CHAPTER 6

CONDENSER CALCULATION

CONDENSER CALCULATION

6.1.1 Analysis options condensers

Condenser is used to transfer heat energy of refrigerating agent at high temperature for cooling substances. Go into the steam condenser is usually a bit too, so first of all it must be cooled to saturation temperature, and in the process of condensation, finally gets too cold a few degrees before out of the condenser.

Cooling way of condensers can be divided into two groups after:

+ Water-cooled condenser.

+ Condenser cooling by air.

In the works at "**VIETNAMESE-GERMAN UNIVERSITY (VGU)**" select the condenser water-cooled condensers because water-cooled condenser better cooling cooling by air.

6.1.2 Condenser design calculation

A. The initial data.

- Cooling capacity: **Q_0 = 1170,82 (kW)**

- *The temperature outside investigation under TCVN 5687-2010 "Design standards for ventilation And Air-conditioning"*.

$$t_N = 36^0C$$

- The water temperature in the condenser

$$t_{W1} = t_{ư} + (3\div5)^0C$$

We choose $t_{W1} = 32^0C$

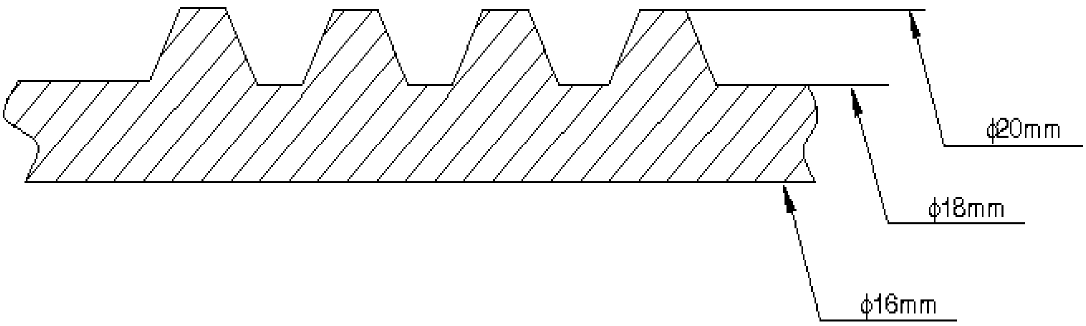
We choose: $\Delta t_W = 5^0C$

- Water temperature to condenser : $t_{W2} = 37^0C$

- Condensation temperature: $t_k = 42^0C$

Condenser heat transfer surface is chosen as copper pipes alternate layout, on the floor with the parameters:

- Impeller diameter D = 0,02 (m)
- External diameter of the pipe d_{ng} = 0,0181 (m)
- The internal diameter of the pipe d_{tr} = 0,016 (m)
- Step off S_c = 0,0018 (m)
- Tube Steps S = 0,026 (m)
- Wing thickness $\delta_đ$ = 0,0002 (m)
- Foot wing thickness δ_c = 0,0003 (m)



B. Calculate:

The other parameters of the beam pipe.

The stand surface area of 1 m pipe wing.

$$F_d = \frac{\pi(D^2 - d_{ng}^2)}{2S_c}$$

$$= 3{,}14 \times \frac{(0{,}02^2 - 0{,}0181^2)}{2 \times 0{,}0018}$$

$$= 0{,}0631(m^2/m)$$

- Horizontal surface area of 1 m pipe.

$$F_n = \pi \times d_{ng} \times \left(1 - \frac{\delta_c}{S_c}\right) + \frac{\pi \times D \times \delta_d}{S_c}$$

$$F_n = 3{,}14 \times 0{,}0181 \times \left(1 - \frac{0{,}0003}{0{,}0018}\right) + \frac{3{,}14 \times 0{,}02 \times 0{,}0002}{0{,}0018} = 0{,}0543(m^2/m)$$

- Outside Surface area of the wing pipe 1 m

$F_{ng} = F_n + F_d$

$= 0{,}0543 + 0{,}0631 = 0{,}1174\ (m^2/m)$.

- Inside surface area of the wing pipe 1 m

$$F_{tr} = \pi d_{tr} = 3{,}14 \times 0{,}016 = 0{,}05024 \ (m^2/m)$$

- The coefficient making wing

$$\beta = \frac{F_{ng}}{F_{tr}} = \frac{0{,}1174}{0{,}05024} = 2{,}34$$

The average temperature of condenser cooling water:

$$\Delta t = \frac{t_{w1} + t_{w2}}{2} = \frac{32 + 37}{2} = 34{,}5^0C$$

The thermal properties of water in the condenser in the average temperature

$t_w = 34{,}5^0C$

Look under Appendix 28 Reference [1] must be*:*

At a temperature $t = 30^0C$

$v = 0{,}805.10^{-6} \ (m^2/s)$

$\rho = 995{,}7 \ (kg/m^3)$

$Pr = 5{,}42$

$C_w = 4{,}174 \ (kJ/kgK)$

$\lambda = 0{,}617 \ (W/mK)$

At a temperature $t = 40^0C$

$v = 0{,}659.10^{-6} \ (m^2/s)$

$\rho = 992{,}2 \ (kg/m^3)$

$Pr = 4{,}31$

$C_w = 4{,}174 \ (kJ/kg)$

$\lambda = 0{,}6338 \ (W/mK)$

Using interpolation method is:

At a temperature $t = 34{,}5^0C$

$v = 0{,}7393.10^{-6} \ (m^2/s)$

$\rho = 994{,}125 \ (kg/m^3)$

$Pr = 4{,}92$

$C_w = 4{,}174 \ (kJ/kg)$

$\lambda = 0{,}6245 \ (W/mK)$

Thermal properties of R134a in the heat exchange condensing

$t_k = 42^0C$

$\gamma = 0.925.10^{-6}(m^2/s)$

$\rho = 1154,9(kg/m^3)$

$\lambda = 0,0759(W/mK)$

Load condenser temperature:

$$Q_k = \frac{Q_0}{q_0}.\Delta h = \frac{1170,82}{151,45}.176,07 = 1361,15(kW)$$

Cooling water passes through a condenser

$$G_w = \frac{Q_k}{C_w.\Delta t} = \frac{1361,15}{4,174x5} = 65,23(kg/s)$$

To calculate the heat transfer to the country choose the rate of water movement in the tube is ω = 2(m/s)

$$n_1 = \frac{4.G_w}{\pi.\rho.d_{tr}^2.\omega}$$

$$n_1 = \frac{4x65,23}{3.14x994,125x0,016^2x2} = 163$$

The pipe in a water line is 163

Water velocity reassessed according to n_1 = 163

We have:

$$\omega = \frac{4.G_w}{\pi.\rho.d_{tr}^2.n_1} = \frac{4x65,23}{3,14.994,125.0,016^2.163} = 2\ m/s$$

Reynold coefficient:

$$Re = \frac{\omega.d_{tr}}{v}$$

$$Re = \frac{2.0,016}{0,7393.10^{-6}} = 43284,18$$

From these results we have Re = 43284,18 > 10000 deduce this is the turbulent regime.

Nusselt coefficient:

$$Nu = 0,021.\ Re^{0.8}.\ Pr^{0.43}.\varepsilon_1$$

$$= 0,021.\ 43284,18^{0.8}.\ 4,92^{0.43}.\ 1 = 213,22$$

Exothermic coefficient in the water:

$$\alpha_w = \frac{N_u.\lambda}{d_{tr}} = \frac{213,22.0,6245}{0,016} = 8322,24$$

The average temperature difference of the logarithm

$$\theta_m = \frac{t_{W2} - t_{W1}}{ln\frac{t_k - t_{W1}}{t_k - t_{W2}}}$$

$$\theta_m = \frac{37 - 32}{ln\frac{42 - 32}{42 - 37}} = 7{,}21$$

The equation defines the density of heat flow to the water

$$q_W = \frac{\theta_m - \theta_a}{\frac{1}{\alpha_W} + \Sigma\frac{\delta_i}{\lambda_i}}$$

$\sum\frac{\delta_i}{\lambda_i}$: thermal impedance of the layer residue and duct wall. According Reference [1] pages 257, duct wall materials is brass we choose

$\sum\frac{\delta_i}{\lambda_i}$ = 0,2.10^{-3}(m^{2}K/W)

$$q_w = \frac{7{,}21 - \theta_a}{\frac{1}{8322{,}24} + 0{,}2.10^{-3}} = 3123{,}43.(7{,}21 - \theta_a)$$

To be able to determine the density of heat flow q_{tr}, the preliminary selection trick textures of condensers:

Preliminary we can choose: $\theta_a = 0.3\theta_m$

We have :

q_{tr}' = 3123,43.(7,21 – 0,3 × 7,21)

= 15763,95 (W/m^2)

Tubes are arranged on the floor according the edges of the hexagons and on the top of the triangle. So the number of pipes, arranged diagonally along the hexagon of the *m* can be determined according to the formula:

According to reference [2] pages 161 we have

$$m = 0.75\sqrt[3]{\frac{Q_k}{q'_{tr}.s.d_{tr}.k}}$$

In which:

S: Tube steps

D: Diameter of impeller

D = 0,02 (m)

S = 1,3. D = 1,3.0,02 = 0,026 (m)

We choose: k = l/d = 8

We deduce:

$$m = 0,75.\sqrt[3]{\frac{1361,15.10^3}{15763,95.0,026.0,016.8}} = 22,2$$

From the results we choose: m = 22

So the vertical tube caves are: n_z = m = 22

Exothermic condensation of R134a in the surface of the tube is determined by the following formula:

$$\alpha_a = 0,72.\sqrt[4]{\frac{\Delta h.\rho.\lambda^3.g}{\nu.d_{ng}}}.\left(\frac{n_Z}{2}\right)^{-0,167}.\beta.\theta_a^{-0,25}.\psi_c$$

In which:

$$\Delta h = q_k = 176,07 \text{ (kJ/kg)}$$

ψ_c: the coefficient taking into account other conditions when it condenses on the pipe with wings.

$$\psi_c = 1,3\frac{F_d}{F_{ng}}.E^{0,75}\left(\frac{d_{ng}}{h'}\right)^{0,25} + \frac{F_n}{F_{ng}}$$

which:

E: the performance of the wing (E=1, copper pipes with wings)

h': the height of the Convention of the wing

$$h' = \frac{\pi}{4}.\left(\frac{D^2 - d_{ng}^2}{D}\right)$$

$$h' = \frac{3,14}{4}.\left(\frac{0,02^2 - 0,018^2}{0,02}\right) = 0,00298(m)$$

We deduce:

$$\psi_c = 1,3.\frac{0,0631}{0,1174}.1^{0,75}.\left(\frac{0,018}{0,00298}\right)^{0,25} + \frac{0,0543}{0,1174} = 1,557$$

The density of heat flow in the R134a

$$\alpha_a = 0,72.\sqrt[4]{\frac{176,07.10^3.1154,9.0,0759^3.9,81}{0,925.10^{-6}.0,0181}}.\left(\frac{21}{2}\right)^{-0,167}.2,34.1,557.\theta a^{-0,25}$$

$$\alpha_a = 4758,82.\theta a^{-0,25}$$

We have :

$$q_a = \alpha_a.\theta_a$$

$$= 4758,82.\,\theta a^{-0,25}.\theta_a = 4758,82.\theta_a^{0,75}$$

We have the equation to determine the q_{tr} :

$$\begin{cases} q_w = 3123,43.(7,21 - \theta_a) \\ q_a = 4758,82.\,\theta a^{\;0,75} \end{cases}$$

Solve the equation by iteration method

$$q_{tr} = \frac{(x-1)q_{tr}'^{x} + \theta_m B^x}{x q_{tr}^{x-1} + \frac{B^x}{A}}$$

In which: x =1/k=1/0,75=1,333

A = 3123,43; B = 4758,82

θ_m = 7,21; q_{tr}' = 15763,95 (W/m^2)

We deduce:

$$q_{tr1} = \frac{(1,333-1).15763,95^{1,333} + 7,21.4758,82^{1,333}}{1,333.15763,95^{0,333} + \frac{4758,82^{1,333}}{3123,43}} = 12004,13\ (W/m^2)$$

$$q_{tr2} = \frac{(1,333-1).12004,13^{1,333} + 7,21.4758,82^{1,333}}{1,333.12004,13^{0,333} + \frac{4758,82^{1,333}}{3123,43}} = 11910,01\ (W/m^2)$$

Relative errors:

$$\delta_q = \frac{q_{tr1} - q_{tr2}}{q_{tr2}} = \frac{12004,13 - 11910,01}{11910,01} = 0,0079\%$$

Error 0,0079% so we have q_{tr} = 11910,01 (W/m^2).

The surface area of heat transfer pipe:

$$F_{tr} = \frac{Q_k}{q_{tr}}$$

$$F_{tr} = \frac{1361,15x10^3}{11910.01} = 114,27\ m^2$$

The total length of the pipe in condenser:

$$L = \frac{F_{tr}}{\pi.d_{tr}}$$

$$L = \frac{114{,}27}{3{,}14.0{,}016} = 2274{,}48$$

Preliminary have been calculated and choose m = 22 so the total number is pipes:

$$n = 0{,}75.\ (m^2 - 1) + 1$$
$$= 0{,}75.\ (22^2 - 1) + 1 = 363 \text{ (pipes)}$$

Of the water in the condenser:

$$Z = \frac{n}{n_1} = \frac{363}{163} = 2.2$$

Choose Z = 2 (Choose Z = 2, k value = 8.86 does not meet the requirements)

When it:

$$n = Z.\ n_1$$
$$= 2.163 = 326$$

$$m = \sqrt{\frac{n-1}{0.75}} + 1 = \sqrt{\frac{326-1}{0.75}} + 1 = 21{,}82$$

So we can re-choose *m*, we must chose *m = 22*

We have: $n = 0.75 \times (m^2 - 1) + 1 = 0.75 \times (22^2 - 1) + 1 = 363$*(pipe)*

To use the condenser capacitor, we must retrench 5 row bottom pipes.

The pipe is removed:

$$n' = i.\frac{m+1}{2} + \sum_{n=1}^{n=i-1} n_i$$

$$n' = 5.\frac{22+1}{2} + 1 + 2 + 3 + 4 = 68$$

With: *i* is the number of rows the pipe removed.

So the number of rows the pipe remaining: n” = n – n’ = 363 – 68 = 295 (pipes)

So missing 31 pipes anymore is enough tube 326 will arrange them in a above condenser.

The length of the tube

$$l = \frac{L}{n}$$

$$l = \frac{2274{,}48}{326} = 6{,}98$$

Tube step:

$$S = 1{,}3.\ D = 1{,}3.0{,}02 = 0{,}026 \text{ (m)}$$

Sieve diameter:

$$D = m.S = 42 \,.\, 0,026 = 1.092(m).$$

The ratio:

$$K = \frac{l}{D} = \frac{6,98}{1.092} = 6,4$$

I have according Reference [2] page 158, k = 6,4 range allows (4÷8)

6.1.3 Hydrodynamic condenser calculation:

In addition to calculating the average evaporation heat transfer, it is also the resistance of the water when cold through condenser. According to formula 9.25 pages 353 Reference [2]:

Hindrance to the water through the condenser:

$$\Delta P = \left(\lambda \frac{L}{d_{tr}} + \xi_v + 1 + \frac{\xi_v + 1}{z}\right) . \frac{\omega^2 . \rho}{2} . z$$

In which:

ξ_v - coefficient of local resistance when water in tube: $\xi_v = 0,5$.

L - length of average between the two manifestations: L = 4,5 m

d_{tr} - diameter of pipe: $d_{tr} = 0,016$ m

z - number of lines water in equipment: z = 2

ω - the velocity of water flow in pipes: ω= 2 m/s

ρ - the density of cold water: ρ = 994,125 kg/m^3

λ - coefficient of friction.

Because the water in the tube in a turbulent state, so for copper pipe friction coefficient is calculated as follows:

$$\lambda = \frac{0,3164}{Re^{0,25}} = \frac{0,3164}{43284,18^{0,25}} = 0,022 \qquad [2]$$

So pressure drop water through the condenser:

$$\Delta P = \left(0,022 . \frac{4,5}{0,016} + 0,5 + 1 + \frac{0,5 + 1}{4}\right) . \frac{2^2 . 994,125}{2} . 4 = 64121,06\ Pa$$

6.1.4 Strength condenser calculation:

A. Strength case condenser calculation:

Evaporator in the air conditioning system is the low pressures side pressure device. Therefore, we have to calculate reliable equipment to ensure the safety of the device when operating …

Due to the structure of the average evaporation cylindrical geometry, so under pressure. The thickness of the cylindrical body S is chosen to satisfy the following conditions: (Formula 10.1page 364 Reference [3])

$$S \geq \frac{P_R\ .\ \mathrm{D_{tr}}}{2\ .\ [\sigma]\ .\ \phi_d - P_R} + C$$

In which:

P_R - Calculation of the pressure equipment, MPa. According to table 10.1 pages 360 Reference [3] was chosen: $P_R = 12$ bar $= 1{,}2$ MPa..

$[\sigma]$ - Allowing stress of metal fabrication body average, MPa. According to table 10.2 pages 361 Reference [3], choose body building materials per evaporation is steel CCT38, with the calculation of the wall temperature is: $t = 42^0C$ we have: $[\sigma] = 138{,}35$ MPa

D_{tr} - Diameter of the body evaporation comment: $D_{tr} = 754\ mm$

ϕ_d - Vertical weld strength coefficient, $\phi_d = 0{,}9$ (Reference [3], pages 364, table 10-3)

C - Additional thickness, mm

$$C = C_1 + C_2 + C_3$$

C_1 - The additional thickness to compensate for corrosion when exposed to hazardous substances: $C_1 = 0{,}001$ m

C_2 - additional thickness to compensate for the negative thickness tolerance:

$C_2 = 0{,}001$ m

C_3 - the additional thickness due to the relative thickness of Votes thinning during pulling, stamping, bending, etc…: $C_3 = 0{,}001$ m

So: $S \geq \frac{1{,}2\ .\ 0{,}754}{2\ .\ 138{,}35\ .\ 0{,}9 - 1{,}2} + 0{,}003 = 0{,}0066$ m

Choose the standard TEMA: $S = 0{,}0095\ m = 9{,}5\ mm$ (Table CB 3.13, pages 5.3-1 Reference [4]).

Condenser has the following dimensions:

$D_{tr} = 0{,}754\ m$

$D_{ng} = D_{tr} + 2.S = 0{,}754 + 2\ .0{,}0095 = 0{,}773\ m$

B. Calculate the thickness of floating tube sheet:

In the condenser the ground is soldered to the cylindrical body of the condenser. The copper tube is tight on the floor, so that the thickness of the floor to ensure tight tube and must meet the following conditions:

$$S_m \geq 0{,}5\ .\ \mathrm{D}_E \sqrt{\frac{|\ \mathrm{P}_o - P_R|}{[\ \sigma]}} + C$$

In which:

P_R - Calculate the pressure outside the tube, is the calculation of the pressure equipment. According to table 10.1 pages 360 Reference [3] was chosen: $P_R = 12$ bar $= 1{,}2$ MPa.

P_o - Calculate the pressure inside the pipe: $P_o = 1{,}5$ bar =0,15 MPa

$[\sigma]$ - Allowing stress of metal fabrication place, MPa. According to the table 10.2 pages 367 Reference [3], choose body building materials per evaporation is steel CCT38, with the calculation of the wall temperature is: $t = 42^0 C$ we have: $[\sigma] = 138{,}35$ MPa.

D_E - The diameter of the circle can accommodate the largest in the area do not have the tube on the floor: $D_E = 104{,}15$ mm

C - Additional section thickness: $C = 0{,}003$ m

So: $S_m \geq 0{,}5\ .\ \mathrm{D}_E \sqrt{\frac{|\ \mathrm{P}_o - P_R|}{[\ \sigma]}} + C = 0{,}5.0{,}10415. \sqrt{\frac{|\ 0{,}15-1{,}2|}{138{,}35}} + 0{,}003 = 0{,}0075$ m

We choose the thickness of floating tube sheet: $S_m = 0{,}0075$ m $= 7{,}5$ mm

C. Strength for the lid calculation:

With condenser cylindrical geometry, we use a curved lid can be removed to open the assembly with two top flange cylindrical body. I choose the bottom of the device is curved circular curved bottom edge boards (Figure 10-4 c, pages 370 Reference [3]).

Round cap thickness is determined as follows: (Formula pages 370 Reference [3])

$$S_n \geq \frac{\mathrm{P}_R\ .\ \mathrm{R}}{2\ .\ \phi_d\ .\ [\sigma] - 0{,}5\ .\ \mathrm{P}_R} + C$$

In which:

R - radius of the curved lid, m.

$R = D_{\mathrm{tr}} = 0{,}754$ m

$H_{\mathrm{tr}} = 0{,}25\ .\ \mathrm{D}_{\mathrm{tr}} = 0{,}25\ .\ 0{,}754 = 0{,}1885$ m - The height of the inside of the lid.

ϕ_d - Weld strength coefficient along, $\phi_d = 0{,}9$

P_R - Calculation of pressure equipment: $P_R = 1{,}2$ MPa

$[\sigma]$ - Allowing stress of metal fabricated cap:

$[\sigma] = 138{,}35\text{MPa}$.

C - Additional thickness: $C = 0{,}003$ m

$$\text{So: } S_n \geq \frac{1{,}2 \,.\, 0{,}754}{2 \,.\, 0{,}9 \,.\, 138{,}35 - 0{,}5 \,.\, 1{,}2} + 0{,}003 = 0{,}0066 \text{ m}$$

We choose the thickness of the lid: $S_n = 0{,}007 \text{ m} = 7 \text{ mm}$

6.2 CONDENSER OPTIONS

Specification | 60Hz

LG Life's Good

R134a (60Hz)

Model		Units	RCWW022CA2A	RCWW024CA2A	RCWW026CA2A	RCWW028CA2A	RCWW032CA2A	RCWW036CA2A	RCWW040CA2A
Standard Condition	Cooling capacity	kW	726	783	849	912	1,095	1,217	1,298
		usRT	206.4	222.5	241.5	259.3	311.4	346.1	369.1
	Input Power	kW	151.87	164.01	177.89	182.51	227.03	240.05	261.89
	COP		4.8	4.8	4.8	5	4.8	5.1	5
AHRI Conditions	Cooling capacity	kW	734.53	791.98	859.6	922.88	1,108.64	1,231.96	1,314.09
		usRT	208.9	225.2	244.4	262.4	315.2	350.3	373.6
	Input Power	kW	145.73	157.38	170.67	175.14	217.82	230.33	251.25
	COP		5	5	5	5.3	5.1	5.3	5.2
	IPLV		6.44	6.43	6.47	6.74	6.53	6.85	6.73
General Unit Data	Number of Circuits		2	2	2	2	2	2	2
	Refrigerant, R-134a	kg	95 / 95	100 / 100	110 / 110	115 / 115	145 / 145	160 / 160	175 / 175
	Oil Charge	l	18 / 18	20 / 20	23 / 23	20 / 20	28 / 28	28 / 28	28 / 28
Weight	Shipping Weight	kg	4,460	4,600	4,720	4770	5580	5910	5930
	Operating Weight	kg	4,780	4,940	5080	5150	6040	6430	6480
Compressors	Compressor type					Semi-hermetic twin screw			
	Quantity	EA	2	2	2	2	2	2	2
Condenser	Evaporator type	kW				Shell and Tube			
	Water Volume	kW	59	61	61	65	80	86	86
	Max. Water Pressure	MPa	1	1	1	1	1	1	1
	Max. Refrigerant Pressure	Mpa	1	1	1	1	1	1	1
	Min. Cooling Water Flow Rate	l/s	13.6	14.6	14.6	16.9	19	21.6	21.6
	Max. Cooling Water Flow Rate	l/s	54.4	58.6	58.6	67.7	76	86.5	86.5
	Water Connections	DN	150	150	150	150	200	200	200
Evaporator	Evaporator type					Shell and Tube			
	Water Volume	l	67	83	83	87	92	112	112
	Max. Water Pressure	MPa	1	1	1	1	1	1	1
	Max.Refrigerant Pressure	Mpa	1	1	1	1	1	1	1
	Min.Chilled Water Flow Rate	l/s	12.6	13.8	13.8	15.7	18	20.2	20.2
	Max.Chilled Water Flow Rate	l/s	50.2	55.1	55.1	62.8	71.8	80.9	80.9
	Water Connections	DN	150	150	150	150	200	200	200