assuming the temperature drop through the condensing film as 5°C.
At the condensing temperature (54.4°), the properties of the liquid refrigerant (R-22) are obtained from EFPROP.
k_f = 0.07136 W/m-K
μ_f = 1.2 x 10-4 kg/m-sec
h_{fg} = 151.6 kJ/kg (Latent heat of vaporization)
Assuming the total number of tubes are 28 and the number of rows are 8.
Then the average number of tubes per row, N = 28/8 = 3.5
Δt = 5°C
d_{wo}= 15.875 mm or 0.015875 m
ρ_f = 1072 kg/m3
Condensing coefficient,
$h_o = 0.725 [(0.07136^3 \times 1072^2 \times 9.81 \times 151600)/(3.5 \times 0.015875 \times 1.2 \times 10^{-4} \times 5)]^{0.25}$
$\therefore h_o$= 1506 W/m^2-k

b) Heat rejected in the condenser, Q_k: -

Heat rejected in the condenser, Q_k= 2 x 3.5 = 7 kW

c) Mass flow rate of the water, m_w: -

Mass flow rate of the water, $m_w = Q_k/(C_{pw} \times \Delta t_w)$
Where, C_{pw} = Specific heat of the water.
= 4.2 kJ/kg-K
Δ_{tw} = Temperature rise of the water in the condenser =10.43oC
Then, m_w = 7000/4200 x 10.43
mw = 0.1598 kg/sec

d) Finding water-side heat transfer coefficient, h_i: -

Bulk mean temperature, t_m = Inlet temperature of the water +(temperature rise/2)
= 23.53+(10.43/2)
t_m = 29°C
At this temperature, the properties of the water are
k_w = 0.618 W/m-k
μ_w = 0.797 x 10-3 kg/m-sec
Prandtls no., Pr = $(C_{pw} \times \mu_w)/k_w$
= $(4.2 \times 1000 \times 0.797 \times 10^{-3})/0.618$
$\therefore$ Pr = 5.42
Reynolds no., Re = $(\rho_w \times v_w \times d_{wi})/\mu_w$
Where,
v_w = Velocity of the water in the tubes .
We know that, volume flow rate of the water, V = $A_i \times v_w$
or $v_w = (m_w/\rho_w)/A_i \times f$
Where, f = Number of tubes per pass.
A_i = Cross sectional area of the water tube.
Here, the numbers of tubes are 28 and the numbers of passes are 8.
Then the velocity of the water is given as
$v_w = 0.1598 \times 10^{-3} / ((\Pi/4) \times 0.014875^2 \times 28/8)$
v_w = 0.263 m/s
Reynolds no., Re = $(1000 \times 0.4247 \times 0.014875)/0.797 \times 10^{-3}$
Therefore, Re = 4908.56 > 2100 So the flow is turbulent.
We know that, the waterside coefficient, (for turbulent flow)
Nusselt number, $N_u = (h_i \times d_i)/k_w = 0.023 \times (Re)^{0.8} \times (Pr)^{0.4}$
$= 0.023 \times (4908.56)^{0.8} \times (5.42)^{0.4}$
$N_u = 30.48 = (h_i \times 0.014875)/0.618$
$h_i = (0.618 \times 49.107)/0.014875$
Therefore, h_i = 1308 W/m^2-k

e) Finding the Overall heat transfer coefficient, U_o: -

We know that the overall heat transfer coefficient is
$1/U_O = 1/h_o + 1/h_i$
=1/1506 +1/1308
Therefore, U_o = 700 W/m^2-K

f) Log-mean temperature difference (LMTD): -

Refrigerant (R-22) temperatures,
Condenser outlet temperature, T =54.4^{O}C
Water temperatures:
Inlet, t_{wi} = 23.53°C

outlet, t_{wo} = 33.96°C
LMTD, $\Delta t_m = [(T-t_{wi}) - (T-t_{wo})]/[\ln(T-t_{wi}/T-t_{wo})]$
Δt_m = (52.03-23.53)-(52.03-33.96)/(ln (30.87/20.44))
Therefore, Δt_m = 25.29°C

g) Outside or bare tube surface area: -

We know that $\mathbf{Q_k = U_O \times A_O \times \Delta t_m}$
or also $A_O = Q_k/(U_O \times \Delta t_m)$
$= 7 \times 10^3/(700 \times 25.29)$
$A_O = 0.3954\ m^2$

h) Finding the Length of the tube, L: -

Length of the tube, $L = A_O/(n \times \Pi \times d_{wo})$
$L = 0.3954/(28 \times \Pi \times 0.015875)$
L = 0.2832 m or 283.2 mm
Total length of the cooling water tube, $L_t = n \times L$
$= 28 \times 0.2832$
∴ L_t = 7.9296 m or 7929.6 mm

i) Area calculations: -

Cooling water outside tube surface area, $A_o = \Pi \times d_{wo} \times L_t$
$= \Pi \times 0.015875 \times 7.9296$
$A_o = 0.3954\ m^2$
Cooling water Inside tube surface area, $A_i = \Pi \times d_{wi} \times L_t$
$= \Pi \times 0.014875 \times 7.9296$
$A_i = 0.3706\ m^2$

j) Finding the area of the condenser: -

Assuming, Inside diameter of the condenser, D_i = 200 mm
Outside diameter of the condenser, D_o= 225 mm
Length of the condenser, H = 450 mm
Inside Surface Area of the condenser, $A_c = \Pi \times D_i \times H$
$= \Pi \times 0.200 \times 0.450$
$A_c = 0.2827\ m^2$

k) Cover Plate Dimensions:

The numbers of cover plates are 2.
Diameter of the cover plate, d_c = 225 m
Thickness of the cover plate, t = 30 mm.
Number of bolts for each plate = 8
Diameter of the bolt = 8 mm
Length of the bolt = 45 mm

Design Parameters: -

Capacity : 2 Ton
Refrigerant : R-22
Condensing temperature : 54.4°C
Outside diameter of the water tube : 15.875 mm
Inside diameter of the water tube : 14.875 mm
Total number of tubes : 28
Mass flow rate of the cooling water, m_w : 0.1598 Kg/sec
Velocity of the cooling water, v_w : 0.263 m/sec
Overall heat transfer coefficient, U_o : 700 W/m2-K
Total bare-tube surface area, A_t : 0.3954 m2
Length of the one tube : 283.2 mm
Total length of the tube, L_t : 7.93 m
Inside Diameter of the condenser, D_i : 200 mm
Outside diameter of the condenser, D_o : 225 mm
Length of the condenser : 450 mm
Surface area of the condenser : 0.3935 m2
Diameter of the cover plate : 225 mm
Thickness of the cover plate, t : 30 mm
Diameter of the bolt : 8 mm
Length of the bolt : 45 mm
Total no. of bolts : 16
Condenser material : Steel
Tube material : Copper

DESIGN OF THE RETURN GAS CONTROL POT (DESUPERHEATER)

Known parameters: -

Capacity = 2 Ton
Outer diameter of the chiller tube, d_o = 22 mm
Inner diameter of the chiller tube, d_i = 21 mm

Cooling medium (R-404A) temperatures:

Inlet, t_1 = 5°C
Outlet, t_2 = 12°C
Temperature rise, Δt = 7°C
Temperature of the secondary refrigerant (R-134a), T = 35°C

a) Finding the heat transfer coefficient for vapour condensing on outside of the chiller tubes (coil), h_o:

We know that the secondary refrigerant (R-134a)- side heat transfer coefficient, ho is given as

$h_o= 0.725 \{[\rho^2 k^3 g h_{fg}]/ [\mu \Delta t N d_o]\}^{0.25}$

Where,

ρ = Density of the condensate film (i.e. R-134a) in kg/m^3.

μ= Viscosity of the condensate film in kg/m-sec.

k = Thermal conductivity of the condensate film in W/m-K.

g = Acceleration due to gravity in m/sec^2 = 9.81 m/sec^2

Δt = Temperature drop through the condensate film in ^oC.

N = Number of tubes (turns) in a vertical row.

Here we are assuming the number of turns of the chiller coil in the control pot are 6 and temperature drop through the condensing film is 5°C and the temperature of the secondary refrigerant (R-134a) is 35°C.

At the temperature of the secondary refrigerant (i.e. 35°C), the properties of the liquid refrigerant (R-134a) are obtained from **REFPROP.**

$\rho = 1168\ kg/m^3$

$\mu = 0.1743 \times 10^{-3}$ kg/m-sec.

k = 0.07685 W/m-K

$\Delta t = 5^oC$

$h_{fg} = (h_g – h_f)$ =(417.2 - 249) =168.2 kJ/kg – K

Then,

$h_o = 0.725[(1168^2 \times 0.07685^3 \times 9.81 \times 168.2 \times 10^3)/(0.1743 \times 10^{-3} \times 5 \times 6 \times 0.022)]^{0.25}$

Therefore, h_o = 1259.32 W/m^2–K

Refrigerant Capacity, Q = 2 Ton = 2 x 3.5 x 10^3 = 7 kW

b) Finding the mass flow rate of the cooling medium (R-404A), m_c: -

The mass flow rate of the cooling medium (R-404A) flowing through the chiller coil is given as

$m_c =[Q/(C_{pc}\Delta t)]$

Where,

C_{pc} = Specific heat of the cooling medium at 8.5°C = 1.425 kJ/kg-K

Δt = Temperature rise of the cooling medium = 7 °C

Q = 7 kW

$m_c = (7 \times 10^3)/ (1.425 \times 10^3 \times 7)$

Therefore, m_c = 0.7017 kg/sec

c) Finding the cooling medium (R-404A)– side heat transfer coefficient, h_i: -

Bulk mean temperature, t_m = Inlet temperature of the refrigerant+(temperature rise)/2

= 5 + (12 - 5)/2

∴ t_m = 8.5°C

At this temperature (i.e. 8.5°C), the properties of the cooling medium (R-404A) are

kc = Thermal conductivity of the cooling medium (R-404A) = 0.07423 W/m-K

μc = Viscosity of the cooling medium (R-404A) = 0.1586 x 10-3 kg/m-sec

ρc = Density of the cooling medium (R-404A) = 1118 kg/m3

Prandtl number, Pr =$(Cpc \times \mu_c)/ k_c$

= $(1.425 \times 10^3 \times 0.1586 \times 10^{-3})/(0.07423)$

∴ Pr = 3.045

Reynolds number, Re = $(\rho_c \times v \times d_i) / \mu_c$

Where, v = velocity of the cooling medium (R-404A)

We know that,

Volume flow rate of the cooling medium (R-404A),

V = A_i x v x (no. of tubes / no.of passes)

Where,

A_i = Cross-sectional area of the tube = $\pi/4 \times di^2$

The number of passes is1.

Therefore,

$(m_c/\rho_c) = \pi/4 \times di^2 \times v \times$ (no.of tubes / no.of passes) Or

v = $(m_w/ \rho_w)/ [\pi/4 \times di^2 \times$ (no of tubes / no.of passes)]

= $(0.7017/1118)/ (\pi/4 \times 0.021^2 \times 6/1)$

∴ v = 0.333 m/sec

Then,

Reynolds number, Re = (1248 x 0.333 x 0.020) /$(0.1968*10^{-3})$

Therefore, R_e = 46947.5

We know that the nusselt no., $N_u = 0.023 (Re)^{0.8}(Pr)^{0.4} = (h_i d_i)/ k_w$

= $0.023 \times (46947.5)^{0.8} \times (3.045)^{0.4}$

=165.9 = $(h_i \times 0.021)/0.07423$

h_i = 615.795W/m^2 – K

d) Finding the overall heat transfer coefficient, U_o: -

Overall heat transfer coefficient, U_o is given as

$1/U_o = 1/h_i + 1/h_o$

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

Where,
h_i = 1259.32 W/m^2 – K
h_o = 615.795W/m^2 – K
Therefore,
U_o = 413.57 W/m^2 – K

f) Finding the log mean temperature difference, Δt_m: -

Log mean temperature difference, Δt_m is given as
$\Delta t_m = [(T- t_1) – (T - t_2)]/\ln [(T- t_1) /(T - t_2)]$
By substituting t_1, t_2 and T values in the above equation, we get
$\Delta t_m = 26.35^oC$

g) Finding the outside or bare tube surface area, A_t : -

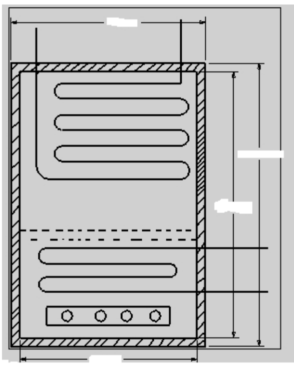
Capacity, Q = U_o x A_t x Δt_m
Or A_t = (7 x 10^3)/(413.57 x 26.35)
A_t = 0.5760 m^2
But,
At = n x π x d_o x L
Where, n = Number of turns of the chiller coil
L = Length of the tube (i.e. length of one turn of the coil)
Then, 0.5760 = 6 x π x 0.022 x L
Length of the one turn of the coil, L = 1.5493 m
Mean diameter of the coil, dm = L/ Π= 1.5493/Π = 0.493
m = 493 mm
Total length of the tube, L_t = 9.2958 m

e) Finding the surface area of the return gas control pot: -

Assuming the inside diameter (D_{Pi})& height (h_p) of the pot are
545 mm & 800 mm respectively.
Internal Surface area of the pot, $A_P = \pi$ x D_{Pi} x h_p
= π x 0.545 x 0.800
Therefore, A_P = 1.3722 m^2
Let the thickness of the pot be 12.5 mm.
Then the outside diameter of the pot, D_{Po} = 570 mm
And the total height of the pot, H = 825 mm
Total surface area of the pot = π x D_{Po} x H
= π x 0.570 x 0.825
= 1.4773 m^2

DESIGN DATA:

Capacity : 2 Ton Primary
refrigerant : R-22
Secondary refrigerant : R-134a
Cooling medium : R-404A
Cooling medium inlet temperature : 5OC
Cooling medium outlet temperature : 12oC
Secondary Refrigerant (R-134a) temperature : 35oC
Outer diameter of the chiller tube : 22 mm
Inside diameter of the chiller tube : 20 mm
Number of turns of the primary refrigerant coil : 2
Number of turns of the chiller coil : 6
Secondary Refrigerant- side heat transfer coefficient, ho : 1259.32 W/m2- K
Cooling medium (R-22)- side heat transfer coefficient, hi : 615.795W/m2- K
Overall heat transfer coefficient, Uo : 413.57W/m2- K
Outside or bare tube surface area, At : 0.5760 m2
Length of the tube, L : 1.5493m
Inside diameter of the pot, DPi : 545 mm
Height of the pot, hp : 800 mm
Internal Surface area of the pot, AP i : 1.3722 m2
Outside diameter of the pot, DPo : 570 mm
Total height of the pot, H :825 mm
Total surface area of the pot, APo : 1.4775 m2
Primary refrigerant coil & Chiller coil material : Copper
Return gas control pot material : Steel



DESIGN OF SECONDARY EVAPORATOR POT KNOWN PARAMETERS:

Capacity : 2 Ton

Evaporating Temperature, to **:** 7.2^{o}C

Outside diameter of the copper tube, d_o **:** 22.225 mm
Inside diameter of the copper tube, d_i **:** 21.225 mm

a) Finding the heat transfer coefficient on outside of the primary refrigerant (R-22) tubes, h_o: -

We know that the heat transfer coefficient on outside of the tubes (i.e. secondary refrigerant (R-134a) – side heat transfer coefficient)is given below h_o= 0.725 $[(\rho^2 k^3 g h^{fg})/(\mu \Delta t N d^i)]^{0.25}$

Where,
ρ = Density of the liquid refrigerant (R-134a), kg/m^3
k = Thermal conductivity of the secondary refrigerant (R-134a), W/m-K
g = Acceleration due to gravity, m/sec^2
h_{fg} = Latent heat of vaporization, kJ/kg
Δt = Temperature drop through the condensing film, ^{o}C = 5^{o}C Assuming the total number of turns of the coil are 12.
μ = Viscosity of the refrigerant, kg/m-sec
d_i =Outside diameter of the tube = 0.022225 m
At the evaporating temperature (i.e. 7.2^{o}C), the properties of the liquid refrigerant R-134a are obtained from the **REFPROP** as below.
ρ = 1271 kg/m^3
k = 0.08884 W/m-K
g = 9.81 m/sec^2
h_{fg}= h_g - h_f = 402.7- 209.7
h_{fg} =193 KJ/ kg
μ = 0.2474 x 10^{-3} kg/m - sec
Assuming the no. of tubes in a vertical row, N = 12 (i.e.number of turns of the evaporator coil) Then,
ho= 0.725 x {[12712 x 0.088843 x 9.81 x 193 x 103]/[0.2474 x 10-3 x 5 x 12 x 0.022225]}0.25
h_o = 1160.6 W/m^2-K

b) Finding mass flow rate of the primary refrigerant (R-22), mr: -

The mass flow rate of the primary refrigerant is given as
$m_r = [Q_e /(C_{pr} \Delta t)]$ Where,
C_{pr} = Specific heat of the primary refrigerant, kJ/kg-K
Δt = Temperature rise of the primary refrigerant
= **35** – 7.2
Δt = 27.8^{o}C

At the evaporating temperature (i.e.7.2^{o}C), the properties of the liquid refrigerant are obtained from the **REFPROP** as below.
C_{pr} = 1.24 kJ/kg-K
Q_e = 2 x 3.5 kW = 7 kW Then,
m_r = [(7 x 1000)/ (1.24 x 10^3 x 27.8)]
∴ **m_r = 0.2030 kg/sec**

c) Finding the velocity of the primary refrigerant (R-22), v: -

We know that**,**
Volume flow rate of the Refrigerant,
V = A_i x v x (no. of tubes / no.of passes)
(m_{pr}/ρ_{pr})= A_i x v x (no. of tubes / no.of passes)
Or v = (m_{pr}/ρ_{pr})/[A_i x (no. of tubes / no.of passes)]
Here, A_i = Cross sectional area of the tube
= Π/4 x d_i^2
= Π/4 x 0.021225^2
A_i = 314.16 x 10^{-6} m^2
Then the no. of passes is 1.
v = (0.2030/1257)/(314.16 x 10^{-6} x 12/1)
v = 0.0428 m/sec.

d) Finding the primary refrigerant – side heat transfer coefficient, h_i: -

The primary refrigerant- side heat transfer coefficient, hi is given
Nusselt number, N_u = 0.023 (Re)$^{0.8}$ (Pr)$^{0.4}$ = (h_i x d_i)/k
Where ,
Re = Reynolds number = $(\rho_r v d_i)/\mu_r$
Pr = Prandtl number = $(C_{pr} \mu_r)/k_r$
Bulk mean temperature, t_m is given as
t_m = Inlet temperature of water + (Temperature rise)/2
= 7.2 +27.8/2 = **21.1^{o}C**
At this temperature (i.e. 21.1^{o}C), the properties of the primary refrigerant are obtained from REFPROP as below.
k_{pr} = Thermal conductivity of the primary refrigerant = 0.08540 W/m-K
μ_{pr} = Viscosity of the primary refrigerant = 0.1732 x 10^{-3} kg/m-sec

1) Prandtl number, Pr: -

We know that, Pr = $(C_{pr}$ x $\mu_{pr})/k_{pr}$

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

=(1.24 x 103 x 0.1732 x 10-3)/ 0.854
∴ **Pr = 2.515**

2) Reynolds number, Re: -

We know that**,**
$\mathbf{Re = (\rho_{pr}\ v\ d_i)/\mu_{pr}}$
Where,
ρ_{pr} = Density of the primary refrigerant at 21.1°C = 1257 kg/m^3
v = Velocity of the primary refrigerant in m/sec = 0.0428 m/sec
d_i = 0.021225 m
μ_{pr} = 0.1732 x 10^{-3} kg/m-sec Then,
Re = (1257 x 0.0428 x 0.021225)/ 0.1732 x 10^{-3}
Re = 6212.4
Nusselts number, N_u = 0.023 $(6212.4)^{0.8}$ $(2.515)^{0.4}$ = (h_i d_i)/k_r
= 36.02 = (h_i x 0.02)/0.0854
Then we get,
h_i = 154 W/m^2- K

e) Finding the overall heat transfer coefficient, U_o: -

The overall heat transfer coefficient, Uo is given as
$1/U_o = 1/h_i + 1/h_o$
Where,
h_i = 154 W/m^2-K
h_o = 1160.6 W/m^2-K
$1/U_o$ = 1/151.6 + 1/ 1061.15
Or U_o = 136 W/m^2-K
f) Finding log mean temperature difference, (LMTD), ΔT_m: -
Log mean temperature difference, ΔT_m is given as
$\Delta T_m = \{[(t_1\text{-to}) – (t_2\text{-to})]/ [\ln(t_1\text{-to})/ (t_2\text{-to})]\}$ Where,
Taking the temperatures of the R-134a from the test repots,
t_1 = 43.5oC
t_2 = 38.5oC
Evaporating temperature, t_o = 7.2°C
ΔT_m = {[(43.5-7.2) – (38.5-7.2)/ ln [(43.5-7.2)/(38.5-7.2)]}
Therefore, **ΔT_m = 33.74°C**

g) Finding the outside or bare tube surface area, A_t: -

Evaporating Capacity, **$Q_e = U_o$ x A_t x Δt_m**
2 x 3.5 x 103 = 136 x At x 33.74
A_t = 1.5255 m^2
But, **A_t = n x π x do x L**
1.5255 = 12 x π x 0.022225 x L
Length of the tube (i.e. length of one turn of the evaporator coil)
L = 1.839 m
Mean dia of the coil, d_m = L/ π = 1.839/ π =0.5854 m =585.4 mm
Total length of the tube, L_t = n x L = 12 x 1.839 = 22.068 m

h) Area Calculations: -

Outside Tube surface area**, A_o= π x d_o x L_t**
= π x 0.022225 x 22.068
= 1.525 m^2
Inside tube surface area, **A_i = π x d_i x L_t**
= π x 0.021225 x 22.068
= 1.3866 m^2

i) Finding area of the secondary evaporator pot: -

Assuming the inside diameter of the pot, D_i = 638 mm
Height of the pot, h = 850 mm
Internal surface area of the pot, A_{pi} = π x D_i x h
= π x 0.638 x 0.85
= 1.704 m^2
Internal volume of the pot, V = π/4 x D_i^2 x h
= π/4 x 0.6382 x 0.85
V = 0.2717 m^3
Outer diameter of the pot, D_o = 663 mm
Total Height of the pot, H = 875 mm
Outside surface area of the pot, A_{po} = π x D_o x H
= π x 0.663 x 0.875
A_{po} = 1.823 m^2

DESIGN DATA: -

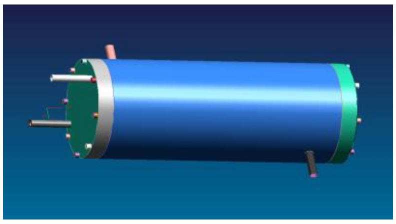
Capacity : 2 Ton
Primary refrigerant : R-22
Secondary refrigerant : R-134a
Evaporating temperature : 7.2oC
Outside diameter of the copper tube, do : 22 mm
Inside diameter of the tube, di : 20 mm

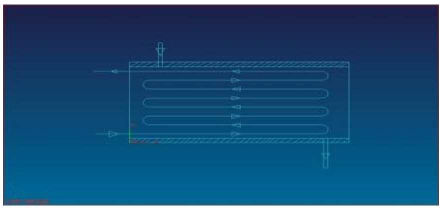
ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

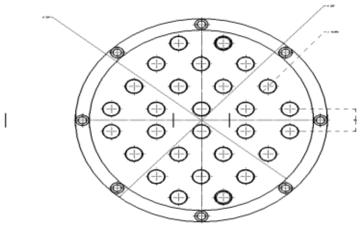
Total number of turns of the coil, n : 12
Mass flow rate of the primary refrigerant, mr : 0.2030 kg/sec
Velocity of the primary refrigerant, v : 0.0428 m/sec
Overall heat transfer coefficient, Uo : 133.4 W/m2-K
Total bare tube surface area, At : 1.56 m2
Length of the one turn of the coil, L : 1880 mm
Total length of the coil : 22.068 m
Inside diameter of the pot, Di : 638 mm
Outside diameter of the pot, Do : 663 mm
Inside Height of the pot : 850 mm
Total Height of the pot, H : 875 mm
Internal surface area of the pot, Api : 1.704 m2
Internal volume of the pot, Vp : 0.2717 m3
Outside surface area of the pot, Apo : 1.823 m2
Tube material : copper
Pot material : steel

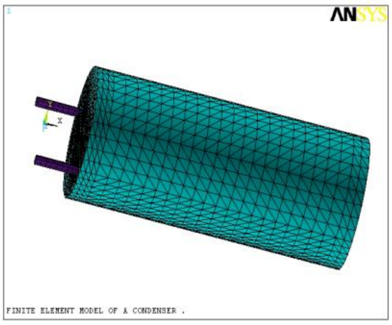
Result and discussion

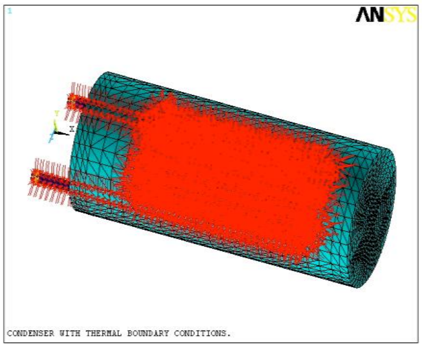
The adsorber is a component that includes a liquid shell-and- tubes heat exchanger. The choice of the geometric configuration was based on practical aspects, such as the simplicity of the manufacturing technology employed, and the required number of tubes, which depends on the amount of adsorbent they can hold. Thus, the chosen exchanger was that of a single pass, with a square arranged tube bank. Segmental baffles were installed on the shell in order to enhance turbulence in the water flow around the tubes. The following figure shows that the variation of temparatures in and around of the shell and tube heat exchanger.

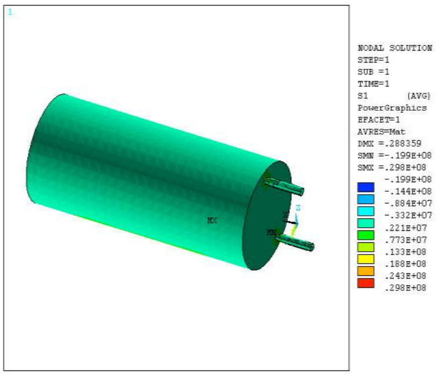




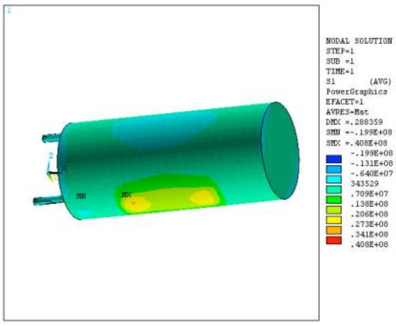


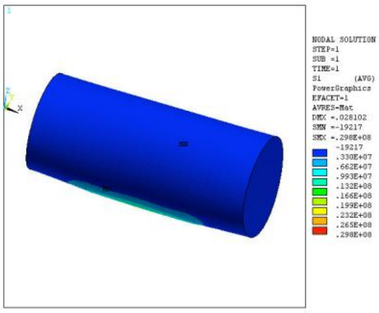


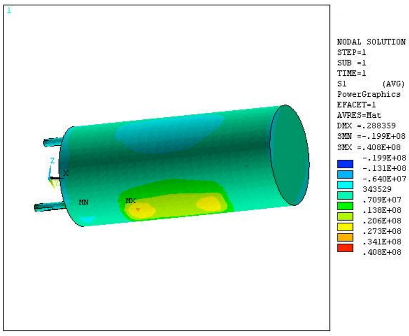


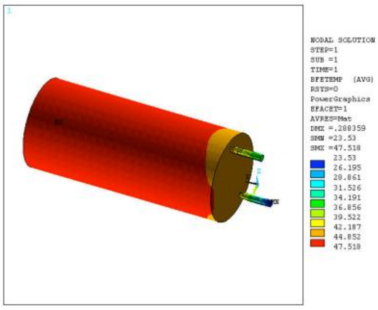


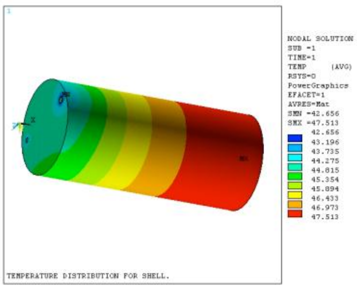
ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

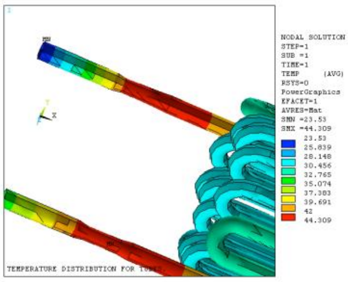


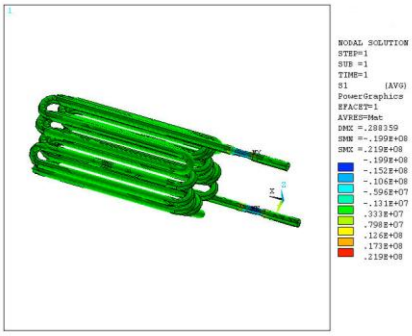


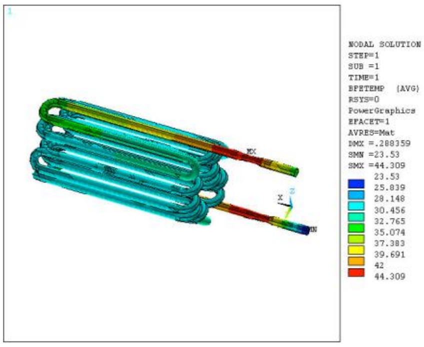












CONCLUSION

It was presented a description of the mathematical models to simulate the dynamic behavior of the shell-and-tube exchanger as an adsorber. The one-dimensional energy balance equations have been written, using the mixed finite-element formulation. The simulation results show that the system performance is strongly dependent on the operating conditions such as the operating temperatures, flow rates and cycle times. The performance of this exchanger is mostly influenced by

ISSN No: 2348-4845
International Journal & Magazine of Engineering, Technology, Management and Research
A Peer Reviewed Open Access International Journal

the cycle time; it needs long time of operation, resulting lower specific cooling power. This is due to its great volume of water inside the shell. For the present application, other faster exchangers (compacts) are under study to build a comparison and find out the most suitable exchanger.

REFERENCES

1. Bell, K.J., 1963, Final report of the cooperative research program on shell-and-tube heat exchangers. Bulletin No 5, University of Delaware Engineering Experimental Station, Newark, Delaware, USA.

2. Dubinin, M.M., Astakhov, V.A., 1971, Description of adsorption equilibria of vapors on zeolites over wide ranges of temperature and pressure, Adv. Chem. Series, Vol. 102, pp. 69.

3. Guilleminot, J.J., Meunier, F., Pakleza, J., 1987, Heat and mass transfer in a non- isothermal fixed bed solid adsorbent reactor: a uniform pressure/non-uniform temperature case. Int. J. Heat Mass Transfer, Vol. 30 (8), pp. 1595-1606

4. Halle, H., Cheoweth, J.M., Wambsganss, M.W., 1984, Shell side water flow pressure drop distributions measurements in an industrial-sized test heat exchanger. In: Proc. 22th ASME Heat Transfer Conf. and Exhibition, USA, pp. 39- 48.

5. Leite, A., Belo, F.A., Martins, M.M., Riffel, D.B. e Meunier, F., 2006, Central air conditioning based on adsorption and solar energy, In: Proc. of 2nd Int. Solar Cities Congress, Oxford, UK.

6. Leite, A.P.F., 1998, Thermodynamic analysis and modelling of an adsorption-cycle system for refrigeration from lowgrade energy sources. Journal of the Brazilian Society of Mech. Sci., Vol. 20, pp. 301-324.

7. Passos, E.F., 1986, Etude des couples charbon actif-methanol et de leur application à la refrigeration solaire. PhD Thesis, Ecole Polytechnique Fédérale de Lausanne, Swiss, p. 101.

8. Pletcher, L.S., Andrews, M.J., 1994, Technical/market assessment heat exchanger technology for users of natural gas. GRI Report GRI-94/0248.

9. Srivastava, N.C., Eames, I.W., 1998, A review of adsorbents in solid-vapor adsorption heat pump systems: Applied Thermal Engng., Vol. 18, pp. 707-714.

Author Details

Taranisen Barik
M.Tech (Thermal Engineering) Student
Department of Mechanical Engineering
Aditya Institute of Technology and Management, Tekkali

Professor **Dr. D. Sreeramulu** Was Born Srikakulam, Andhra Pradesh, India. He Has Received P.Hd (ME) And M.Tech(CIM) From NIT ,Warangal . HE IS WORKING AS Professor In Department Of Mechanical Engineering, Aditya Institute Of Technology & Management, Tekkali, AP