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The Development of Calculation and Selection Procedure for Plate Condenser of Vapor Compression Refrigerating Machines

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Abstract

Background: The relevance of the article is determined by the lack of publicly available calculation and selection procedures for plate condensers of refrigerating machines, and the impossibility for the operating organizations to make recalculation of thermal-hydraulic performance of condensers. **Method**: This article proposes procedures for plate condenser calculation and selection, based on semi-empirical design master curves for calculation of heat transfer and pressure drop processes during R407C refrigerant vapor condensation. These methods of calculation and selection are based on design and checking calculation and performance-based calculation method. **Findings**: Algorithm for thermal design calculation is based on the mean temperature difference method. Algorithm for thermal checking calculation is based on the performance method, enabling to reduce the number of iterations. All the obtained procedures can be automated using of public domain programming software. **Improvements**: These findings are of practical value for the organizations that operate plate condensers as part of a vapor compression refrigerating machine.

Keywords: Checking Calculation, Condensation, Design Calculation, Droplet Entrainment, Heat Exchange and Hydraulics Master Curves, Plate Condenser

1. Introduction

In recent years, obsolete equipment of refrigerating machines of industrial facilities and technological systems has been replaced. Previously used shell-and-tube condensers are taken out of service and, as a rule, semi-gasketed or brazed plate heat exchangers are installed in their place. Plate heat exchange apparatuses have higher heat transfer coefficients than the shell-and-tube ones with smooth tubes and they are considerably smaller in size, which is useful in the design and operation. To manufacture plates special stainless steel grades and alloys are used (mainly *AISI 316* steel grade) that have relatively high thermal conductivity and strength. The plates have a special profile enabling significantly turbulize the flow, which ultimately results in reduction of deposits on the heat transfer walls, but it increases the hydraulic resistance of the circuit and increased energy consumption for the coolant pumping.

The main difficulty in the selection, calculation and further improvement of ***Chloro Fluoro Carbons*** (CFC) plate condensers is associated with the fact that they are selected and calculated according to the manufacturers' computer programs, where partial thermal hydraulic

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characteristics obtained as a result of full-scale tests of samples of each item of the standard series are used to write these programs. Manufacturers do not provide the operating organization with complete information about the geometric characteristics of the plates and channels formed by them, and about calculated heat transfer and hydraulic resistance dependencies.

At the same time when the system load changes, the operating organization adds or removes a certain number of plates. This results in creation of a new thermo hydraulic mode in the heat exchanger. Lack of calculated dependencies does not allow the experts of the operating organization to recalculate performance.

The obtained master curves for thermal-hydraulic performance of plate heat exchangers and development of calculation and selection procedures on their basis is rather relevant and useful task from scientific and practical points of view. This will make calculation methods more versatile, facilitate their improvement and enable to further explore the processes occurring in them.

As noted in[1], most part of the cold is produced by means of vapor compression refrigerating machines, structurally composed of a compressor, condenser, throttle and evaporator.

This paper discusses the brazed plate condensers of refrigerating machines. Brazed plate condenser is a welded unit, with the operating medium channels formed by corrugated stainless steel plates. The channels are made by welding two adjacent plates along the side elements and joining them to the respective collectors. The direction of the corrugations of two adjacent plates differs by 180°. The channels for the flow of the medium are formed by two adjacent pairs of plates. Heat exchange surface is formed of individual corrugated plates and the channels for the operating medium have a variable slotted cross-sectional shape in the flow direction with a hydraulic diameter less than 6 mm. Pronounced turbulence also contributes to intensification of heat exchange.

In[1] criterial heat transfer master curves were obtained during R407C refrigerant vapor condensation for various

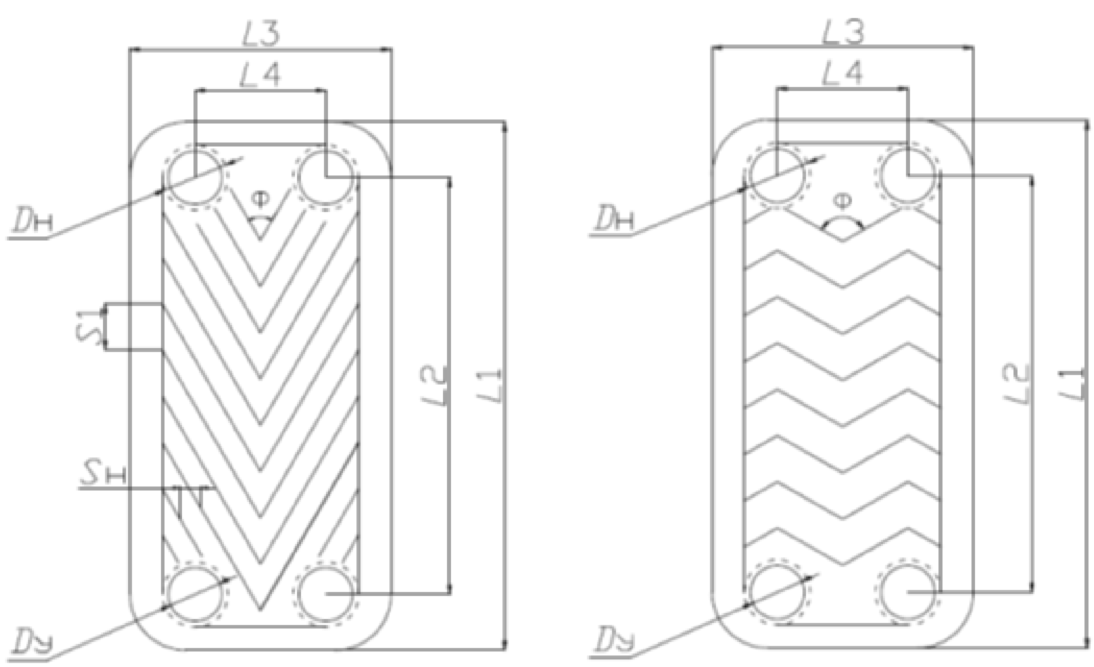

Figure 1. Basic geometrical characteristics of plates.

Table 1. Geometric parameters of the plates*

Type	D_y, mm	F_0, m^2	L_2, mm	L_4, mm
1	20	0.024	40	278
2	20	0.033	50	250
3	100	0.0347	239	862
4	25	0.036	65	242
5	50	0.147	150	520
6	25	0.073	65	446

* d = 0.00465 m at φ = 60°, d = 0.0487 m at φ = 120° and d = 0.00476 m for channels formed by the plates with corrugation angles φ = 60° and φ = 120°

types of corrugated plates with a V-shaped profile of the corrugations, whose opening angle φ was 60° and 120° (Figure 1). At a corrugation angle φ = 60° the longitudinal corrugation pitch S_1 = 26 mm, normal pitch S_n = 12mm. At φ = 120°, S_1 = 10 mm, S_n = 9 mm. Corrugation height h = 3 mm. The plate thickness δ = 0.5 mm. The geometrical characteristics of the plates and the channels formed by them are given in Table 1.

Initial data for the calculation included:

- type of the heated coolant;
- consumption of heated coolant, kg/s;
- initial and final temperature of the heated coolant, °C;
- allowable pressure loss, kPa;
- vapor temperature, °C
- standard size of the plate;
- plate material;
- maximum coolant pressure, MPa;
- maximum coolant temperature, °C.

The following conditions were set for all calculations:

- vapor type - Freon R407C;
- type of the heated coolant - water;
- allowable pressure loss - 50kPa;
- plate material - AISI 316 stainless steel;
- maximum coolant pressure – 2.5 MPa;
- maximum coolant temperature - 150 °C.

The study was performed on the assumption that the saturated steam is used, condensation occurs along the entire length of the plate (condensate is not sub-cooled), heat losses to the environment are not available. Freon vapor temperature at the condenser input was set in the range from 45 to 70°C. Thermal power of the apparatus varied by changing the heated coolant flow and the initial and final temperatures. We considered only one-way capacitors.

The algorithm incorporated temperature dependences to account for changes in thermo-physical properties of coolants:

Freon R407C:

- density of condensate and vapor:

$$\rho_c = -0.0007t^3 + 0.0413t^2 - 4.7477t + 1240.2;$$

$$\rho_v = 0.0006t^3 - 0.0343t^2 + 1.6058t + 16.374;$$

specific enthalpy of condensate and vapor:

$$h_c = 0.0057t^2 + 1.2404t + 201.26;$$

$$h_v = -2 \cdot 10^{-6} \cdot t + 0.0002t^3 - 0.0104t^2 + 0.5891t + 409.31;$$

specific heat capacity of condensate and vapor:

$$c_{pc} = 5 \cdot 10^{-10} \cdot t^6 - 10^{-7} \cdot t^5 + 8 \cdot 10^{-6} \cdot t^4 - 0.003t^3 + 0.0047t^2 - 0.0225t + 1.4304;$$

$$c_{pv} = 4 \cdot 10^{-6} \cdot t^6 - 8 \cdot 10^{-8} \cdot t^5 + 6 \cdot 10^{-6} \cdot t^4 - 0.0002t^3 + 0.0036t^2 - 0.0143t + 0.9675;$$

thermal conductivity of condensate and vapor:

$$\lambda_c = -0.005t + 0.0958;$$

$$\lambda_v = 2 \cdot 10^{-9} \cdot t^4 - 2 \cdot 10^{-7} \cdot t^3 + 9 \cdot 10^{-6} \cdot t^2 - 2 \cdot 10^{-5} \cdot t + 0.0123;$$

dynamical viscosity of condensate and vapor:

$$\mu_c = 0.0055t^2 - 2.2598t + 208.58;$$

$$\mu_v = 5 \cdot 10^{-7} \cdot t^4 - 5 \cdot 10^{-5} \cdot t^3 + 0.0022t^2 + 0.0111t + 11.323;$$

Prandtl number of condensate and vapor:

$$Pr_c = 4 \cdot 10^{-10} \cdot t^6 - 7 \cdot 10^{-8} \cdot t^5 + 6 \cdot 10^{-6} \cdot t^4 - 0.0002t^3 + 0.0034t^2 - 0.0329t + 3.1104;$$

$$Pr_v = 2 \cdot 10^{-10} \cdot t^6 - 4 \cdot 10^{-8} \cdot t^5 + 3 \cdot 10^{-6} \cdot t^4 + 0.0015t^2 - 0.0061t + 0.8989;$$

the coefficient of surface tension of condensate and vapor:

$$\delta_c = 4 \cdot 10^{-7} \cdot t^2 - 0.0002t + 0.0107;$$

$$\delta_v = 3 \cdot 10^{-7} \cdot t^2 - 0.0002t + 0.0111;$$

water:

the coefficient of surface tension σ_c, N/m

$$\sigma_c = 0{,}0797 - 0.0002 \cdot t_{11};$$

$$\rho_w = \frac{997}{0.99534 + 0.466 \cdot 10^{-3} \cdot \frac{1}{2} \cdot (t_{21} - t_{22})};$$

water heat capacity c_{pw}, kJ/(kg·°C)

$$c_{pw} = 4.20511 - 0.00136578 \cdot \frac{1}{2} \cdot (t_{21} + t_{22}) + 0.152341 \cdot 10^{-4} \cdot \frac{1}{4} \cdot (t_{21} + t_{22})^2;$$

water thermal conductivity λ_w, W/(m·°C)

$$\lambda_w = 0.5678 - 0.0017 \cdot \frac{1}{2} \cdot (t_{21} + t_{22}) - 6 \cdot 10^{-6} \cdot \frac{1}{4} \cdot (t_{21} + t_{22})^2;$$

$$\mu_w = \rho_w \cdot 10^{-6} \cdot \left(\exp\left(\exp 33.22999 - 5.93043 \cdot \ln\left(\frac{1}{2} \cdot (t_{21} + t_{22}) + 273\right)\right) - 0.87\right);$$

In[2] similar calculations for 'steam – water' systems were carried out. In[1] an algorithm for obtaining heat transfer and hydraulic resistance master curves is described in detail. Briefly, this algorithm is presented below.

With the help of the programs calculations for brazed plate Freon condensers were verified. As a result, thermal power, Freon Vapor flow rate, final temperatures of water, mean temperature differences and heat transfer coefficients of condensers have been identified for all the heat exchangers. Next, using convective heat exchange dependencies in case of forced coolant flow in the channels of such heat exchangers we calculated convective heat-transfer coefficients for water:

$$\alpha_2 = A_3 \cdot \frac{\lambda_2 \cdot Re_2^n \cdot Pr_2^{0.4}}{d};$$

the vapor and heat transfer surface temperature difference:

$$\Delta t_1 = \Delta t_{\log} - \frac{Q}{F} \cdot \left(\frac{1}{\alpha_2} + \frac{\delta}{\lambda} + 2 \cdot R_{foul}\right);$$

and determined convective heat-transfer coefficients for condensing vapor

$$\alpha_1 = \frac{1}{\frac{1}{k} - \frac{1}{\alpha_2} - \frac{\delta}{\lambda} - 2 \cdot R_{foul}};$$

In[2] the factors were identified that affect the heat transfer intensity: the Reynolds number Re_c, Prandtl number Pr_c, Froude number Fr_c, Weber number We_c, Galileo number Ga for condensate film, the phase transition K, as well as the parameter to account for the effect of the vapor and condensate density ratio $((\rho_c / \rho_v)^{0.5} + 1)$. The process of the moving vapor condensation is characterized by flow separation: in the vicinity of the longitudinal axis of the channel vapor or mist flow (at higher speeds) is moving, with the condensate being on the walls, i.e. there is a dispersed-annular flow regime, the presence or absence of which is directly determined by the Kutateladze number Ku, whose values ranged from 3.72 to 27.14 in the studied regimes.

When generalizing the results of heat exchange calculation, it was required to enter the relationship $2F_0/f_0$ as the determining geometric factor.

Therefore during vapor condensation in the profiled channels the heat transfer master curve is written as follows:

$$Nu_c = A_c \cdot Re^{n} \cdot K^{m_c} \cdot Ku^{n_1} \cdot We^{n_2} \cdot Fr^{n_3} \cdot Ga^{n_4} \cdot \left(\left(\frac{\rho_c}{\rho_v}\right)^{0.5} + 1\right)^{l_c} \cdot \left(\frac{2F_0}{f_0}\right)^{b} \cdot Pr_c^{0.4};$$

The analysis of the obtained data showed small influence of the Froude and Weber numbers. The Kutateladze number analysis showed the existence of dispersed-annular flow regime in the channels[2]. However, its effect on heat transfer process is not essential.

The final results of the heat transfer calculation during vapor condensation were presented in the form of dependence

$$Nu_c = A_c \cdot Re_c^{n_{kc}} \cdot \left(\frac{2F_0}{f_0}\right)^{b} \cdot K^{m_c} \cdot \left(\left(\frac{\rho_c}{\rho_v}\right)^{0.5} + 1\right)^{l_c} \cdot Pr_c^{0.4} \cdot X_{t_0};$$

Table 2. Basic design data

Plate No*	$A_к$	$m_к$	$l_к$	b	n	$\delta Nu_к$, %
φ = 60°						
3						16.9
5	0.0527	-0.068	0.0578	0.1	0.9	19.7
6						7.8
φ = 120°						
1						16.9
2						12.0
3	11.43	0.4	-0.35	-0.55	0.8	19.1
4						13.7
5						19.5
6						8.73
φ = 60° and φ = 120°						
3						13.3
5	0.07	-0.078	-0.16	0.22	0.8	12.0
6						18.2

It should be noted that the process of condensation strongly depends on the temperatures of the heated medium. Therefore, generalization was performed individually for several values of the refrigerant input and output temperature. Values of proportionality factor A_K for each standard size of the plates are given in Table 2. The values of the exponents for the chosen criteria for each layout of plates (φ = 60°, φ = 120° and a mixed arrangement) are constant. Table 2 also shows the investigated ranges of the influencing factors.

When calculating Nu_c, Re_c, *Ku* and *K*, thermo physical properties of the condensate were taken according to the saturation temperature, which was 125-150°C. The Prandtl number varied from 1.208 to 1.405. The maximum sub-cooling of the condensate for the given input conditions did not exceed 24.3 ° C.

The average speed of heated water was been calculated by the continuity equation, and ranged from 0.2 - 1.2 m/s. The hydraulic diameter *d* of the channel was used as a characteristic size.

The Reynolds number at the forced flow of condensate film under the action of friction forces at the phase boundary was calculated as:

$$\mathrm{Re_c} = \frac{\mathrm{Q \cdot L_{calc}}}{\mathrm{F \cdot (h_v - c_{pc} \cdot t_c) \cdot \mu_c}};$$

2. Results and Discussion

The algorithm for thermal design calculation is based on the mean temperature difference method. The algorithm for thermal checking calculation is based on the performance method, enabling to reduce the number of iterations. The procedures for calculating and selecting plate condensers of cooling systems are described below.

These procedures make it possible to generalize the results of many authors[3-13] who did not give their calculation procedures in their works.

2.1 Thermal Design Calculation

Initial data for design calculation are:

- vapor temperature at the heater inlet t_{11};
- water flow rate G_2;
- initial t_{21} and final t_{22} temperatures of water.

Additional conditions:

- vapor condensates on the entire heat exchange surface, condensate is not sub-cooled;
- single-pass heater is used, both in terms of vapor and heated water;
- allowable pressure losses of the heated coolant Δp2 = 30 kPa;

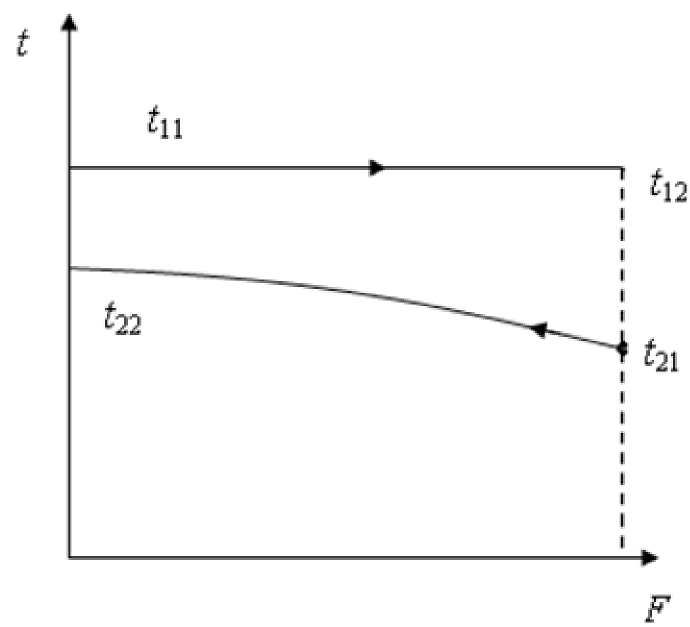

Figure 2. Condensation process diagram.

- thermal conductivity of the plate material $\lambda wall=20.0$ W/(m·K);
- heat transfer wall thickness $\delta wall=0.0005$ m.

Schematically the condensation process is shown in *t*-*F*-diagram (Figure 2).

Design calculation is performed according to the algorithm on the basis of master curves in the following sequence:

1. Thermal power of the apparatus is calculated according to

$$Q = G_2 \cdot c_{p2} \cdot (t_{22} - t_{21});$$

2. Vapor flow rate *D*:

$$D = \frac{Q}{h_{11} - h_{12}};$$

3. Heat transfer coefficient *k* is preset;

4. Logarithmic mean temperature difference Δt_{log} is calculated according to

$$\Delta t_{log} = \frac{(t_{11} - t_{22}) - (t_{12} - t_{21})}{\ln(\frac{(t_{11} - t_{22})}{(t_{12} - t_{21})})};$$

5. Average temperatures of coolants

$$t_1 = 0.5 \cdot (t_{11} + t_{12});$$

$$t_2 = 0.5(t_{21} + t_{22});$$

6. Design heat exchange surface area

$$F_p = \frac{Q}{k \cdot \Delta t_{log}};$$

7. Standard size with appropriate values of F_0 and f_0 (given in Table 1), coefficients in heat exchange and hydraulics dependencies (given in 1) is selected;
8. Number of plates

$$n_{pl} = \frac{F_p}{F_0} + 2;$$

9. Number of channels:

in case of even number of plates:

$$n_1 = \frac{n_{pl}}{2} - 1;$$

for cold coolant

$$n_2 = \frac{n_{pl}}{2};$$

in case of odd number of plates

$$n_1 = n_2 \frac{n_{pl} - 1}{2};$$

10. Open flow area of the plate package from the hot and cold sides

$$f_1 = f_0 \cdot n_1;$$

$$f_2 = f_0 \cdot n_2;$$

11. Actual heat exchange surface area

$$F = F_0 \cdot (n_{pl} - 2);$$

12. With the known flow rates by continuity equation velocity of vapor, condensate and water is calculated as:

$$\omega_v = \frac{D}{\rho_{v\pi} \cdot f_1};$$

$$\omega_c = \frac{D}{\rho_c \cdot f_1};$$

$$\omega_w = \frac{D}{\rho_w \cdot f_2};$$

13. The Reynolds numbers of condensate and water are calculated by

$$Re_c = \frac{\omega_c \cdot \rho_c \cdot d}{\mu_c};$$

$$Re_w = \frac{\omega_w \cdot \rho_w \cdot d}{\mu_w};$$

14. Water side convective heat-transfer coefficient α_2 using values *A* and *n* from[1], according to the selected standard size of plates:

$$\alpha_2 = A_2 \cdot \frac{\lambda_2 \cdot Re_2^n \cdot Pr_2^{0.4}}{d}$$

15. "Vapor-wall" temperature difference

$$\Delta t_1 = \Delta t_{log} - \frac{Q}{F} \cdot \left(\frac{1}{\alpha_2} + \frac{\delta_{wall}}{\lambda_{wall}}\right);$$

16. The Kutateladze number Ku is calculated according to the ratio:

$$Ku = \frac{\rho_v^{0.5} \cdot \omega_v}{(\sigma \cdot g \cdot (\rho_c - \rho_v)^{0.25})}$$

17. Vapor side convective heat-transfer coefficient α_1 according to the selected standard size of plates

$$\alpha_1 = \frac{\lambda_c}{d} \cdot A_c \cdot Re_c^{0.9} \cdot \left(\frac{2F_0}{f_0}\right)^b \cdot K^{m_c} \cdot \left(\left(\frac{\rho_c}{\rho_v}\right)^{0.5} + 1\right)^{l_k} \cdot Ku^{n_1} \cdot Pr_c^{0.4} \cdot X_{t_0};$$

18. Wall temperature t_{wall}

$$t_{wall} = \frac{t_1 \cdot \alpha_1 + t_2 \cdot \alpha_2}{\alpha_1 + \alpha_2};$$

19. Verified values α_1 and α_2

$$\alpha_1 = \alpha_1 \cdot \left(\frac{Pr_{condwall}}{Pr_{conds}}\right)^{0.25};$$

$$\alpha_2 = \alpha_2 \cdot \left(\frac{Pr_{liqwall}}{Pr_{liq}}\right)^{0.25};$$

20. Heat-transfer coefficients for clean and dirty heat exchange surface (with regard to contamination factor φ=0.9)

$$k_0 = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_2}};$$

$$k = 0.9 \cdot k_0;$$

Heat exchange surface reserve

$$\delta F\% = \frac{F - F_p}{F_p} \cdot 100\%;$$

If there is no heat exchange surface reserve, the recalculation is carried out adding some plates or another standard size of plates is selected.

2.2 Thermal Checking Calculation

2.2.1 Verification Option No. 1

Initial data are:

- vapor temperature at the heater inlet t_{11};
- initial and final temperatures of water t_{12} and t_{22};
- geometric characteristics of the heater (heat exchange surface area of one plate F_0, flow area of one channel between the plates f_0, hydraulic diameter *d*);
- total number of plates n_{pl};
- thermal power of the apparatus *Q*.

Additional conditions:

- vapor condensates on the entire heat exchange surface, condensate is not sub-cooled;
- single-pass heater is used, both in terms of vapor and cooled water;

- allowable pressure losses of the heated coolant Δp_2 =30 kPa;
- thermal conductivity of the plate material λ_{wall}=20.0 W/(m·K);
- heat transfer wall thickness δ_{wall}=0.0005 m.

Calculation was performed as follows:

1. Number of channels by dependencies in case of even number of plates:

$$n_1 = \frac{n_{pl}}{2} - 1;$$

for cold coolant

$$n_2 = \frac{n_{pl}}{2};$$

in case of odd number of plates

$$n_1 = n_2 \frac{n_{pl} - 1}{2};$$

2. Open flow area of the plate package from the hot and cold sides by equations:

$$f_1 = f_0 \cdot n_1;$$

$$f_2 = f_0 \cdot n_2;$$

3. Logarithmic mean temperature difference Δt_{log} according to

$$\Delta t_{log} = \frac{(t_{11} - t_{22}) - (t_{12} - t_{21})}{\ln(\frac{(t_{11} - t_{22})}{(t_{12} - t_{21})})};$$

4. Actual heat exchange surface area by dependency

$$F = F_0 \cdot (n_{pl} - 2);$$

5. Vapor flow rate D

$$D = \frac{Q}{h_{11} - h_{12}};$$

6. Water flow rate G_2

$$G_2 = \frac{Q}{c_{p_2} \cdot (t_{22} - t_{21})};$$

7. Velocities of vapor, condensate and water by dependencies

$$\omega_v = \frac{D}{\rho_v \cdot f_1};$$

$$\omega_c = \frac{D}{\rho_c \cdot f_1};$$

$$\omega_w = \frac{D}{\rho_w \cdot f_1};$$

8. The Reynolds numbers of condensate and of water

$$Re_{kc} = \frac{\omega_c \cdot \rho_c \cdot d}{\mu_c};$$

$$Re_w = \frac{\omega_w \cdot \rho_{Bw} \cdot d}{\mu_w};$$

9. Water side convective heat-transfer coefficient α_2 with regard to coefficients A and n according to the standard size of the plate[1]

$$\alpha_2 = A_2 \cdot \frac{\lambda_2 \cdot Re_2^n \cdot Pr_2^{0.4}}{d}$$

10. “Vapor-wall” temperature difference by dependency

$$\Delta t_1 = \Delta t_{log} - \frac{Q}{F} \cdot \left(\frac{1}{\alpha_2} + \frac{\delta_{wall}}{\lambda_{wall}}\right);$$

11. The Kutateladze number Ku according to

$$Ku = \frac{\rho_v^{0.5} \cdot \omega_v}{(\sigma \cdot g \cdot (\rho_c - \rho_v)^{0.25})}$$

12. Vapor side convective heat-transfer coefficient α_1

$$\alpha_1 = \frac{\lambda_c}{d} \cdot A_c \cdot Re_c^{0.9} \cdot \left(\frac{2F_0}{f_0}\right)^b \cdot K^{m_c} \cdot \left(\left(\frac{\rho_c}{\rho_v}\right)^{0.5} + 1\right)^{l_c} \cdot Ku^{n_1} \cdot Pr_c^{0.4} \cdot X_{t_0};$$

13. Wall temperature (at a first approximation) by dependency

$$t_{wall} = \frac{t_1 \cdot \alpha_1 + t_2 \cdot \alpha_2}{\alpha_1 + \alpha_2};$$

14. Verified values of convective heat transfer coefficients with regard to the influence of lateral temperature gradient on the thermo physical properties by dependencies

$$\alpha_1 = \alpha_1 \cdot \left(\frac{Pr_{condwall}}{Pr_{conds}}\right)^{0.25};$$

$$\alpha_2 = \alpha_2 \cdot \left(\frac{Pr_{liqwall}}{Pr_{liq}}\right)^{0.25};$$

15. Heat-transfer coefficients for clean and dirty heat exchange surface

$$k_0 = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_2}};$$

$$k = 0.9 \cdot k_0;$$

16. Heat equivalence ratios

$$\omega_1 = \frac{D \cdot c_{pc}}{G_2 \cdot c_{p2}};$$

$$\omega_2 = \frac{1}{\omega_1};$$

17. Number of transfer units of coolants at the non-contaminated heat exchange surface

$$N_{10} = \frac{k_0 \cdot F}{D \cdot c_{pc}}$$

$$N_{20} = \frac{k_0 \cdot F}{G_2 \cdot c_{p2}}$$

18. Number of transfer units of coolants at the contaminated heat exchange surface

$$N_1 = \frac{k \cdot F}{D \cdot c_{pc}};$$

$$N_2 = \frac{k \cdot F}{G_2 \cdot c_{p2}};$$

19. Cooling and heating performance with the non-contaminated heat exchange surface

$$\varepsilon_{20} = \frac{1 - \exp(-N_{20} \cdot (1 - \omega_2))}{1 - \omega_2 \cdot \exp(-N_{20} \cdot (1 - \omega_2))};$$

20. Cooling and heating performance with the contaminated heat exchange surface

$$\varepsilon_2 = \frac{1 - \exp(-N_2 \cdot (1 - \omega_2))}{1 - \omega_2 \cdot \exp(-N_2 \cdot (1 - \omega_2))};$$

21. Design values of thermal power for non-contaminated and at the contaminated heat exchange surface

$$Q_{p0} = G_2 \cdot c_{p2} \cdot (t_{11} - t_{21}) \cdot \varepsilon_{10};$$

$$Q_p = G_2 \cdot c_{p2} \cdot (t_{11} - t_{21}) \cdot \varepsilon_1;$$

22. Thermal power reserve for clean and contaminated heat exchange surface

$$\%Q_{po} = \frac{Q_{p0} - Q}{Q} \cdot 100\%;$$

$$\%Q_{p} = \frac{Q_{p} - Q}{Q} \cdot 100\%;$$

23. Design final value of condensate t_{12} and of water t_{22} temperatures

$$t_{12} = t_{11} - (t_{11} - t_{21}) \cdot \varepsilon_1;$$

$$t_{22} = t_{21} + (t_{11} - t_{21}) \cdot \varepsilon_1;$$

Next recalculation was performed based on the values obtained, after which the values Q_{po}, Q_{p}, t_{22} were compared. If the difference is more than 5%, the new recalculation is done.

The recalculation was carried out in view of the fact that the value of the wall temperatures t_{wall} is known from the previous iteration

$$K = \frac{r}{c_{pc} \cdot (t_{11} - t_{wall})};$$

After the last iteration the thermal resistance value of the contamination layer was calculated:

$$R = \frac{1}{2} \cdot \left(\frac{1}{k} - \frac{1}{k_0}\right);$$

2.2.2 Verification Option No. 2

Initial data are:

- vapor temperature at the heater inlet t_{11};
- initial and final temperatures of water t_{21};
- geometric characteristics of the heater (heat exchange surface area of one plate F_0, flow area of one channel between the plates f_0, hydraulic diameter d);
- total number of plates n_{pl};
- flow rates of coolants D and G_2.

Additional conditions:

- vapor condensates on the entire heat exchange surface, condensate is not sub-cooled;
- single-pass heater is used, both in terms of vapor and heated water;
- allowable pressure losses of the heated coolant Δp_2=30 kPa;
- thermal conductivity of the plate material λ_{wall}=20.0 W/(m·K);
- heat transfer wall thickness δ_{wall}=0.0005 m.

Calculation was performed as follows:

1. Number of channels in case of even number of plates:

$$n_1 = \frac{n_{pl}}{2} - 1;$$

for cold coolant

$$n_2 = \frac{n_{pl}}{2};$$

in case of odd number of plates

$$n_1 = n_2 \frac{n_{pl} - 1}{2};$$

2. Open flow area of the plate package from the hot and cold sides

$$f_1 = f_0 \cdot n_1;$$

$$f_2 = f_0 \cdot n_2;$$

3. Vapor h_{11} and condensate h_{12} enthalpies by saturation temperature according to[1];
4. Thermal power of the apparatus

$$Q = D \cdot (h_{11} - h_{12});$$

5. Average temperature of the heated coolant was assumed as equal to the preset temperature t_{21};

6. Temperature t_{22}

$$t_{22} = t_{21} + \frac{Q}{G_2 \cdot c_{p2}};$$

7. Logarithmic mean temperature difference Δt_{log} according to:

$$\Delta t_{log} = \frac{(t_{11} - t_{22}) - (t_{12} - t_{21})}{\ln(\frac{(t_{11} - t_{22})}{(t_{12} - t_{21})})};$$

8. Actual heat exchange surface area

$$F = F_0 \cdot (n_{pl} - 2);$$

9. Velocities of vapor, condensate and water

$$\omega_v = \frac{D}{\rho_v \cdot f_1};$$

$$\omega_c = \frac{D}{\rho_c \cdot f_1};$$

$$\omega_w = \frac{D}{\rho_w \cdot f_1};$$

10. Water side convective heat-transfer coefficient α_2 with regard to coefficients *A* and *n* according to the standard size of the plate[1];

$$\alpha_2 = A_2 \cdot \frac{\lambda_2 \cdot Re_2^n \cdot Pr_2^{0.4}}{d}$$

11. “Vapor-wall” temperature difference

$$\Delta t_1 = \Delta t_{log} - \frac{Q}{F} \cdot \left(\frac{1}{\alpha_2} + \frac{\delta_{wall}}{\lambda_{wall}}\right);$$

12. The Kutateladze number according to

$$Ku = \frac{\rho_v^{0.5} \cdot \omega_v}{(\sigma \cdot g \cdot (\rho_c - \rho_v)^{0.25})}$$

13. Vapor side convective heat-transfer coefficient α_1

$$\alpha_1 = \frac{\lambda_c}{d} \cdot A_c \cdot Re_c^{0.9} \cdot \left(\frac{2F_0}{f_0}\right)^b \cdot K^{m_c} \cdot \left(\left(\frac{\rho_c}{\rho_v}\right)^{0.5} + 1\right)^{l_c} \cdot Ku^{n_1} \cdot Pr_{kc}^{0.4} \cdot X_{t_0};$$

14. Wall temperature (at a first approximation)

$$t_{wall} = \frac{t_1 \cdot \alpha_1 + t_2 \cdot \alpha_2}{\alpha_1 + \alpha_2};$$

15. Verified values of convective heat transfer coefficients with regard to the influence of lateral temperature gradient on the thermophysical properties

$$\alpha_1 = \alpha_1 \cdot \left(\frac{Pr_{condwall}}{Pr_{conds}}\right)^{0.25};$$

$$\alpha_2 = \alpha_2 \cdot \left(\frac{Pr_{liqwall}}{Pr_{liq}}\right)^{0.25};$$

16. Heat-transfer coefficients for clean and dirty heat exchange surface (with regard to contamination factor φ=0.9)

$$k_0 = \frac{1}{\frac{1}{\alpha_1} + \frac{\delta_{wall}}{\lambda_{wall}} + \frac{1}{\alpha_2}};$$

$$k = 0.9 \cdot k_0;$$

17. Heat equivalence ratios

$$\omega_1 = \frac{D \cdot c_{pc}}{G_2 \cdot c_{p2}};$$

$$\omega_2 = \frac{1}{\omega_1};$$

18. Number of transfer units of coolants at the non-contaminated heat exchange surface

$$N_{10} = \frac{k_0 \cdot F}{D \cdot c_{pc}}$$

$$N_{20} = \frac{k_0 \cdot F}{G_2 \cdot c_{p2}}$$

19. Number of transfer units of coolants at the contaminated heat exchange surface

$$N_1 = \frac{k \cdot F}{D \cdot c_{pc}};$$

$$N_2 = \frac{k \cdot F}{G_2 \cdot c_{p2}};$$

20. Cooling and heating performance with the non-contaminated heat exchange surface

$$\varepsilon_{20} = \frac{1 - \exp(-N_{20} \cdot (1 - \omega_2))}{1 - \omega_2 \cdot \exp(-N_{20} \cdot (1 - \omega_2))};$$

21. Cooling and heating performance with the contaminated heat exchange surface

$$\varepsilon_2 = \frac{1 - \exp(-N_2 \cdot (1 - \omega_2))}{1 - \omega_2 \cdot \exp(-N_2 \cdot (1 - \omega_2))};$$

22. Design values of thermal power for non-contaminated and at the contaminated heat exchange surface

$$Q_{p0} = G_2 \cdot c_{p2} \cdot (t_{11} - t_{21}) \cdot \varepsilon_{10};$$

$$Q_p = G_2 \cdot c_{p2} \cdot (t_{11} - t_{21}) \cdot \varepsilon_1;$$

23. Thermal power reserve for clean and contaminated heat exchange surface

$$\%Q_{po} = \frac{Q_{p0} - Q}{Q} \cdot 100\%;$$

$$\%Q_P = \frac{Q_p - Q}{Q} \cdot 100\%;$$

24. Design final value of condensate t_{11} and of water t_{22} temperatures

$$t_{12} = t_{11} - (t_{11} - t_{21}) \cdot \varepsilon_1;$$

$$t_{22} = t_{21} + (t_{11} - t_{21}) \cdot \varepsilon_1;$$

Next recalculation was performed based on the values obtained, after which the values Q_{po}, Q_p, t_{22} were compared. If the difference is more than 5%, the new recalculation is done.

The recalculation is carried out in view of the fact that the value of the wall temperatures t_{wall} is known from the previous iteration

$$K = \frac{r}{c_{pc} \cdot (t_{11} - t_{wall})};$$

After the last iteration the thermal resistance of the contamination layer was calculated:

$$R = \frac{1}{2} \cdot \left(\frac{1}{k} - \frac{1}{k_0}\right);$$

2.3 Hydraulic Calculations

Under the condition of the problem allowable pressure losses on the the heated coolant side make 30kPa. The calculation was made by using the hydraulic resistance data summarizing results during the flow of coolants in water-to-water heaters[1].

1. The Reynolds number for water in the collector, Re_{coll}:

$$Re_{coll} = \frac{G_2}{0.785 \cdot D_i \cdot \mu_2};$$

2. Pressure losses in the water circuit, Δp_2:

$$\Delta p_1 = \xi \cdot \frac{\rho_п \cdot \omega_0^2}{2} \cdot \left(\frac{2 \cdot F_0}{f_0}\right) \cdot \left(\frac{\mathrm{Pr}_{\mathrm{condwall}}}{\mathrm{Pr}_{\mathrm{conds}}}\right)^{0.65} \cdot \left[1 - \frac{k \cdot \Delta t_1 \cdot \left(\frac{2 \cdot F_0}{f_0}\right) \cdot \left(\frac{1}{\rho_v} - \frac{1}{\rho_{kc}}\right)}{2 \cdot 1000 \cdot r \cdot \omega_0}\right];$$

3. Conclusion

TThe procedures obtained on the basis of heat transfer and hydrodynamic master curves enabled to make the appropriate calculations without using computer programs of manufactures of plate condensers for vapor compression refrigerating machines. The maximum deviation of the resulting data with regard to convective heat-transfer coefficients was 8.3%, and 10.2% with regard to hydraulic resistance. The obtained data at first glance may seem overestimated, but it should be taken into account that the calculation was carried out for the steam and condensate system with phase transformations. The deviation of the indicators up to 20% is considered a normal practice. Therefore the proposed procedures may be implemented as a program which will automatically recalculate the thermal-hydraulic performance of plate condensers for any changes of the affecting factors.

3.1 Symbol List

D and *G* – vapor and water flow rates; *k* – heat transfer coefficient; α – convective heat transfer coefficient; α_1 – heat transfer coefficient from vapour to wall; α_1 – heat transfer coefficient from wall to water; δNu – root-mean-square error for Nusselt numbers; R_{foul} – fouling resistance; λ – thermal conductivity of the plate material; λ_c, c_{pc}, ν_c, μ_c – thermal conductivity, isobar heat capacity and kinematic and dynamical viscosity of condensate at saturation temperature; δ – plate thickness; *Q* – thermal power of heat exchanger; F_0 – heat exchange surface area of one plate; *F* – heat exchange surface area of heat exchanger; f_0 – cross-section area of one inter-plate channel; φ – plate corrugation angle; h_v – vapor enthalpy; *r* – latent heat of vaporization; L_1, L_3 – plate height and width; L_2, L_4 – vertical and horizontal center-to-center distances between neighboring coolant inlets and outlets; L_{calc} – total length of the plate; D_o – outer diameter of the nozzle; D_i – inner diameter of the nozzle; g – gravitational acceleration; ρ_v – vapor density; w_v – vapor velocity; $Pr_{condwall}$ – condensate Prandtl number is calculated by a wall temperature; $\mathrm{Pr}_{\mathrm{conds}}$ – condensate Prandtl number is calculated by a saturation temperature; $\mathrm{Pr}_{\mathrm{liqwall}}$ – water Prandtl number is calculated by a wall temperature; $\mathrm{Pr}_{\mathrm{liq}}$ – water Prandtl number is calculated by a saturation temperature;

$K = \frac{r}{c_{pc} \cdot \Delta t_1}$ – the phase transition number; $Ku = \frac{\rho^{0.5} \cdot \omega_v}{[\sigma \cdot g \cdot (\rho_c - \rho_v)]^{0.25}}$ – the Kutateladze number .

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